Research on the Thermal-Hydraulic Performance of Twisted Helical Bundle Heat Exchangers

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ABSTRACT

A recuperative heat exchanger is an important component in a recuperative cryocooler system, which is commonly applied for the cooling of superconducting electronics, infrared sensors, power systems, etc. Since the flow in this kind of heat exchanger goes through a large temperature span, a twisted helical bundle heat exchanger is being designed and optimized to achieve miniaturization and high effectiveness. The thermo-hydraulic characteristics of flow in the shell side is being explored using CFD simulation. In this geometry, the curve of each inner tube is generated by 3-D sinusoidal equations, and the configuration can be specified by the number of bundles, number of tubes per bundle, twist pitch of the bundle and tube diameter. The influence of these parameters on the thermal-hydraulic performance has been investigated. The temperature and velocity distributions under steady state have been analyzed. Another geometry tool that has been utilized to identify better thermal performance and to optimize the design, is the geometry based conduction shape factor. Although it directly applies to solid conduction rather than convective heat transfer, it provides guidance regarding the influence of geometry on optimal thermal designs. Compared with spiral wound heat exchangers and staggered stacked slotted plate heat exchangers, the increase of both Nusselt number and friction factor is obvious. The tortuous flow path is assumed to be the reason for the thermal enhancement, due to its promotion of bulk flow mixing and redistribution of energy. The design of the overall heat exchanger includes manufacturability considerations, and a model of the complete heat exchanger will be built to obtain its overall effectiveness.

INTRODUCTION

Recuperative heat exchangers play an important role and have a large influence on the overall efficiency in many cryogenic systems. The motivation of this study is to design a compact meso-scale tube heat exchanger which can achieve large temperature range (300 K to 30 K), high effectiveness (>99%) and miniaturization at the same time. There have been many studies on the shell side flow of a circular tube fitted with helical structures, flow in the helical coiled tubes and flow in the spiral wound heat exchangers, but little studies on twisted helical-tube-bundle meso-scale heat exchangers were reported. Thus, a novel geometry is proposed in this study.

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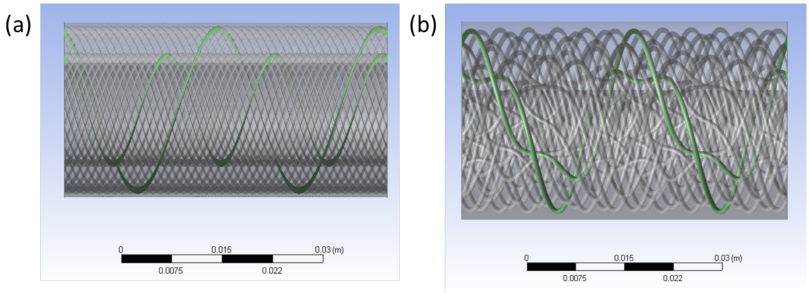

Figure 1. Comparison of how tubes are wound in (a) spiral wound heat exchanger (b) twisted helical-tube-bundle heat exchanger.

GEOMETRY AND MODEL

Geometry

In the twisted bundle geometry, the curve of each tube in the bundle is generated by different three-dimensional sinusoidal equations, and the equations are designed to produce the same end-to-end length and average coil diameter for each tube. From Figure 1, it is demonstrated that in a spiral wound heat exchanger, tubes in the first layer stay in the first layer without changing their position in the radial direction, while in twisted helical-tube-bundle heat exchanger, all tubes can spiral inward and outward in the radial direction during one revolution. This could enhance the fluid mixing and promote uniformity of the temperature distribution in the radial direction. Unlike the spiral geometry, the shapes of the cross sections parallel to the axial direction vary in the circumferential direction, shown in Figure 2, and the continuously changing cross-sectional shape is expected to enhance the heat transfer coefficient. Both results will increase the overall effectiveness of the heat exchanger. Based on Figure 3, it is reasonable to expect that the flow will move across more tube banks in the twisted helical-tube-bundle heat exchanger, which could augment fluid mixing and meanwhile increase both form drag and frictional drag. To optimize the design, the influence of three geometry factors will be explored, i.e., the tube diameter, the pitch, and the tube distribution pattern.

