Contrastive Study of Flow and Heat Transfer Characteristics in a Helically Coiled Tube under Uniform Heating and One-side Heating

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Abstract: One-side heated helically coiled tubes, which are generally applied in various industrial applications such as the water cooled wall in power plant boilers though, have not been thoroughly studied. To investigate the flow and heat transfer characteristics in this case, numerical simulation of the flow in a helically coiled tube is performed under uniform and non-uniform (heating on the inner coil side wall) heat flux boundary conditions for both laminar and turbulent flows. Temperature distributions, secondary flow distributions, average Nusselt number variation with respect to Reynolds number and local Nusselt number along the periphery on the wall in the fully developed section are discussed contrastively under the two different heating conditions. It is found that the secondary flow distributions are hardly affected by changing heating method, however, a larger temperature gradient can be found for one-side heating condition. The average Nusselt numbers are close for laminar flow under the two heating methods, but one-side heating shows 7%-10% lower average Nusselt numbers than uniform heating for turbulent flow, thus a new correlation of average Nusselt number for turbulent flow and one-side heating is proposed. Furthermore, a special point on the inner wall where the local Nusselt numbers are almost the same when carrying out different heating conditions in laminar and turbulent flows is found, which should be useful for measuring unknown parameters.

Keywords: helically coiled tube; flow and heat transfer characteristics; one-side heating condition

Nomenclature			
A	Area (m^2)	*Greek symbols*	
b	Coil pitch (mm)	δ	Curvature ratio
Cp	Specific capacity ($J \cdot kg^{-1} \cdot$	μ	Viscosity ($kg \cdot m^{-1} \cdot s^{-1}$))

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	K^{-1})		
d	Tube diameter (mm)	ρ	Density (kg· m^{-3})
D	Coil diameter (mm)	τ	Shear stress (kg· $m^{-1} \cdot s^{-2}$)
De	Dean number, $Re\sqrt{\delta}$	Ψ	Circumferential angle
f	Friction factor	ω	Mass flux (kg· $m^{-2} \cdot s^{-1}$)
k	Thermal conductivity (W· $m^{-1} \cdot K^{-1}$)	*Subscripts*	
N	Grid number	av	Average
Nu	Nusselt number	bu	Bulk
Pr	Prandtl number	lo	Local
q	Heat flux (W· m^{-2})	one	One-side heating
Re	Reynolds number	uni	Uniform heating
T	Temperature (K)	w	Wall
V	Velocity (m· s^{-1})		

1. Introduction

It is known that due to the existence of secondary flow, curved tubes perform better in heat transfer compared with straight tubes [1]. In addition, owing to the compact structure, it requires smaller room for installation, and the less melding lines make it safer [2]. Therefore the helically coiled tubes are widely used in solar energy equipment [3], nuclear equipment [4], GSHPs [5] and so on and so forth as heat exchangers [6]. Most researches on the flow and heat transfer characteristics in a helically coiled tube, conducted experimentally or numerically, were focused or based on uniform heating by giving a constant wall temperature or constant heat flux boundary condition. However, plenty of helically coiled tubes are applied in industrial engineering with non-uniform heating conditions. Such utilizations are commonly seen in water cooled wall in power plant boilers, the cooling pipe in fusion reactors, some particular heat exchangers for chemical reaction process and solar energy systems, as long as the heat source is in one side of the coil. Just a few studies on the non-uniformly heated helically coiled tube can be found in previous literatures. Therefore, it is necessary to investigate into flow and heat transfer characteristics in a helically coiled tube heated non-uniformly.

Secondary flow is the flow perpendicular to the mainstream direction. Although the velocity magnitude order of secondary flow is much smaller than that of the mainstream in a helically coiled tube, it can significantly affect the heat transfer rate [7]. In the cross section of a helically coiled where the flow is fully developed, the secondary flow is shown as two nearly symmetrical vortex cells, as shown in Figure 1, and the main reason for such phenomenon is the centrifugal force caused by the tube bending [8]. Whether the change of heating method has influence on the secondary flow and further on the heat transfer is the main point to study in this paper.

