Research on the thermal performance of a heat exchanger with meso-scale twisted helical tube bundles

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Abstract. Heat exchangers with helical tubes are applied widely in various industrial fields and cryogenic applications. Currently, the most common ones are the spiral wound heat exchangers, which consists of many layers of spiral tubes with different coil diameters in a closed shell. Most of the fluid in the shell-side passes directly through the gap between two adjacent layers, which weakens the turbulence and the associated heat transfer coefficient. A novel twisted helical bundle geometry is proposed in this paper of which the thermal performance is explored using CFD simulation. The heat exchanger uses helium gas as the working fluid and operates between 300 K and 30 K. The curve of each tube in the bundle is generated by different 3D sinusoidal equations, and the tubes at adjacent layers are twisted together thereby producing the same end-to-end length for each tube. By simulating a spiral wound heat exchanger using different turbulent equations and comparing the results with experimental data from literature, the k-omega SST model is found to be the most suitable one to simulate this kind of geometry. Nusselt numbers and friction factors are obtained under different Re values and compared with those of standard spiral wound heat exchangers. The creative twisted geometry can improve the Nu number although also increasing the friction factor. Geometry parameters such as pitch length are being further explored to obtain an optimal thermal performance.

1. Introduction

Heat exchangers with meso- and micro-scale channels have been a research trend due to their high ratio of surface area to volume, lower total mass and low inventory of working fluids. Many studies have focused on the flow and heat transfer characteristics in micro- and meso-scale channels, and early transition to turbulence and thermal enhancement was noticed [1]. In 1994, Wang and Peng found that the fully developed turbulent heat transfer regime was initiated at Reynold’s number (Re) values ranging from 1000 to 1500. They suggest that the early transition to turbulence might be partially attributed to the variable Re values [2]. In 1995, one of the first studies, carried out by Gui and Scaringe, ascribe enhanced heat transfer coefficients in meso-scale channels to three key factors, which are the channel entrance effects, the pre-existing turbulence at the inlet and the surface roughness which leads to a thinner thermal boundary layer. They also stated that the critical Re number is around 1400, instead of 2300 [3].

In recent years, CFD simulation has become an important method to explore various geometry shapes and help to optimize the design [4 5 6 7 8 9 10]. Many Nu and friction factor correlations have been proposed based on different geometry factors [11 12 13]. Along with advances in manufacturing,
more and more shapes have been analysed to improve the thermal performance of heat exchangers, usually by producing a thinner average thermal boundary layer as a result of adding interruptions and promoting flow mixing via secondary flows. Since highly compact flow passages are generally continuous without any interruptions [1], helical structures turn out to be a good option. In 1999, Rush and Newell found that in sinusoidal wavy passages, even when the flow is in the laminar regime, some macroscopic mixing occurs near the channel exit at a relatively low Re values that is different than the regular circulation. The mixing moves toward the entrance as the Reynold’s number is increased, as does the Nusselt number enhancement [14]. Heat exchangers with compact helical tubes are applied widely in various industrial fields and cryogenic applications. Currently, the most common ones are the spiral wound heat exchangers, which consist of many layers of spiral tubes with different coil diameters in a closed shell. There have been many experimental and simulation studies on spiral wound heat exchangers and various correlations have been proposed [15 16 17 18].

The flow and heat transfer characteristics in micro- and meso-scale channels are complicated and still unclear. In spiral wound heat exchangers, the coil diameters of the small tubes on different layers are quite different, which will result in a large variation of the internal flow heat transfer coefficient [13]. This problem will further lead to a non-uniform flow and temperature distribution in both tube- and shell-side, reducing the effectiveness. In this paper, a novel geometry using twisted helical tubes with same length and coil diameter is proposed, of which the thermal performance has been studied and compared with that of spiral wound geometry.

2. Model analysis

2.1. Design specification and geometry

The detailed design objectives are listed in Table 1. The working fluid is helium, and the temperature range is from 300 K to 30 K, which means the thermal properties and the Re value of the working fluid vary significantly through the heat exchanger channel. To build a whole heat exchanger model and obtain its effectiveness correlation, the Nu and friction factor correlation in a fully developed region based on the Re value and the geometry factor should be developed first, for both the tube- and shell-side. This paper will focus on the simulation for the shell side flow in two kinds of geometry. Both spiral geometry and twisted geometry are shown in Figure 1. For the spiral wound heat exchangers shown in Figure 1(a), the pitch on each layer is related to their coil diameter so that the tubes in different layers end up with the same length, thereby avoiding a non-uniform flow distribution. However, most of the fluid in the shell-side passes directly through the gap between two adjacent layers, which weakens the turbulence and the associated heat transfer coefficient.