CFD model

The details of mesh, boundary conditions, working conditions and Fluent model have been introduced in the previous paper [1]. The shear stress transport (SST) k-omega model is used to simulate this turbulence problem. The model has been verified by simulating a spiral wound heat

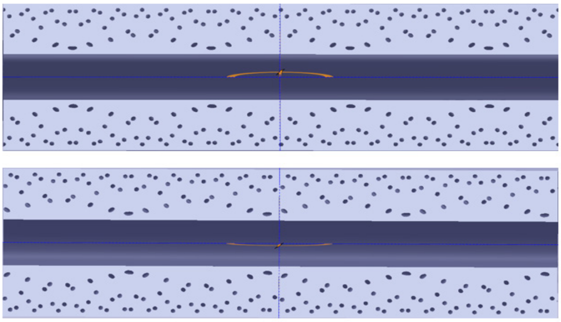

Figure 2. The shapes of two longitudinal cross sections staggered by 35°.

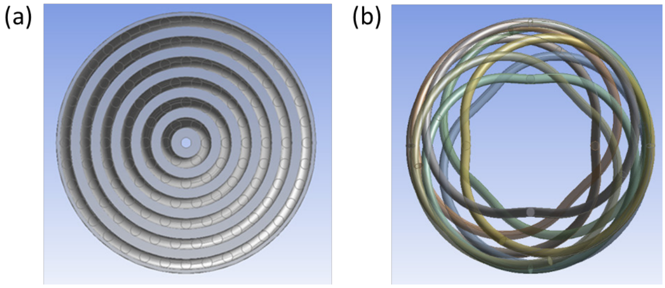

Figure 3. Comparison of (a) spiral wound heat exchanger (b) twisted helical-tube-bundle heat exchanger from the inlet view.

exchanger with experimental data from literature [2], which is also shown in the previous paper [1]. This paper will focus on the results analysis.

VELOCITY AND TEMPERATURE DISTRIBUTION PROFILE

Velocity and temperature field for the geometry with 36 tubes, pitch 25 mm and tube diameter 0.8 mm

For the geometry with 36 tubes, pitch length 25 mm and tube diameter 0.8 mm, the temperature distributions at steady state of two sections perpendicular to each other and parallel to the axis direction are shown as Figure 4. It is demonstrated that the shapes of the two sections are quite different. While the distribution of tubes in the radial direction in both sections seems not to be uniform, every tube continually changes its radial position in one cycle. These spiral-in-and-out tubes promote the fluid mixing. The distribution of temperature in the radial direction tends to be more and more uniform along the flow direction.

The temperature and velocity distributions on the sections perpendicular to the axis direction under steady state are shown in Figure 5 and Figure 6 respectively. Each section is separated by 10

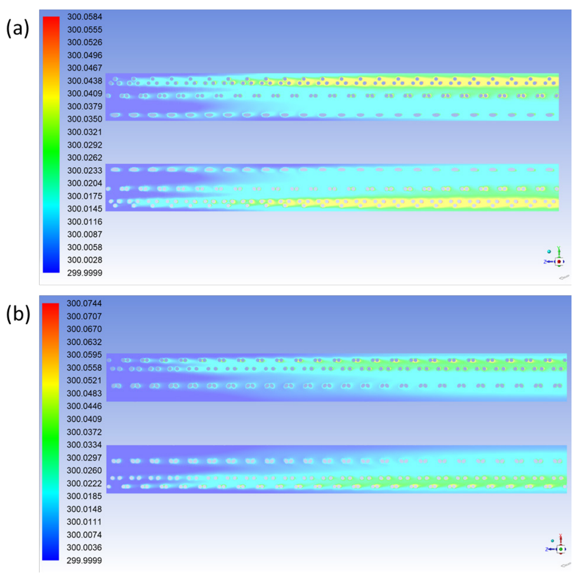

Figure 4. The temperature distributions of two sections perpendicular to each other and parallel to the axis direction.

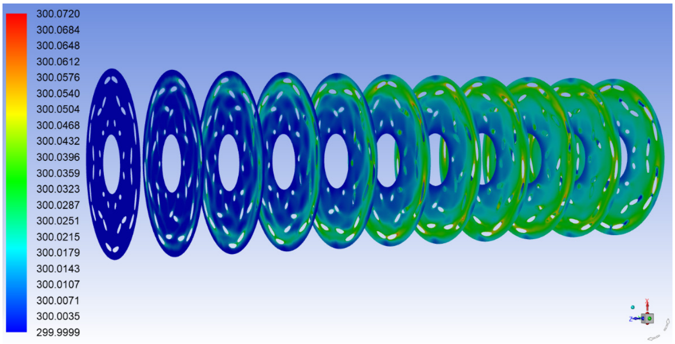

Figure 5. Temperature field under steady state of the geometry with 36 tubes, pitch length 25 mm, tube diameter 0.8 mm.