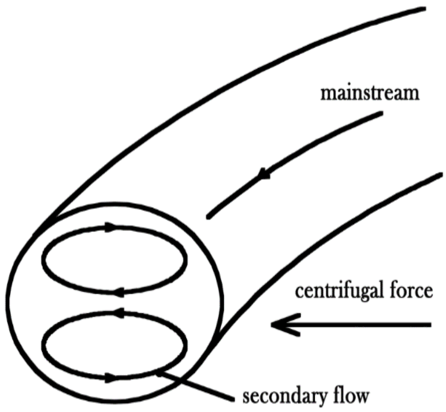


Figure 1 Secondary flow in the cross section of a helically coiled tube

Flow and heat transfer characteristics in a helically coiled tube have been studied numerically, experimentally and theoretically in a great number of literatures [9]. Dean firstly studied the secondary flow in a helical tube theoretically and presented the flow characteristics in helical tubes with a mathematic model [10]. Ferng [11] numerically studied the heat transfer characteristic variation with respect to Dean number and pitch size in a helically coiled tube. Berger et al. [12], Shah and Joshi [13] and Naphon and Wongwises [14], who reviewed the flow and heat transfer characteristics respectively, comprehensively presented most of the previous work on curved tubes. Fsadni et al. [15] reviewed the pertinent literature on frictional pressure drop reduction for laminar and turbulent flow in helically coiled tubes, which provided the summary of the relevant correlations of the frictional pressure drop with drag reducing additives in coiled tubes. Most of the researches concerned with single phase flow are based on uniform heating conditions, and just a few of them are related to non-uniform heating. Jensen and Bergles [16] studied the CHFs of the flow in a helical coil with non-uniform heating, but they did not investigate the heat transfer coefficient. Niu et al. [17] numerically studied single phase turbulent flow and multiphase flow in a one-side heating helically coiled tube and found the different heating conditions, uniform heating and one-side heating have slight influence on the secondary flow. The main variable for Nusselt number in their work was heat flux.

Numerical simulation of the flow and heat transfer in a helically coiled tube was conducted under different heating conditions, the uniform heating condition and one-side heating condition, for both laminar and turbulent flows. Secondary flow distributions, temperature profiles, average Nusselt number variation with respect to Reynolds number and local Nusselt numbers along the periphery on the wall in the fully developed section are discussed contrastively under the two different heating conditions.

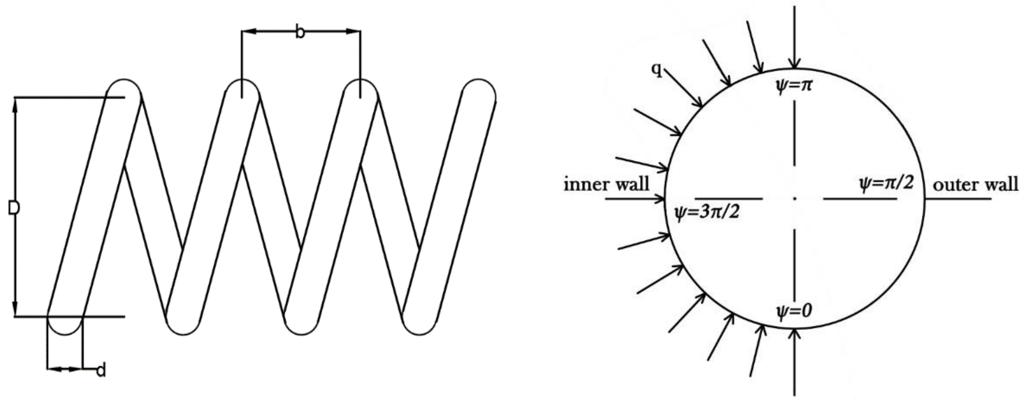
2. Methodology

2.1 Characteristics of helically coiled tubes

Figure 2 presents the geometrical parameters of a helically coiled tube, and the curvature ratio δ can be expressed by the ratio of tube diameter and coil diameter:

$$\delta = \frac{d}{D} \quad (1)$$

In this paper, d is fixed at 10mm, D is 315mm for laminar flow ($\delta = 0.032$) and 100mm for turbulent flow ($\delta = 0.05$), and b is 100mm for laminar flow while for turbulent flow, it is set as 20mm. Different tube parameters are used in the modeling process for validation with different correlations proposed in previous literatures. The non-uniform heating is simplified in this model, with uniform heating in the inner coil side as shown in Figure 2(b). It is assumed the inner wall is uniformly heated with constant heat flux q, and the outer wall is adiabatic. The heat flux q is $5kW/m^2$ and $20kW/m^2$ for laminar and turbulent flow separately. In such cases, the temperature rises in the fully developed region are less than 10K, so that the fluid properties would not change significantly.



