Numerical Study of Flow Structure and Heat Transfer in Straight, Helical and Spiral Tubes Under Constant Heat Flux and Constant Temperature

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Abstract
In this paper, the structure of flow and convection heat transfer in straight, helical and spiral tube heat exchangers with a circular cross section under constant heat flux and constant temperature have been studied and compared. The simulation was performed using the finite volume method with the help of Fluent software. The numerical study performed in this paper is in a turbulent flow regime with water fluid and the Reynolds range is 37917-70,000 under constant heat flux and constant temperature. The results showed that the Fanning friction coefficient and pressure drop do not depend on the boundary condition on the wall and for heat transfer coefficient and Nusselt number, there is a difference of less than 1% these two boundary conditions. In the case of the fluid mean temperature at constant heat flux, The straight tube has fluid mean temperature and consequently a higher heat transfer rate than the helical and spiral tubes. However, the fluid mean temperature and heat transfer rate are higher for the helical and spiral tubes at the constant wall temperature. For the constant heat flux boundary condition, the fluid mean temperature and the total heat transfer rate in a straight tube are 1.25% and 0.75% higher than their values in helical and spiral tubes, respectively. For the constant temperature boundary condition, the fluid mean temperature and the total heat transfer rate in the helical and spiral tubes are 5.02% and 5.64% higher than their values in straight tube, respectively. Regarding the overall performance (FoM), The results show that in both condition, the straight tube has a higher overall performance (FoM) than the helical and spiral tubes. The results showed that for the constant heat flux mode, the overall performance in straight tube is 23.66% and 24.37% higher than helical and spiral tubes, respectively. Also in the constant temperature boundary condition, the increase in overall performance in straight tube compare to helical and spiral tubes is 18.67% and 19.5% , respectively.
Keywords: Structure of flow, convection heat transfer , constant heat flux, constant temperature, fluent

1- Introduction

Processes under constant heat flux and constant temperature are the most important topics of heat transfer and have many applications in various sectors of the industry. One of the most widely used of these two boundary conditions is in heat exchangers. Nazari et al., 2013 [1] conducted experimental studies using nanofluids and pure water despite the constant heat flux on the outer wall of the tube. The results performed on pure water are compared with the relations predicted by Genlinsky and Detus-Bolter for turbulent flow, which shows and expresses given that there is good agreement between the experimental results and the Genlinsky relation, which emphasizes the accuracy and reliability of the results, the Genlinsky relation predicts Nusselt values better than the Dietos-Bolter equation. Eskandari et al., 2011 [2] investigated and compared analytical solution and numerical solution of flow and heat transfer inside the tube under constant heat flux and constant wall temperature. The results showed that the Nusselt changes and the coefficient of friction for solving the analysis and numerical solution were very close to each other. Fadaei Tehrani et al., 2016 [3] studied the forced heat transfer coefficient of laminar flow of water-copper nanofluid in a parallel channel (between two parallel flat plates) with discrete heat sources of constant temperature on the walls, numerically, and by numerically solving the governing equations. He studied the effect of Reynolds number parameters, nanoparticle volume fraction and number of heat sources on flow and temperature fields and heat transfer rate, the most important of which are to divide a heat source into smaller heat sources and distribute them on the wall. The channel noted that it causes the formation of contiguous border layers. These successive boundary layers increase the temperature gradient over the heat sources and hence the heat transfer rate increases. Kahani et al., 2013 [4] studied the effect of curvature ratio and helical step length and concluded that with increasing step length and decreasing curvature ratio, the heat transfer rate increases and also observed that the curvature ratio has a greater effect on the number compared to the step length. Has Nusselt and pressure drop. Sayadi and Ghiasi Tabari, 1396 [5] have studied the effect of helix diameter on thermal and hydraulic parameters of fluid flow inside the heat exchanger. They concluded that the smaller the helix diameter and the larger the number of helix turns instead, the greater the heat transfer and the space occupied by the heat exchanger is economically viable. Kurnia et al., 2011 [6] studied heat transfer, pressure drop, and overall performance of straight, helical, spiral and conical tubes with a constant flux boundary condition and constant temperature with a two-phase water-air fluid. When a constant heat flux is applied to the wall, the best overall performance (FoM) resulting from heat transfer over pressure drop is for a straight tube. Kurnia and Sasmito, 2018 [7] examined straight and helical tubes with different cross sections despite constant heat flux and found that the best overall performance (FoM) is the result of heat transfer over pressure drop for a straight tube. Sasmito et al, 2018 [8] investigated straight and spiral tubes with different cross sections with two-phase water-air fluid under constant heat flux and constant temperature and it was found that under constant heat flux conditions heat transfer in straight tube, it is more than spiral tube and also in terms of hydraulic and overall performance, it has better performance than spiral tube, that Between these different cross sections, the rectangular surface has more heat transfer. Abdi et al, 2019 [9] compared straight, helical and spiral tubes under constant temperature, and their results showed that helical tube with rectangular cross section has the highest heat transfer coefficient compared to other tubes. In this paper, the numerical study of flow structure and heat transfer in straight, helical and spiral tubes under constant heat flux and constant temperature is studied to determine the relationship between the flow and thermal parameters of the internal flow and the boundary condition of the wall.