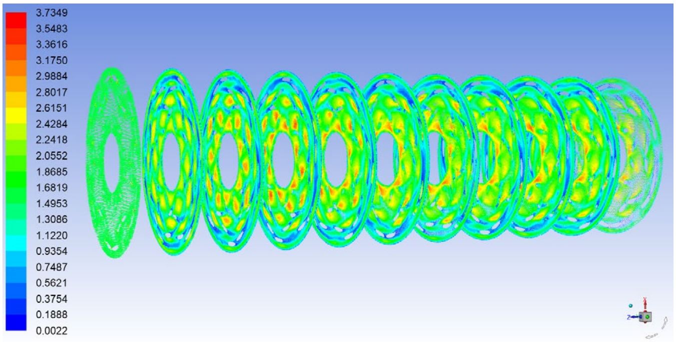

Figure 6. Velocity field under steady state of the geometry with 36 tubes, pitch length 25 mm, tube diameter 0.8 mm.

mm and the shape also varies with the position. In the first half of the geometry, the temperature and velocity distributions change a lot, which implies a developing region. In the second half of the geometry, the distributions tend to be steady, and a fully developed region is assumed. Thus, the average Nu value is calculated from the second half of the geometry.

Velocity and temperature fields for the geometry with 20 tubes, pitch 50 mm and tube diameter 0.6 mm

For the geometry with 20 tubes, pitch length 50 mm and tube diameter 0.6 mm, the temperature and velocity distributions are shown as Figure 7 and 8 respectively. Compared with the geometry

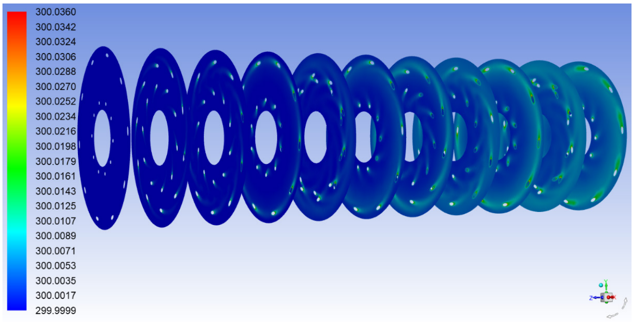

Figure 7. Temperature field under steady state of the geometry with 20 tubes, pitch length 50 mm, tube diameter 0.6 mm.

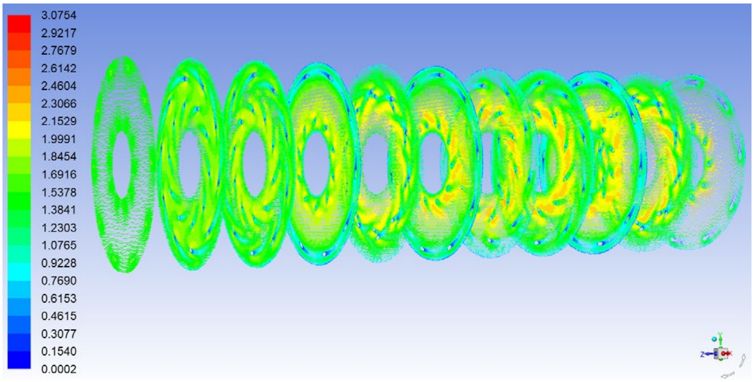

Figure 8. Velocity field under steady state of the geometry with 20 tubes, pitch length 50 mm, tube diameter 0.6 mm.

with 36 tubes, the distributions are more uniform. And this geometry provides a larger Nu value and friction factor. Since there is more space behind each tube for a vortex structure to develop, both the heat transfer coefficient and friction factor are larger compared to those of the geometry with 36 tubes. However, due to a smaller tube number, longer pitch length and smaller tube diameter, the heat transfer area is much smaller than that of the geometry with 36 tubes, thus the geometry with larger Nu number may have smaller conductance.