(a) (b)

Figure 2 The helically coiled tube geometry (a) the coil geometry (b) the peripheral geometry of the tube with one-side heating

The working fluid is water with the inlet temperature 307.15K. Viscosity, density, thermal conductivity and specific capacity are estimated by the following equations [18]:

$$\mu(T) = 2.1897 \cdot 10^{-11}T^4 - 3.055 \cdot 10^{-8}T^3 + 1.6028 \cdot 10^{-5}T^2 - 0.0037524T + 0.33158$$

$$\rho(T) = -1.5629 \cdot 10^{-5}T^3 + 0.011778 \cdot 10T^2 - 3.0726T + 1227.8 \quad (3)$$

$$k(T) = 1.5362 \cdot 10^{-8}T^3 - 2.261 \cdot 10^{-5}T^2 + 0.010879T - 1.0294 \quad (4)$$

$$Cp(T) = 1.1105 \cdot 10^{-5}T^3 - 3.1078 \cdot 10^{-3}T^2 - 1.478T - 4631.9 \quad (5)$$

Experimental findings indicate that after two turns of a helically coiled tube, the flow becomes fully developed [19]. Therefore all the data obtained to calculate the Nusselt numbers or friction factors are from the cross section after 2.5th turns.

On account of the existence of secondary flow, the critical Reynolds number for a helical tube is higher than that in a straight tube. Details can be found in Jayakumar et al. [18], which summarized correlations from Ito [20], Schmidt [21], Srinivasan et al. [22], and Janssen et al. [23] with regard to the critical Reynolds number in a helical tube. All the data tested in this paper are in the Reynolds number range for both laminar and turbulent flows.

2.2 Numerical approach

The numerical simulation for studying flow and heat transfer characteristics in a helical tube is carried out using Gambit 2.4.6 and Fluent 15.0. Structured grid with 1350,000 cells in 3 loops of helically coiled tube are used in the model, where the grid number of cross section is 4500, as shown in Figure 3. Grid independence is tested for both laminar and turbulent flow: the errors of average Nusselt number of the fully developed section after refreshing a denser grid are less than 0.5% for the selected grid, while mass and energy errors do not decrease in any appreciable way. Table 1 shows the results of the grid independence study for turbulent flow, which is complemented from the considerations of axial grid and cross section grid respectively.

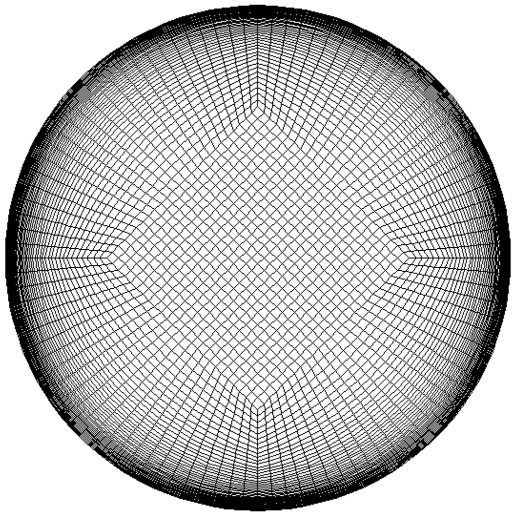


Figure 3 Grid in the cross section

Table 1 Grid independence results for $Re = 41300$

$N_{sec} = 4500$		$N_{axi} = 300$	
N_{axi}	Nu	N_{sec}	Nu
300	305.34	1500	308.48
500	305.34	3000	306.17
700	305.34	4500	305.34
900	305.35	6000	304.68

Velocity-inlet with uniform velocity and pressure-outlet of 0 Pa are used in the model as the boundary conditions. Both inner wall and outer wall are treated as no-slip boundary conditions, and the difference is that the thermal boundary condition for inner wall is constant heat flux while the outer wall is specified as adiabatic wall. In addition, intensity and hydraulic diameter are chosen as turbulence inlet boundary condition, and realizable k-ε turbulent model with enhanced wall treatment is used, which has been reported to perform well in simulating flows involving rotation [24]. SIMPLEC scheme is used for pressure-velocity coupling. Convergence criteria for continuity, momentum equations are 1e−06, and 1e-08 for energy equation.

Nusselt number is one of the most important dimensionless number for evaluation of the heat transfer characteristic in flowing fluids. Average Nusselt number is taken into consideration in a specific cross section in the fully developed section of the helically coiled tube. Friction factors are also calculated in order to validate the model with correlations from other researchers. Following are the equations for computing average Nusselt number and average friction factor.