In the twisted bundle geometry, the curve of each tube in the bundle is generated by different three-dimensional sinusoidal equations, and the tubes at adjacent layers are twisted together thereby producing the same end-to-end length and coil diameter for each tube, shown by Figure 1b and 1c. Such an approach eliminates the non-uniform heat transfer coefficient issues in the radial direction. Unlike the spiral geometry, the shapes of the cross sections parallel to the axial direction vary in the circumferential direction, and the continuously changing cross-sectional shape is expected to enhance the heat transfer coefficient. The influence of four geometry factors will be explored, the tube diameter, the pitch, the number of tubes in one bundle, and the coil diameter.

| Table 1. Design objectives. |
|-----------------------------|
| Working fluid | Helium |
| Temperature span | 300 K to 30 K |
| Operating pressure | 320 psig (supply side), 100 psig (return side) |
| Pressure drop | <0.5 bar for both sides |
Flow rate | 13 g/s  
Materials | Stainless steel or copper  
Mass | <60 lbs  
Approximate size | <0.7 m height, width/length<br/>&lt;0.3 m  
Effectiveness | &gt;0.99

Figure 1. (a) Spiral geometry, $D_t$=1 mm. (b) Twisted geometry, pitch=50 mm, $D_t$=1 mm. (c) Twisted geometry, pitch=25mm, $D_t$=1 mm.

2.2. Mesh generation and Fluent setup
The mesh is generated using the tetrahedral independent method. Since there are many curves and small gaps in the geometry, not only the maximum and minimum cell size but also the curvature normal angle and the number of cells across the gaps have a large influence on the cell. These four parameters have been manipulated to obtain different cell numbers for the mesh independence analysis, and results show that the heat transfer coefficient and pressure drop tend to be steady when the number of cells is larger than 9.6 million, for a twisted model with 36 tubes and a pitch length of 50 mm.

The heat exchanger uses helium gas as the working fluid and operates between 300 K and 30 K. Since the thermal properties of helium vary significantly in this temperature range, and so does the Re value, a fully developed region of the heat exchanger is simulated to obtain the Nu value and friction factor correlation based on the Re value and geometry parameter. Fluent is used to simulate the heat transfer performance of a short shell-side part of this novel geometry. By simulating a spiral wound heat exchanger using different turbulence equations (k-epsilon model, k-omega model, and LES model) and comparing the results with experimental data from literature, the k-omega SST model is found to be the most suitable one to simulate this kind of geometry. Both the wall of the mandrel and the outer wall are adiabatic. The velocity is specified at the inlet and the pressure is specified at the outlet. A constant heat flux boundary condition is used on the wall where heat transfer occurs. Since a small amount of heat flux (20W) is applied to the wall, the temperature of the flow in the domain is close to uniform, as is that of the wall. Thus, the difference between the average flow temperature and the average wall temperature is used to calculate the heat transfer coefficient. The hydraulic diameter is defined as four times the volume of flow region divided by the heat exchange area.

3. Results and discussions
The aim of the simulation is to develop a Nusselt number and friction factor correlation for both the twisted bundled geometry and the spiral wound geometry. However, during the simulation, we observed strong and continuous vortex shedding in the spiral wound geometry, and the computational cost is very large for this kind of unsteady flow. Thus, a Nusselt number correlation for the spiral wound geometry from the literature based on experimental data is also used for the comparison.

3.1. Comparison of the thermal performance
A transient model is used to track the variation of the velocity distribution as a function of time. Figure 2 shows that under a relatively high Re value, an oscillation begins at around 0.2 s and vortex
shedding continually moves up and down, which means the flow remains unsteady. Under this condition, the simulation can only move forward 0.1 s per day even when run on UW-Madison’s High Performance Computing Center’s cluster. When the Re value is smaller than 7000, the oscillation is much milder, and the flow achieves a steady state after 0.6 s. It is therefore difficult to propose our own correlation for the spiral geometry due to the large computational cost. Figure 3 shows that the Nu values of the twisted bundle geometry with pitches of both 25 mm and 50 mm are larger than those of the spiral wound geometry simulation data and a correlation of the spiral wound geometry [16]. For the spiral wound geometry, there are two layers of tubes, and the pitch of the inner layer is 24 mm while that of the outer layer is 48 mm, which guarantees that all the tubes end up with the same length. Figure 4 shows that although the Nu value of the twisted geometry is obviously higher, the friction factors of these two kinds of geometry are at the same level, which makes the twisted one a promising novel geometry.