INFLUENCE OF SHAPE FACTOR ON NUSSELT VALUE AND FRICTION FACTOR

Introduction of conduction shape factor

The thermal performance of the twisted helical tube heat exchanger is influenced strongly by geometry parameters. Tube number, pitch value, inner tube diameter and tube distribution have been explored. The dimensionless pitch value is defined by the following equation, where p is the pitch length, D is the inner tube diameter.

$$\widetilde{p} = p/D \tag{1}$$

It should be noticed that two completely different geometries can have the same dimensionless pitch length, e.g., a geometry with p=25 mm, D=0.5 mm and a geometry with p=50 mm and D=1 mm. Two geometries with the same tube number, pitch length and inner tube diameter will have the same dimensionless pitch value and hydraulic diameter, but their distribution could be very different, e.g., a geometry with 4 tubes in one bundle, 9 bundles in total and a geometry with 6 tubes in one bundle, 6 bundles in total. A different distribution of tubes may lead to a different separation area formed behind tubes, which will have a large influence on the form drag. To quantify the intrinsic geometric relationship of tubes in a specific geometry, a concept from conductive heat transfer purview is "borrowed", which is the conduction shape factor S defined as:

$$S = 1/kR \tag{2}$$

where k is the conductivity of the material separating the surfaces and R is the thermal resistance between surfaces. The conduction shape factor represents the ratio of the effective area for conduction to the effective length for conduction, which is a measurement of how easy heat can be transferred through conduction between different surfaces. In our case, the conduction shape factor between one tube surface and all the other tube surfaces, the mandrel and the shell inner surface is investigated. Since the geometry is too complicated to calculate the effective area, the effective length or the thermal resistance, a steady state conduction model in Fluent is used. For every geometry, the flow area is changed to be uniform, e.g., Cu, the temperature of one tube wall is fixed, e.g., T_{one}=250 K, and the temperature of all the other tube surfaces, the mandrel and the shell inner surface are fixed at a different value, e.g., T_{others} =300 K. The inlet and outlet are set as adiabatic walls.

The choice of solid and boundary temperatures is arbitrary since it does not influence the result. Then the heat flux between one specific tube wall and all other surfaces is calculated by Fluent, and the conduction shape factor between one specific tube with all other surfaces is calculated by:

$$S=q/(k(T_{others}-T_{one})) \tag{3}$$

In our case, the conduction shape factor is introduced as a metric of geometry rather than a measurement of conductive heat transfer. A case with 27 tubes, 3 tubes in each bundle is shown in Figure 9. In the following part of this paper, the conduction shape factor of one tube refers to the conduction shape factor between this tube wall and all the other surfaces. In Figure 9, the conduction shape factors of the three tubes are calculated separately, and the temperature fields at steady state are displayed. All the three tubes have the same value S=0.28 m, thus it can be concluded that all the 27 tubes have the same conduction shape factor value due to symmetry. The consistency of the conduction shape factor indicates the uniform distribution of the tubes, which is very important for the performance of a heat exchanger.

Conduction shape factor among different geometries and its influence

The consistency of the conduction shape factor for different tubes in one geometry is general, while differences between different geometries exist and are obvious, which is shown in Figure 10.

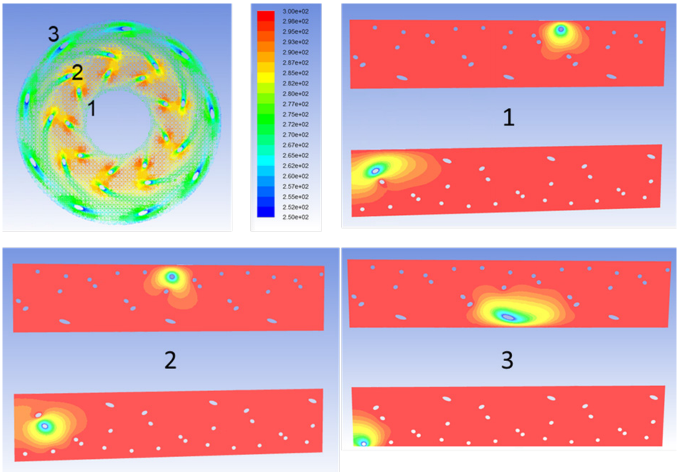

Figure 9. Conduction temperature field used to calculate the shape factor.