average Nusselt number:

$$Nu_{av} = \frac{dq}{k\left(T_{w,\ \mathrm{av}} - T_{bu}\right)} \quad (6)$$

local Nusselt number:

$$Nu_{lo} = \frac{dq}{k\left(T_{w,\ lo} - T_{bu}\right)} \quad (7)$$

where T_{bu} is the fluid bulk temperature computed by the following equation:

$$T_{bu} = \frac{\int_0^A \omega T dA}{\int_0^A \omega dA} \quad (8)$$

average friction factor:

$$f_{av} = \frac{\tau_{w,\ \ av}}{\frac{1}{2}\rho V^2} \quad (9)$$

3. Results and discussion

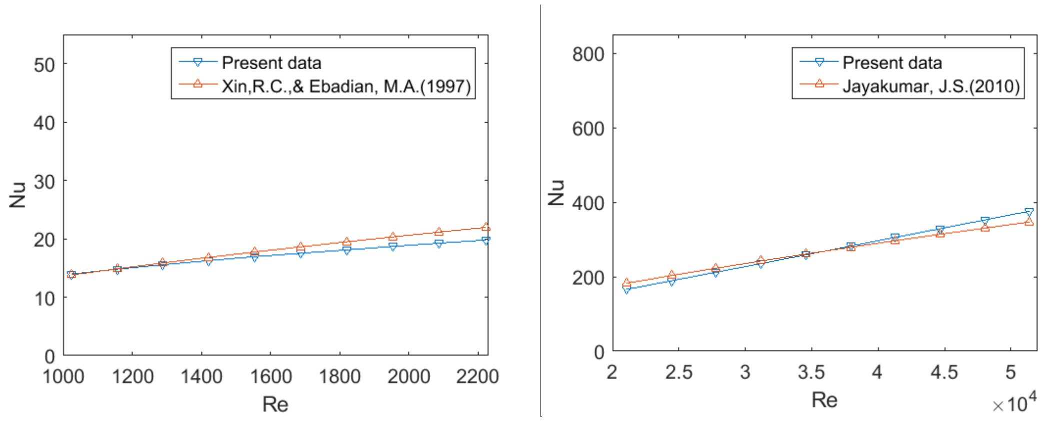
3.1 Validation of the model

In this paper, the model is validated by comparing the results under uniform heating condition with correlations from previous works. Correlations proposed by Xin and Ebadian [19], Jayakumar [18] and Ito [25] cited by Piazza [26] are used for comparison, as shown in Table 2:

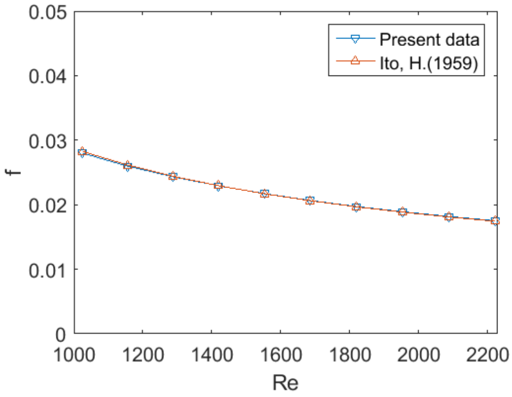
Table 2 Correlations of previous works for validation

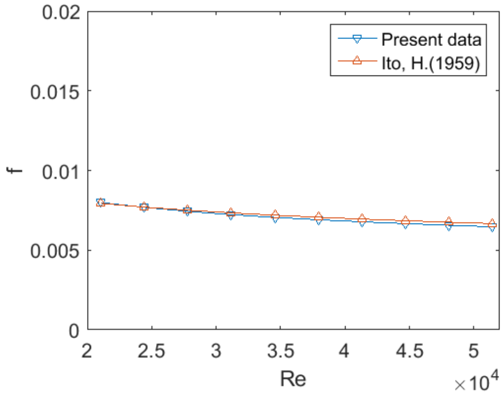
Author	Range of parameters	Correlation	Remarks
Xin and Ebadian (1997)	$20 < De < 2000$ $0.7 < Pr < 175$ $0.0267 < d/D < 0.0884$	$Nu = (2.153 + 0.318De^{0.643})Pr^{0.177}$	Nusselt number, laminar flow
Jayakumar (2010)	$14000 < Re < 70000$ $3 < Pr < 5$ $0.05 < \frac{d}{D} < 0.2$	$Nu = 0.116Re^{0.71}Pr^{0.4}\left(\frac{d}{D}\right)^{0.11}$	Nusselt number, turbulent flow
Ito (1959)	$13.5 < De < 2000$ $5 \cdot 10^{-4} < \frac{d}{D} < 0.2$	$f = \frac{64}{Re} \cdot \frac{21.5De}{(1.56 + log_{10} De)^{5.73}}$	Friction factor, laminar flow
Ito (1959)	$0.034 < \mathrm{Re}\left(\frac{d}{D}\right)^2 < 300$ $5 \cdot 10^{-4} < \frac{d}{D} < 0.2$	$f = 0.304Re^{-0.25} + 0.029\sqrt{\delta}$	Friction factor, turbulent flow

Figure 4 shows the comparison between the present data and the correlations data. From the figures it can be seen that the maximum deviations of the Nusselt numbers and friction factors between the simulation result and the predicted data from correlations are 9.80% and 3.01%, respectively, which means the simulation work is in good agreement with previous works and the numerical model can be used to study the one-side heating situation.