2- Numerical Model

2-1- Tube geometry

In this research, the fluid used is water, which is considered as an incompressible fluid that flows in straight, helical and spiral tubes. The cross section used in this research is circular. For all geometries, the condition of constant heat flux on the wall (250 kW / m2) and constant wall temperature (373.15 K) are used, the length of all geometries is a certain value (1.2 m). Figure 1 shows the geometries in this study and the geometric characteristics of the tubes are presented in Table 1.

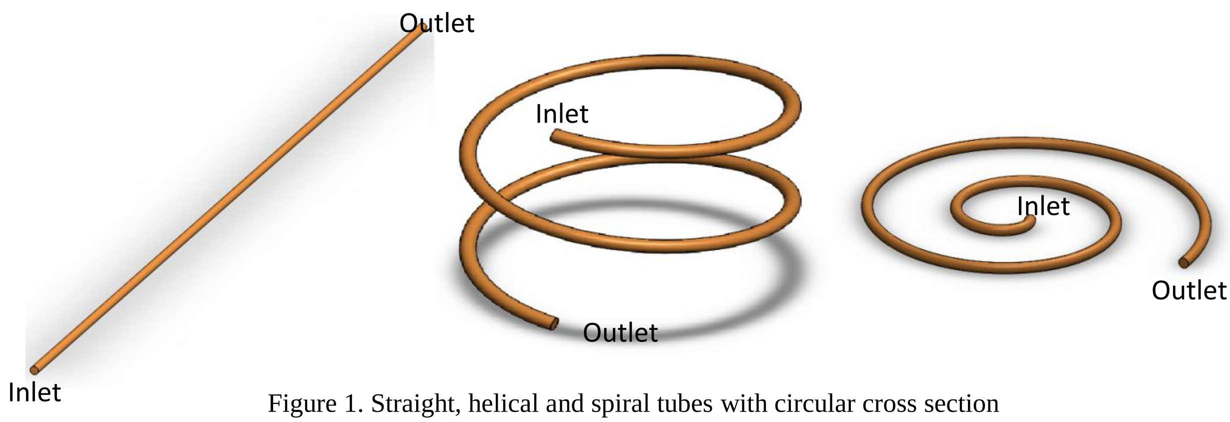

Figure 1. Straight, helical and spiral tubes with circular cross section

Table 1. Geometric characteristics of tubes

Parameters	Value	Unit
Diameter of circular tube	9.525×10^{-3}	m
Total tube length	1.2	m
Diameter of helical	1.8928×10^{-1}	m
Pitch of helical	0.08	m
Large diameter of spiral	0.35	m
Small diameter of spiral	0.033	m
Pitch of spiral	0.08	m

2-2- Governing equations

$$\frac{\partial y}{\partial x}+\frac{\partial y}{\partial x}+\frac{\partial y}{\partial x}=0 \tag{1}$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+v\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+v\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \tag{3}$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+v\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{4}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{5}$$

The Standard k-ε turbulence model is used to simulate the turbulent flow in straight tube and the Relizable k-ε turbulence model is used to simulate the turbulent flow in helical and spiral tubes. The Relizable k-ε model is more accurate than the two standard and RNG models for predicting currents, including rotation, complex secondary current, extreme curvature of flow lines, and separation [10]. Pressure and velocity coupling was performed with SIMPLE algorithm for straight tube and with coupled coupling to reduce analysis time for helical and spiral tubes and the Second order upwind scheme was used to solve momentum equations, turbulent kinetic energy, turbulent dissipation rate and energy equation.

In this research, the fluid used is water, which is considered incompressible with constant thermo-physical properties, and also the tubes is made of copper, which is presented in Table 2 of the thermo-physical properties of water and copper used.

Table 2. Thermo-physical properties of water and copper

	$\rho(kg/m^3)$	$k(W/mK)$	$C_p(J/kgK)$
Water	998.2	0.6	4182
Copper	8978	387.6	381

The heat transfer coefficient and the Nusselt number are obtained from the following equations.

$$h=\frac{Q_{wall}}{(T_w-T_{mean})} \tag{6}$$

$$Nu=\frac{hD}{k} \tag{7}$$

In the above relations, Q_{wall} is the wall heat flux, T_{wall} is the wall temperature and T_{mean} is the average temperature of the fluid.

The average temperature of the fluid is obtained from the following equation.

$$T_{mean}=\frac{1}{VA_c}\int_{A_c}TudA_c \tag{8}$$

In the above relation, A_c is the cross-sectional area of the tube and V is the average velocity, which is defined as follows.

$$V=\frac{1}{A_c}\int_{A_c}udA_c \tag{9}$$

The total heat transfer rate and the overall performance (FoM), which is the ratio of heat transfer to pressure drop, are calculated according to the following equations.

$$\dot{Q}_{total}=\dot{m}C_p\left(T_{mean,L}-T_{mean,i}\right) \tag{10}$$

$$F_{merit}=\frac{\dot{Q}_{total}}{\Delta P} \tag{11}$$

In the above equations $T_{mean,\ i}$ is the mean fluid temperature at the inlet of the tube and $T_{mean,\ L}$ is the mean fluid temperature throughout the tube.