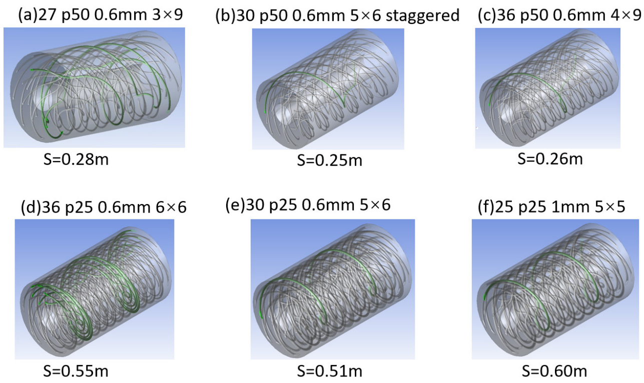

Figure 10. Shape factors of various geometries.

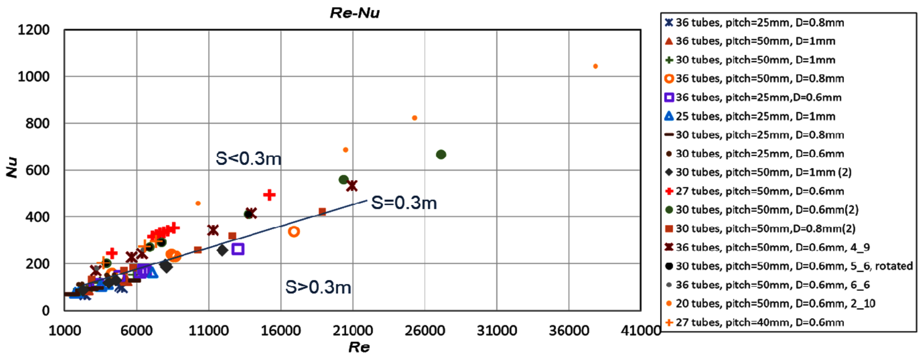

Figure 11. Nu-Re graph of different geometries.

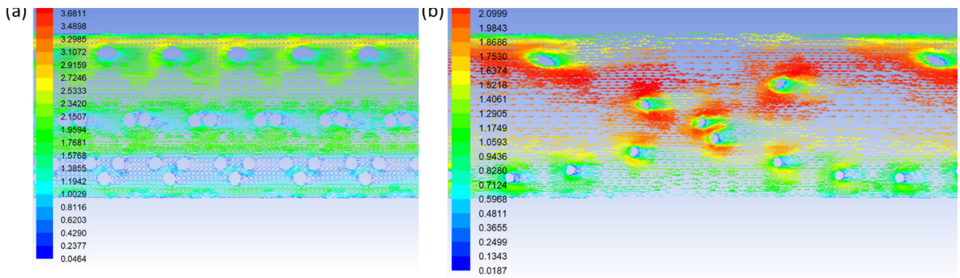

Figure 12. Velocity profile details for geometry (a) 36 tubes, pitch length 25 mm, D=0.8 mm and S=0.73 mm (b) 20 tubes, pitch length 50 mm, D=0.6 mm and S=0.25 m.

Generally, the geometry with a more compact configuration (more tubes, shorter pitch length) has a larger conduction shape factor. The conduction shape factor has a strong correlation with both the Nu value and friction factor. The Nu number of different geometries versus Re are shown in Figure 11. The geometries with a conduction shape factor smaller than 0.3 m have an obviously larger Nu value than other geometries. This is consistent with the conclusion of the previous section. Since the free space behind tubes is larger in geometries with a smaller shape factor, the vortex structure is more developed, thus the Nu value is higher. Furthermore, in a geometry with a small shape factor, there is some jet flow in the large open space where there is no tube, as shown in Figure 12(b), which is the case that should be avoided for heat exchanger design.

CONCLUSION

In this paper, a novel geometry using twisted helical bundled meso-scale tubes with uniform length and coil diameter is proposed, of which the thermal performance has been studied. Temperature and velocity distribution profiles of different geometries have been analyzed, although some geometries may have larger Nu number, their conductance needs to be further compared. The limitation of using the hydraulic diameter and dimensionless pitch length to specify a geometry is noticed, then the conduction shape factor is adopted to measuring the geometric relationship between tubes. The conduction shape factor is strongly related to both Nu number and friction factor. The CFD results will help to optimize the design and develop an overall heat exchanger model.

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