(a) (b)





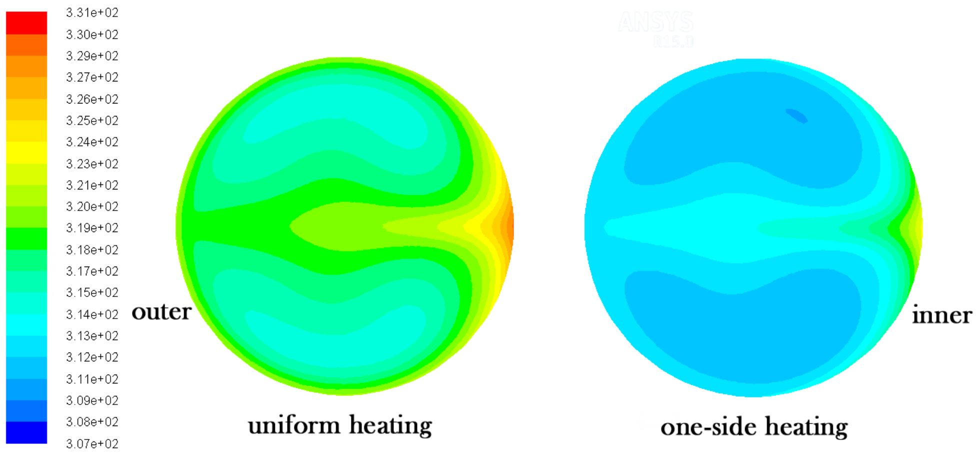
(c) (d)

Figure 4 Comparison between the present results with correlations for (a) laminar flow, Nusselt number (b) turbulent flow, Nusselt number (c) laminar flow, friction factor (d) turbulent flow, friction factor

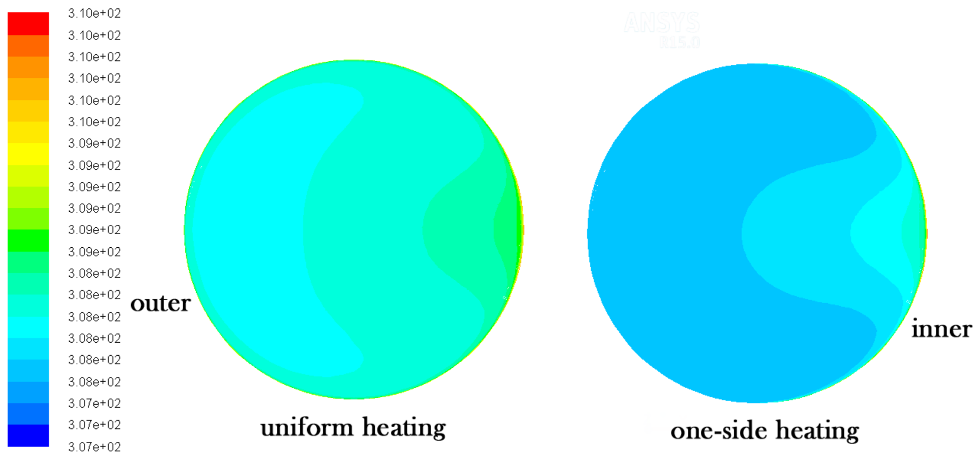
3.2 Flow and heat transfer in the helically coiled tube

3.2.1 Temperature distributions

The first property that should be considered to be affected by change of heating methods is the temperature distribution, which is directly related to heating conditions. Figure 5 shows the temperature distributions in fully developed sections for laminar flow and turbulent flow under different heating conditions. It can be seen from the figures that for both flow states, temperature profiles are similar under different heating conditions, while the differences are the temperature gradients: one-side heating causes a larger temperature gradient. Due to the existence of secondary flow, most of the heat transferred from the heating surface moves along the wall and gathers near the midpoint of the inner wall, then moves towards the interior region following the fluid flow, causing the highest temperature at the innermost area. In addition, two distinct rolling-cells can be seen for laminar flow while not for turbulent flow, which means the secondary flow affects heat transfer more for laminar flow.