2-3- Boundary conditions

At the inlet of the tube, the boundary condition is the inlet velocity at which the liquid enters the tube at the certain temperature.

$$Inlet: u = u_{in}, T = T_{in} \tag{12}$$

At the end of the tube, an outlet pressure boundary condition is considered in which the relative pressure of the fluid is zero and the fluid exits the pipe with a zero temperature gradient.

$$Outlet: P = P_{out}, n.(k\nabla T) = 0 \tag{13}$$

In the wall, the boundary condition of the wall is considered so that there is no slipping and a constant heat flux is applied on the wall.

$$Wall: u = 0, n.(k\Delta T) = Q_{wall} \ or\ T = T_{wall} \tag{14}$$

In the above relations n is normal vectors, u_{in} is inlet velocity, P_{out} is outlet pressure, Q_{wall} is wall heat flux and T_{wall} is wall temperature.

Tables 3 and 4 shows the values related to the boundary conditions.

Table 3. Boundary condition values for constant heat flux condition

Parameters	Value	Unit
Inlet temperature	300	K
Wall heat flux	25×10^4	W/m^2
Outlet pressure (gauge)	0	Pa

Table 4. Boundary condition values for constant temperature condition

Parameters	Value	Unit
Inlet temperature	300	K
Wall temperature	373.15	K
Outlet pressure (gauge)	0	Pa

3- Results

3-1- Fanning friction coefficient

As the Reynolds of internal flow increases, the velocity gradient near the wall increases and therefore the shear stress increases. Also, with increasing Reynolds due to the reduction of the friction force, the friction coefficient of internal flow decreases. In addition, the presence of centrifugal force, which occurs in helical and spiral tubes, increases the shear stress. The results were obtained by showing that the fanning friction coefficient of internal flow diagram was the same for both boundary conditions and it was found that the fanning friction coefficient of internal flow in the tube is independent of the wall boundary condition. In both diagrams, it was found that the fanning friction coefficient of the internal flow in the helical tube is 14.94% and in the spiral tube is 14.38% higher than that of the straight tube.

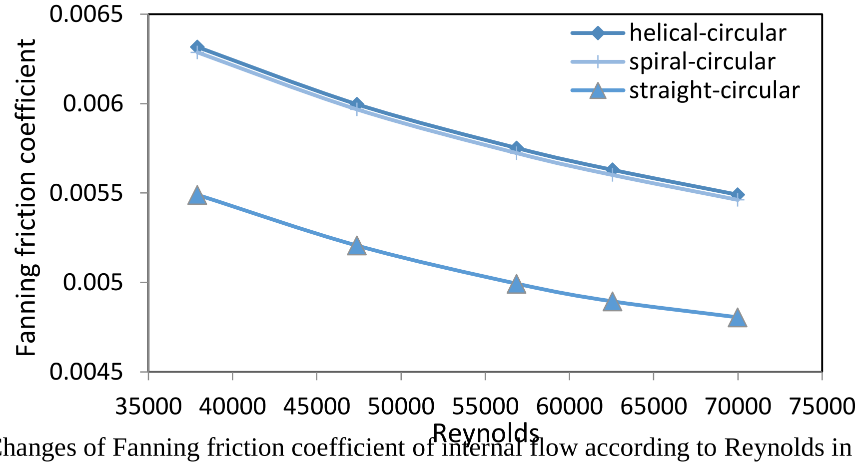

Figure 2. Changes of Fanning friction coefficient of internal flow according to Reynolds in straight, helical and spiral tubes with constant flux condition

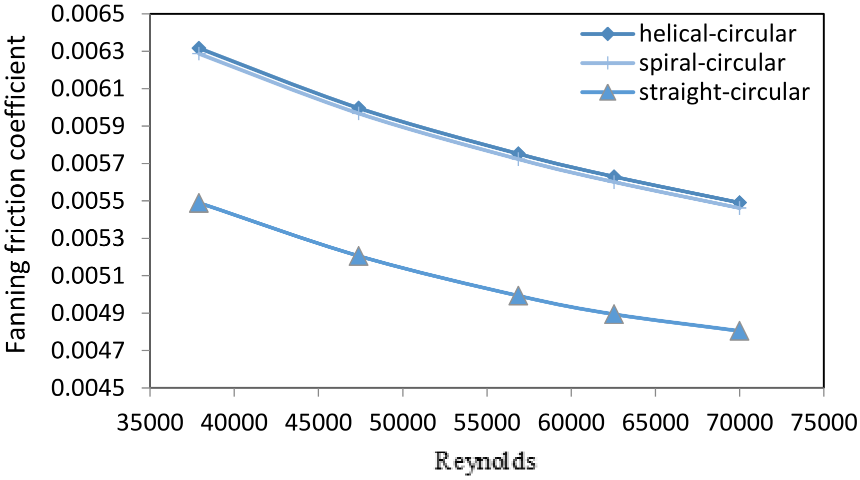

Figure 3. Changes of Fanning friction coefficient of tubes with different Reynolds with constant temperature condition

3-2- pressure drop

The concept of turbulent flow in tube line means that the fluid particles move in any direction while the net flow of the fluid is to reduce the pressure. So that all the fluid is not in the direction of absorbing the net energy of the tube fluid. As the flow rate increases, the energy absorbed by the cross currents perpendicular to the flow axis increases and the pressure drop increases. Also, the pressure drop of internal flow is greater in helical and spiral tubes, that have a secondary flow due to centrifugal force.