3.2. Exploring the entrance effect and early transition
When simulating the model with a whole length of 50 mm, it has been found that even under similar Re values, the Nu value at 30 K is much smaller than that at 300 K. For example, for a model with 36 tubes and a Re value around 2400, the Nu value at 30 K is 61.8 while the Nu value at 300 K is 101.8. One possible explanation is that the entrance effect and transition to turbulence may be different for the two cases. Exploring this idea, the Nusselt number and friction factor have been determined for different sections of the modelled geometry, as shown in Table 2. Both values associated with the second half of the geometry are smaller than those averaged over the whole geometry, while those from the fourth quarter increase again. These values indicate that the flow has not reached the fully developed region in the 50 mm long model, and the entrance effect is promoted by high temperature and high velocity.
To reduce the influence of the entrance effect, the length of the geometric model is extended to 100 mm. Both the Nusselt number and friction factor calculated from various parts of this model are shown in Figure 5 and Figure 6. For all three working conditions, the change of both values from the second half to the last quarter is small, less than 1%, which indicates that the flow is fully developed in the second half. It should be noticed that for a 100 mm long model, the Nu value at 30 K is a little larger than that at 300 K, even though the Re value at 30 K is a litter smaller, while as described above, for a 50 mm long model, the Nu value at 30 K is much smaller than that at 300 K. This difference may be caused by an early transition to turbulence under high temperature and high velocity. An experimental study also found that with a high inlet temperature and velocity, the transition occurs earlier and the transition zone in which the Nusselt number remain constant is longer [2]. More data under different working conditions is needed to explain this phenomenon.

Table 2. Nu and friction factor calculated from different parts, for a model 50 mm long.

| Inlet temperature (K) | Inlet velocity(m/s) | Re   | Nu    | Friction factor |
|-----------------------|---------------------|------|-------|-----------------|
| Whole                 | 300                 | 1.5001 | 2442.5 | 101.8 | 1.30           |
| Second half           | 300                 | 1.8214 | 2965.6 | 96.8  | 0.77           |
| Fourth quarter        | 300                 | 1.6269 | 2648.9 | 98.4  | 0.95           |

Figure 5. Nu calculated from different parts of a 100 mm long model.

Figure 6. Friction factor calculated from different parts of a 100 mm long model.

Figure 7 and Figure 8 show the Nu and friction factor calculated from the second half of the 100 mm long models. The influence of the pitch, the number of tubes, and the diameter of inner tube is initially explored. The model with less tubes, longer pitch and larger inner tube diameter has the largest Nu value. It is supposed that there is more space to develop vortex shedding patterns in such a condition, which will enhance bulk fluid mixing and the heat transfer coefficient. On the friction factor graph, there are still two irregular points, identified with the black arrows and both occur at 30 K. There is a order of magnitude difference between the density and viscosity of helium at 30 K and 300 K, which may cause this abnormality. The heat transfer behaviour under low temperature and the influence of these three geometry factors as well as the coil diameter are going to be further studied. For each factor, three different values will be assigned and the signal-to-noise ratio (SNR) will be calculated based on Nu and friction factor simulation data under different working conditions in every case. The SNR is usually used in quality engineering, which was invented by Taguchi [19] and can help engineers find out which levels of control factors are more efficient. In recent years, the Taguchi method are also adopted to estimate the effect of the geometrical factors on flow and heat transfer and help designing heat exchangers [20 21].
4. Conclusion
A novel twisted helical tube heat exchanger is proposed in this work, for which the Nu value and friction factor in the fully developed region is calculated through a fluent simulation. The resulting values are considered advantageous compared to those obtained with a spiral wound heat exchanger. An early transition from laminar to turbulent flow with a high inlet temperature and high inlet velocity is explored, however further data is needed to explain the phenomenon.

In the future, the Taguchi method will be used to make a thorough inquiry into the influence of different geometry factors and thereby optimize the design. Nusselt number and friction factor correlations based on Re values and a characteristic geometry factor will be proposed when sufficient data has been gathered, which will help to build a complete heat exchanger model and obtain the overall effectiveness.

5. References
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