(a)

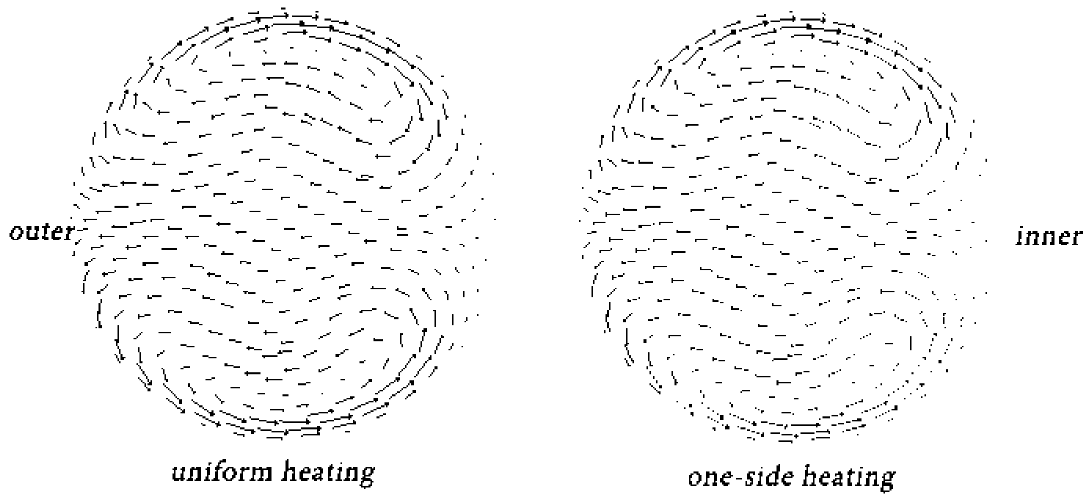


(b)

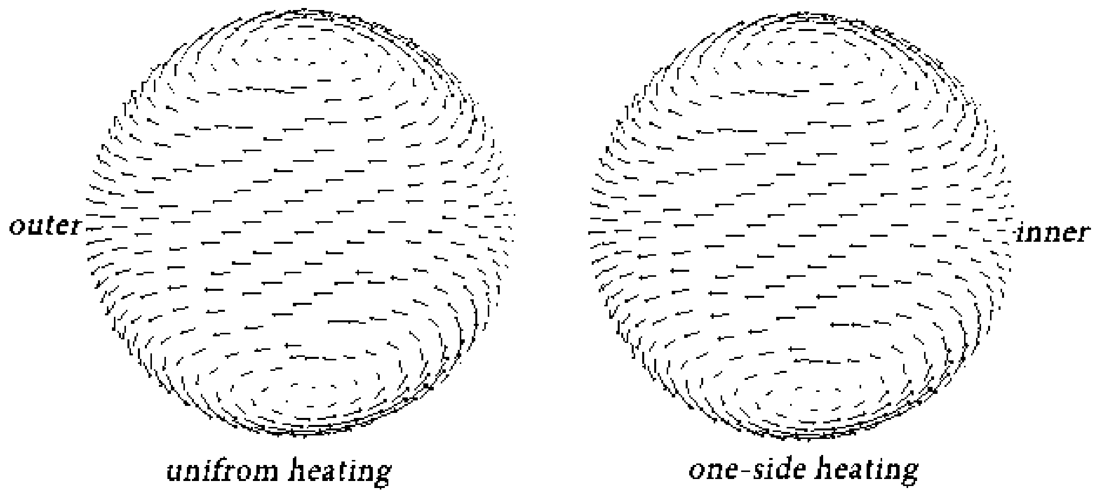
Figure 5 Comparison of temperature distributions for (a) laminar flow, Re=1954 (b) turbulent flow, Re=41300

3.2.2 Secondary flow

Compared to straight tubes, secondary flow is the most specific flow character in curved tubes playing a significant role in enhancing heat transfer. The influence of heating condition on secondary flow is investigated in this section. Figure 6 shows the secondary flow distributions in the cross section under different heating conditions for laminar flow and turbulent flow, respectively. From the figures it can be seen that no matter what the flow state is, there are no obvious differences in the secondary flow distributions when using different heating conditions, uniform heating or one-side heating. This means the water properties change induced by temperature differences between the outer side and the inner side walls are not significant enough to make a difference to the secondary flow distributions. The velocity magnitude order of secondary flow is much smaller than the main stream, and meanwhile the difference of convection flow caused by density and viscosity changes, which have an effect on the centrifugal force and the flow boundary layer, is even much smaller than the secondary flow.