The results are obtained that the pressure drop diagram for both boundary conditions, which shows that the flow pressure drop inside the tube is independent of the boundary condition of the wall. In both diagrams of pressure drop of internal flow, it was found that the pressure drop of internal flow in helical tube is 29.35% and spiral tube 31.24% more than straight tube. Comparing helical and spiral tubes in the same step conditions, uniform curvature ratio and same number of turns, the pressure drop of internal flow in the spiral tube has a 1.89% higher pressure drop than the helical tube and in fact has the worst hydraulic performance.

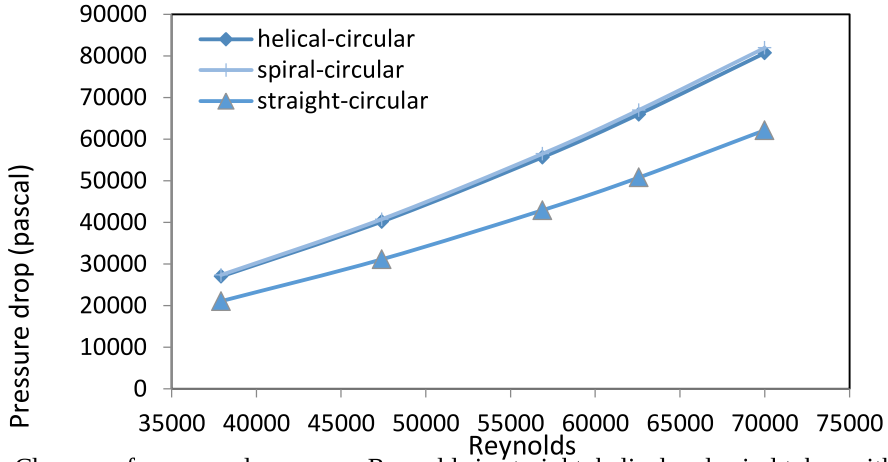

Figure 4. Changes of pressure drop versus Reynolds in straight, helical and spiral tubes with constant heat flux condition

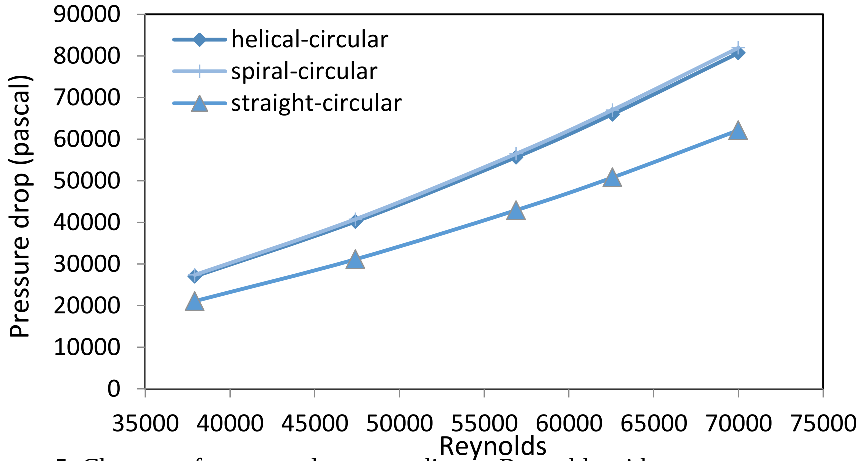

Figure 5. Changes of pressure drop according to Reynolds with constant temperature condition

3-3- Average heat transfer coefficient

As the Reynolds of internal flow increases, the temperature gradient near the wall, which is the heat source, increases and the average heat transfer coefficient increases. In helical and spiral tubes, due to the presence of secondary flow, the heat transfer coefficient of internal flow increases. The results showed that the diagram of the average heat transfer coefficient for each boundary condition of constant heat flux and constant temperature is slightly different from each other, which indicates that the heat transfer coefficient is slightly independent of the boundary condition of the wall. In the heat transfer coefficient diagram for constant flux boundary condition, it was found that the average heat transfer coefficient of internal flow in helical tube is 7.69% and spiral 6.82% higher than straight tube and this diagram for constant temperature boundary condition indicates that the average heat transfer coefficient of internal flow in helical tube is 8.03% and spiral 7.64% more than straight tube.

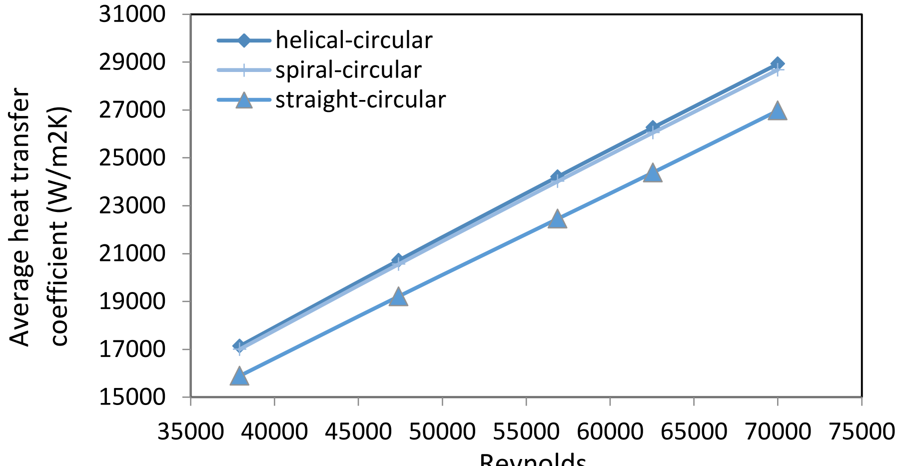

Figure 6. Changes of average heat transfer coefficient of internal flow with different Reynolds in straight, helical and spiral tubes with constant heat flux condition