(a)



(b)

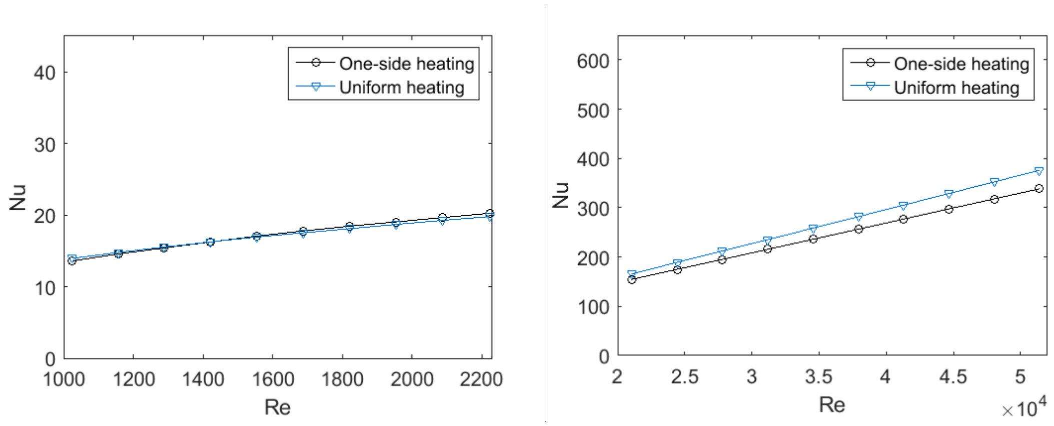
Figure 6 Comparison of secondary flow distributions for (a) laminar flow, Re=1954

(b) turbulent flow, Re=41300

3.2.3 Average Nusselt number variation with respect to Reynolds number

The average Nusselt number at the fully developed section is discussed in this section. Figure 7 shows the average Nusselt numbers variation with respect to Reynolds number for both laminar and turbulent flows. Although the main factor influencing heat transfer characteristic, the secondary flow distributions, are similar under different heating conditions, average Nusselt numbers are different. As the secondary flow distributions are quite close, the heat transfer characteristic should also be close. The main reason for the difference of Nusselt numbers is that under different heating conditions, the definitions of Nusselt number are not on the same standard; in another word, the water bulk temperature should not be used as the same reference temperature for comparison. However, there is no such a standard reference temperature that can be used for both two heating conditions, so the Nusselt numbers from different heating conditions are not comparable. Therefore new correlations should be proposed for one-side heating condition. From Figure 8(a) it can be seen the average Nusselt numbers under one-side heating are close to that under uniform heating for laminar flow, thus the correlations predicted in previous works to calculate Nusselt numbers for uniform heating can be used in one-side heating cases for laminar flow. However, with regard to turbulent flow, as shown in Figure 7(b), the difference of Nusselt numbers between the two heating conditions are larger, and a new correction to calculate Nusselt numbers for turbulent flow under one-side heating is proposed in this paper. The proposed correlation matches well with the present data, with the maximum deviation of 1.16%, as shown in Figure 8.

$$Nu = 0.0163Re^{0.8875}Pr^{0.4}\left(\frac{d}{D}\right)^{0.11} \quad 21061 < Re < 51406, 4.75 < Pr < 4.98, \frac{d}{D} = 0.05$$



(a) (b)

Figure 7 Comparison of average Nusselt numbers for (a) laminar flow (b) turbulent flow

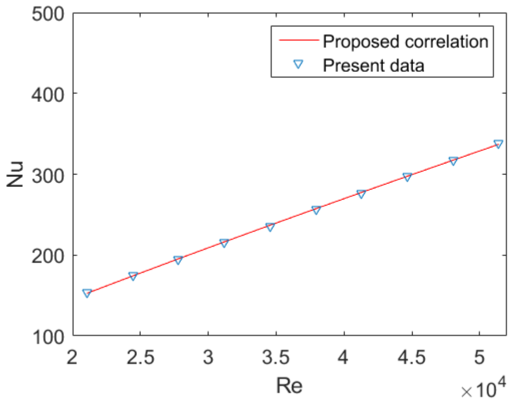
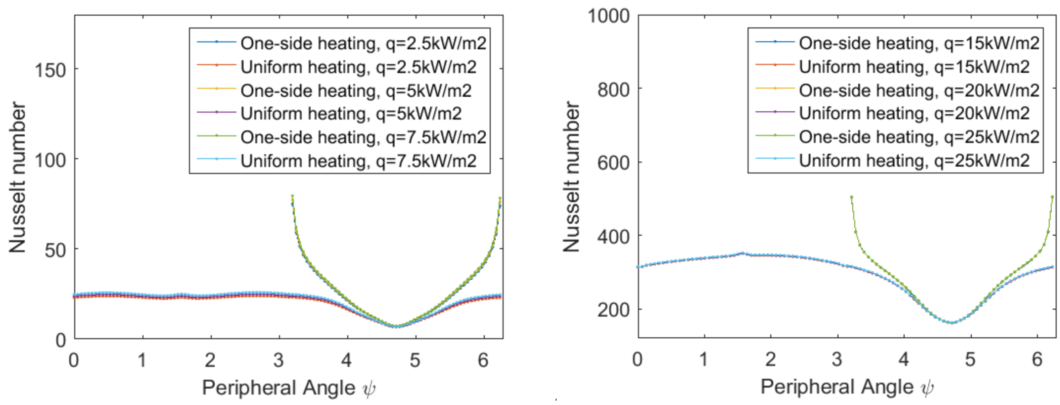


Figure 8 Proposed correlation for turbulent flow with one-side heating

3.2.4 Comparison of local Nusselt numbers

The local Nusselt numbers are studied as well. Figure 7 shows the comparison of local Nusselt numbers along the periphery of the fully developed cross section calculated by equation 5. Both laminar and turbulent flows are simulated for uniform and one-side heating conditions with three groups of heat fluxes. From the figures it can be seen that for both laminar and turbulent flow states, the local Nusselt numbers on the inner wall are higher when conducting one-side heating, while the difference decreases as it is closer to the midpoint of the inner wall. Interestingly, at the midpoint where $\psi = 90°$, the local Nusselt number curves are almost coincident, as shown in the figures. In addition, heating flux variation has little influence on Nusselt numbers. Therefore formula (9) can be easily concluded from the Nusselt equation. This should be very useful in engineering applications since all the temperatures in this formula can be easily measured, thus if one of the heat fluxes is unknown it can be estimated by the corresponding heat flux.

$$\frac{q_{uni}}{k_{uni}\left(T_{w,uni} - T_{bo,uni}\right)} \approx \frac{q_{one}}{k_{one}\left(T_{w,one} - T_{bo,one}\right)} \qquad (11)$$



(a) (b)

Figure 9 Comparison of local Nusselt numbers for (a) laminar flow, Re=1954 (b) turbulent flow, Re=41300

To explain this phenomenon, one specific case can be considered to help understand. Assuming that $q_{uni} = 2q_{one}$ and the flow states are the same for both heating conditions, the bulk fluid temperatures at the same fully developed cross section should be almost the same, namely $T_{bo,uni} = T_{bo,one}$, because the heating area of one-side heating is half of that for uniform heating and the total heat transferred to the bulk fluid does not change. According to the temperature and secondary flow distributions in figure 5 and figure 6, most of the heat obtained from the wall transfers along the wall from the outer side wall to the inner side wall and then to the interior of the bulk fluid after gathering at the vicinity of midpoint of the inner wall, where the temperature is the highest. For one-side heating, the route for heat convection from the heating area to the vicinity of the midpoint of the inner wall is a half of that for uniform heating, which can also be construed as the heat transfer efficiency to the innermost region for one-side heating is twice as much as that for uniform heating. Thus the temperature rise $T_{w,uni} - T_{bo,uni}$ should be approximately a half of $T_{w,one} - T_{bo,one}$ at the same cross section, then the formula can be achieved for this case. Moreover, as Nusselt numbers are hardly affected by changing heating flux as shown in Figure 9, the validity of the formula can be extended to cases that $q_{uni} \neq 2q_{one}$.

4. Conclusions

In this paper, flow and heat transfer characteristics in a helically coiled tube under one-side heating condition are investigated numerically, using water as the working fluid. Both laminar flow ($1025 < Re < 2222$) and turbulent flow ($21061 < Re < 51406$) are studied. The numerical model is validated by comparing the uniform heating condition with previous works, and the present data is in good agreement with the existing correlations. The results of simulation for one-side heated helically coiled tube are contrastively studied with that under uniform heating condition. Conclusions can be drawn as follows:

1. Regardless of the flow states, laminar flow or turbulent flow, the secondary flow distributions are hardly affected by changing the heating condition; while the temperature distributions are quite different: a larger temperature gradient can be found for one-side heating.
2. The average Nusselt numbers are close for laminar flow under different heating conditions, while for turbulent flow, it shows 7%-10% smaller Nusselt numbers for one-side heating than uniform heating. A new correlation for calculating average Nusselt numbers for turbulent flow under one-side heating condition is proposed in this work.
3. For both laminar and turbulent flows, the midpoint of the inner wall shows an interesting phenomenon for the local Nusselt number calculation. At this point of the fully developed section, the local Nusselt numbers are almost the same when using different heating flux or different heating conditions. This characteristic can be applied to calculate the unknown heat flux for one heating condition with the other known one

for the corresponding heating condition.

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