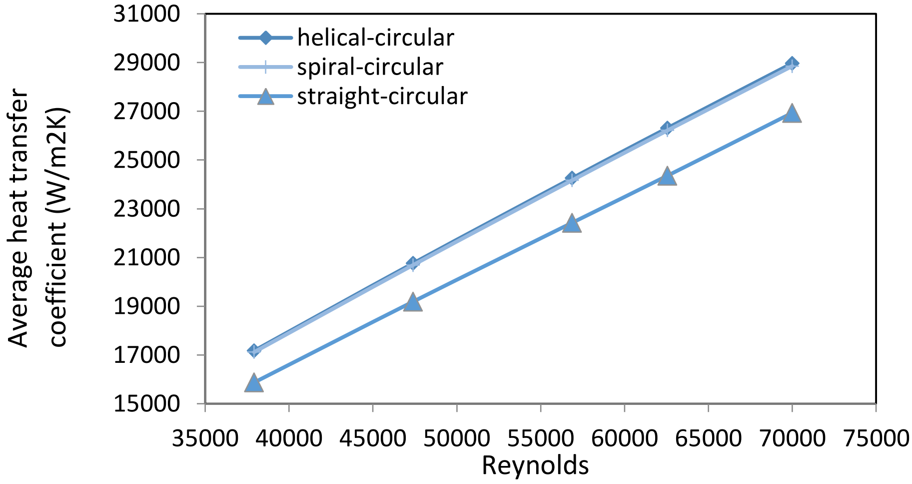

Figure 7. Changes of average heat transfer coefficient of internal flow versus Reynolds with constant temperature condition

3-4- Average Nusselt number

As discussed, the heat transfer coefficient diagram in constant flux mode is slightly different from the heat transfer coefficient diagram in constant temperature mode and is slightly the same, and it can be said that the heat transfer coefficient is almost independent of the wall boundary condition and since the tube cross section The results are the same, so it is concluded that the Nusselt number diagram in constant flux mode is slightly different from the displacement coefficient diagram in constant temperature mode and are almost the same, and it can be said that the Nusselt number is almost independent of the wall boundary condition. In the diagram of the average Nusselt number for the constant flux boundary condition, it was found that the average Nusselt number of internal flow in helical and tubes is 7.69% and spiral is 6.82% more than in the straight tube. Average Nusselt number of internal flow in helical tube is 8.03% and spiral 7.64% more than straight tube.

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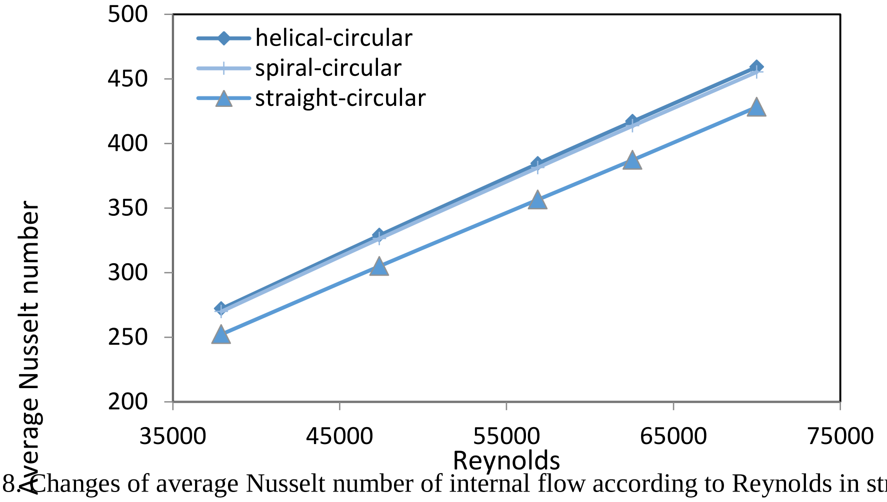

Figure 8. Changes of average Nusselt number of internal flow according to Reynolds in straight, helical and spiral tubes with constant heat flux condition

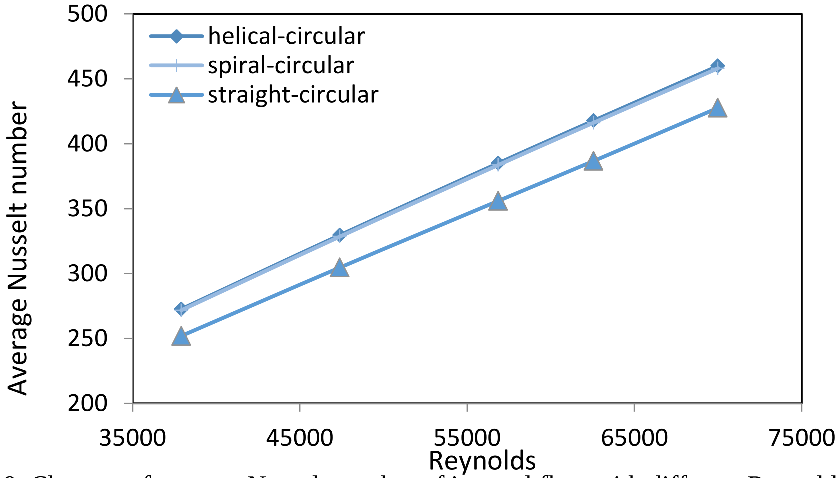

Figure 9. Changes of average Nusselt number of internal flow with different Reynolds with constant temperature condition

3-5- Fluid mean temperature

Numerical studies performed in this study show that the boundary condition of the wall has a great effect on this parameter. When the constant heat flux boundary condition on the wall is used, since the lateral surface area is the same for the tubes, then the straight tube with the lowest heat transfer coefficient of internal flow has the highest fluid mean temperature and in the constant temperature boundary condition The presence of a secondary flow increases the fluid mean temperature, so in the constant temperature boundary condition the constant fluid mean temperature in the helical and spiral tubes is mostly straight tube. Comparing helical and spiral tubes under the same step conditions, the same curvature ratio and the same number of revolutions, the fluid mean temperature in spiral tubes have a higher fluid mean temperature than the helical tubes for both the constant heat flux boundary condition and the constant temperature boundary condition.

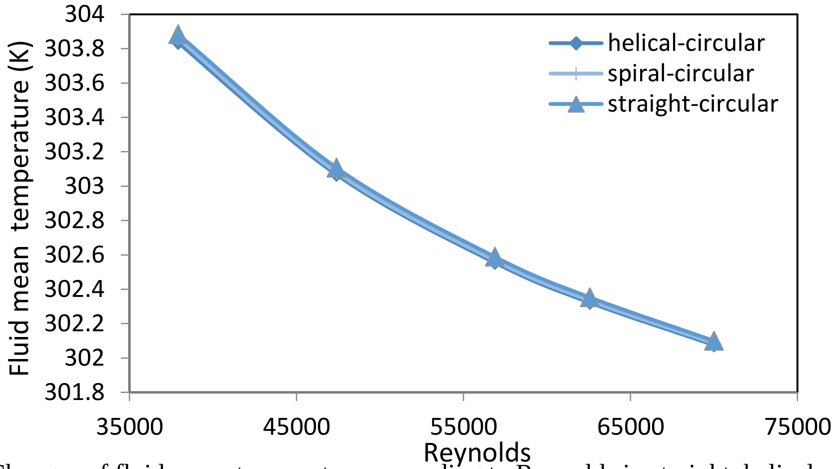

Figure 10. Changes of fluid mean temperature according to Reynolds in straight, helical and spiral tubes with constant heat flux condition

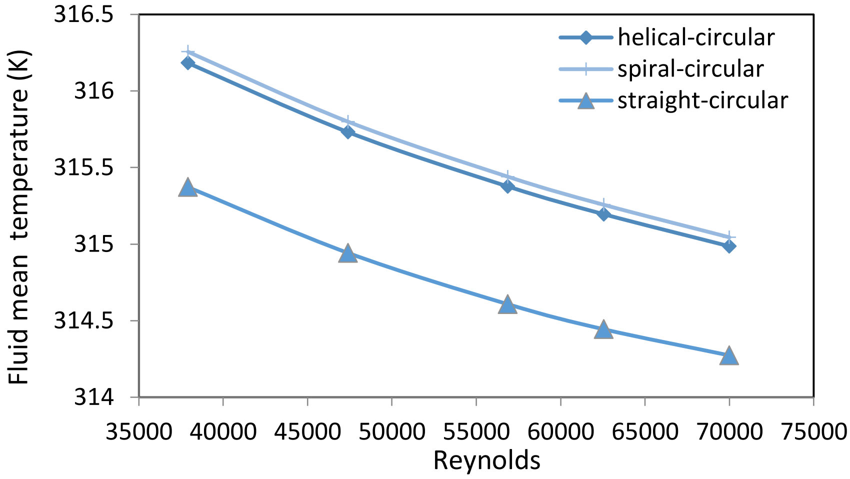

Figure 11. Changes of fluid mean temperature versus Reynolds with constant temperature condition

3-6- Total heat transfer rate

Since the tubes have the same length and cross-section, the mass flow rates of the fluid in the geometries are equal in each other. Therefore, the total heat transfer rate in this case depends only on the fluid mean temperature. Therefore, in constant heat flux condition, the total heat transfer rate of straight tube is 1.25% and 0.74% higher than helical and spiral tubes, respectively, and due to constant heat flux and the same lateral surface area, the total heat transfer rate of tubes in constant flux for different Reynolds is almost constant. At constant temperature condition, the total heat transfer rate of helical and spiral tubes is 5.2% higher than that of straight tube. The total heat transfer rate increases with increasing Reynolds ratio. The same number of revolutions for both the constant heat flux boundary condition and the constant temperature boundary condition of the spiral tube has a higher overall heat transfer rate than the helical tube.

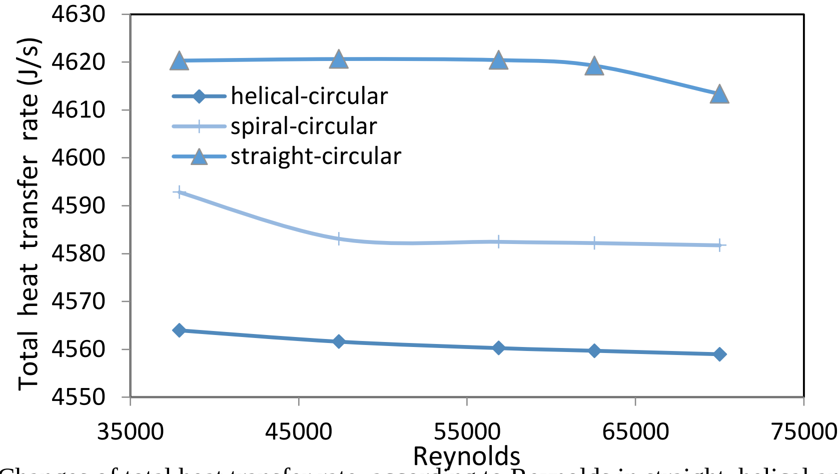

Figure 12. Changes of total heat transfer rate according to Reynolds in straight, helical and spiral tubes with constant heat flux condition

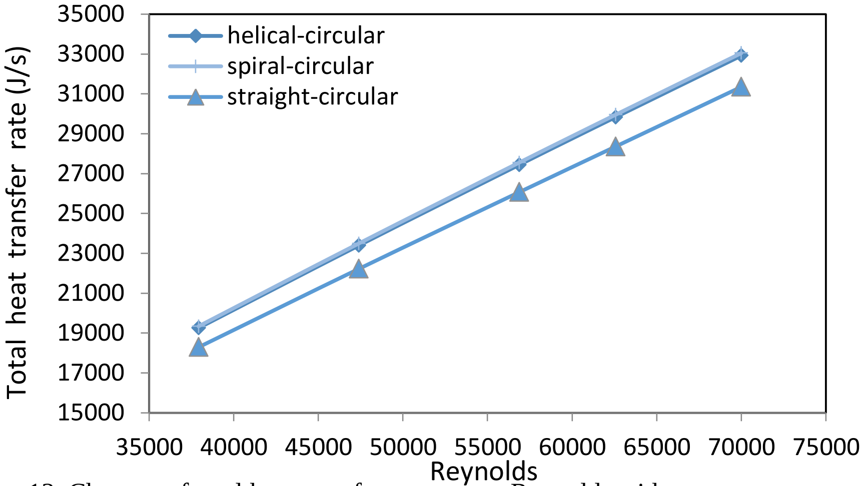

Figure 13. Changes of total heat transfer rate versus Reynolds with constant temperature condition

3-7- Overall performance

Numerical studies show that the overall performance of the internal flow, which is the ratio of heat transfer to pressure drop, is higher for both the constant heat flux boundary condition and the constant temperature boundary condition in a straight tube than in helical and spiral tubes. For the constant heat flux condition, the overall performance of internal flow in the straight tube is 23.66% higher than that of the helical tube and 24.37% higher than that of the spiral tube. This is because it can be said that the pressure drop has the greatest effect on the overall performance and comparing the helical and spiral tubes in the same step conditions, the same curvature ratio and the same number of revolutions for both the constant heat flux boundary condition and the temperature boundary condition, the overall performance of internal flow in the helical tube has a greater overall performance than the spiral tube.

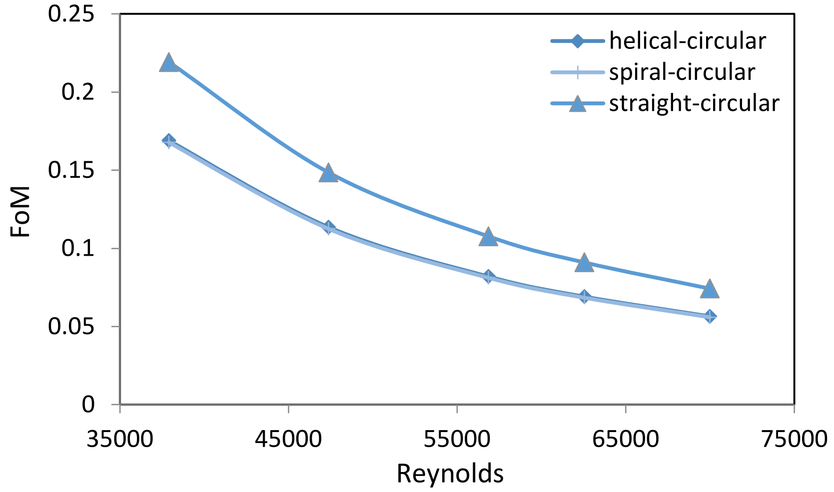

Figure 14. Changes of overall performance according to Reynolds in straight, helical and spiral tubes with constant heat flux condition

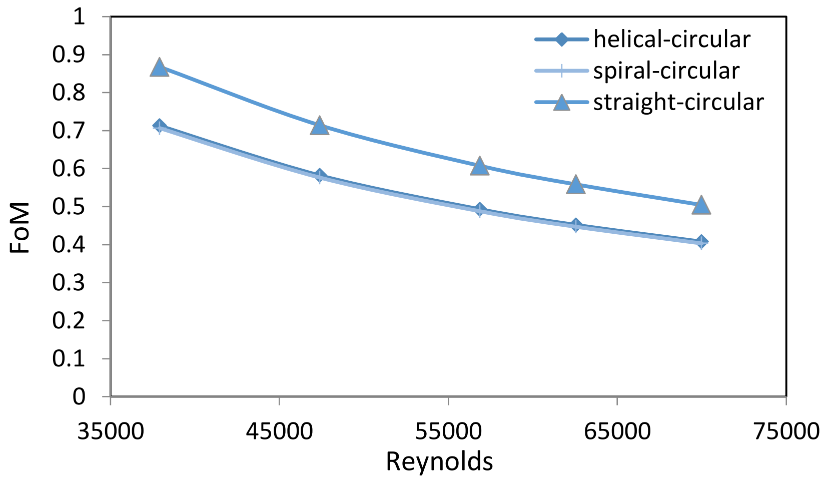

Figure 15. Changes of overall performance with different Reynolds with constant temperature condition

4- Results and discussion

In this study, various flow and thermal parameters including Fanning friction coefficient, pressure drop, average heat transfer coefficient, average Nusselt number, fluid mean temperature, total heat transfer rate and overall performance of internal flow in straight, helical and spiral tubes with the circular cross section were investigated with two different boundary conditions, which are constant heat flux on the wall and constant wall temperature. Based on the results, it is concluded:

1- Fanning coefficient and pressure drop of the flow internal flow are independent of the boundary condition of the tube wall, which for helical and spiral tubes due to the existence of secondary flow are higher than the straight tube.
The fanning friction coefficient of internal flow in helical and spiral tubes is 14.94 and 14.36% higher than its value in straight tube, respectively.
Pressure drop of internal flow in helical and spiral tubes is 29.35 and 31.24% higher than its value in straight tube, respectively.

2- Difference between the average heat transfer coefficient and the average Nusselt number of internal flow in constant heat flux and constant temperature conditions are 0.13% in straight tube, 0.19% in helical tube and 0.64% in spiral tube, which concludes with an approximation of less than one percent independent of the boundary condition of the wall.

It was also observed that the average heat transfer coefficient and the average Nusselt number of internal flow in helical and spiral tubes are higher than straight tube.
For the boundary condition of constant heat flux, the average heat transfer coefficient and the average Nusselt number of internal flow in helical and spiral tubes are 7.69% and 6.82% higher than their values in straight tube, respectively.
For the constant temperature boundary condition, the average heat transfer coefficient and the mean Nusselt number of internal flow in helical and spiral tubes are 8.03% and 7.64% higher than their values in straight tube, respectively.

3- The fluid mean temperature and the total heat transfer rate for the boundary condition of constant flux and constant temperature are different in terms of totality. When the boundary condition of the wall is constant heat flux, from there a constant heat is transferred to the tube. The presence of a secondary flow increases the heat transfer coefficient of the internal flow and the fluid mean temperature and consequently the total heat transfer rate decreases. However, when the boundary condition of the wall is constant temperature, the presence of a secondary flow increases the fluid mean temperature and the total heat transfer rate.
For the constant heat flux boundary condition, the fluid mean temperature and the total heat transfer rate in a straight tube are 1.25% and 0.75% higher than their values in helical and spiral tubes, respectively.
For the constant temperature boundary condition, the fluid mean temperature and the total heat transfer rate in helical and spiral tubes are 5.02% and 5.64% higher than their values in the straight tube, respectively.

4- The overall performance of the internal flow in the tubes for both the constant flux boundary condition and the constant temperature boundary condition was higher in the straight tube than the others, since the lowest pressure drop is for the internal flow in the straight tube and also due to The pressure drop is very important in the overall performance, so the overall performance of the internal flow in a straight tube is greater than in a helical and spiral tubes.
For the constant heat flux boundary condition, the overall performance of internal flow in a straight tube is 23.66% and 24.37% higher than its value in helical and spiral tubes, respectively.
For the constant temperature boundary condition, the overall performance of internal flow in a straight tube is 18.67% and 19.5% higher than its value in helical and spiral tubes, respectively.

5. In comparison, helical and spiral tubes differ in all flow and thermal parameters by less than two percent, which indicates that they have the same results with an approximation of less than two percent, but in terms of occupied space, helical tubes have less space. Occupies relative to the spiral tube and is a better choice.
It should also be noted that these results are completely consistent with the conclusions in various related articles. Since it was found that only one type of tube is not suitable for all purposes, so it is recommended to first examine the boundary condition of the system and the space in space, then select the appropriate tube, it should be noted that for the constant temperature boundary condition is better the heat transfer coefficient increase to increase the heat transfer. To increase the heat transfer coefficient, it is recommended to add a bead to the tube, use nanofluid and use a turbocharger that increases the secondary flow in the tube.

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