Numerical simulation of sinusoidal corrugated fins and serrated fins performance at low temperature

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Abstract. Due to promoting the disturbance of fluid and high heat transfer performance, sinusoidal wavy fins and serrated fins are used in plate fin heat exchangers (PFHE) for cryogenic system. In order to compare their flow and heat transfer performance difference at low temperature, this article adopts the numerical simulation method to investigate the flow and heat transfer characteristics of normal-temperature and low-temperature helium in corrugated wavy fin and serrated fin channels. The influence of different temperature and Reynolds numbers on the fin performance are analyzed. The results can provide theoretical guidance for the optimal design of cryogenic PFHE.

1. Introduction
Due to promoting the disturbance of fluid and high heat transfer performance, sinusoidal wavy fins and serrated fins are used in plate fin heat exchangers (PFHE) for cryogenic system, like Air Separation Unit (ASU), hydrogen liquefier, helium refrigerator/liquefier, etc.[1]. As the core component of the heat exchanger, those fins’ flow and heat transfer performances largely determine the overall performance of the heat exchangers. At present, the flow and heat transfer performance data of the fins for domestic plate-fin heat exchangers are mostly obtained through air blowing experiments at room temperature [2]. However, at low temperatures, the metal and fluid thermo-physical properties will change significantly, which will inevitably affect the performance of the fins and heat exchangers [3].

Based on the commonly used sinusoidal wavy fins and serrated fins, this paper uses a numerical simulation method to explore the flow and heat transfer characteristics of low-temperature and normal-temperature helium in corrugated wavy fin and serrated fin channels. The influences of different temperatures and Reynolds number on the flow and heat transfer performances of fins are analyzed. The research results could provide references for the design and selection of fin types and structure optimization of plate-fin heat exchangers in large-scale cryogenic systems.

2. Geometric structure and numerical model
The computational domains of fin channel are shown in figure 1. Both of them contain heat transfer fins, parting plates, entrance and exit parts. In order to make the flow fully developed, the length of the entrance part is extended 1.5 times of the hydraulic diameter. The downstream region is extended 5 times of the hydraulic diameter to avoid recirculation at the computational domain outlet.
Figure 1. Computational domains and setting of boundary conditions.

2.1. Mathematical model
The governing equations for solving the fluid flow and heat transfer problems could be expressed as follows in a uniform format:

$$\frac{\partial}{\partial t} (\rho \phi) + \sum \frac{\partial (\rho u_i \phi)}{\partial x_i} = \frac{\partial (\Gamma \nabla \phi)}{\partial x_i} + S_\phi$$  (1)
where $\rho$ is density, kg/m$^3$; $\phi$ is general-dependent variable; $u$ is velocity, m/s; $x$ is Cartesian coordinates, $\Gamma_\phi$ is diffusivity variable, $s_\phi$ is source term.

Reynolds number is defined as

$$Re = \frac{\rho u D_h}{\mu}$$

(2)

where, $\rho$ is density, kg/m$^3$; $D_h$ is hydraulic diameters of fins, m; $\mu$ is viscosity, m²/s; $u$ is inlet velocity, m/s;

The heat transmittance coefficient $h_0$ is calculated by:

$$h_0 = \frac{Q}{A_0 \Delta T_m}$$

(3)

where, $Q$ is heat transfer amount, W; $A_0$ is total heat transfer area, m²; $\Delta T_m$ is logarithmic mean temperature difference, K.

The heat transfer amount is calculated by:

$$Q = C_p m (T_{out} - T_{in})$$

(4)

The logarithmic mean differential temperature is calculated by:

$$\Delta T_m = \frac{T_{out} - T_{in}}{\ln \left( \frac{T_w - T_{in}}{T_w - T_{out}} \right)}$$

(5)

where $C_p$ is specific heat at constant pressure, J kg$^{-1}$ K$^{-1}$; $m$ is mass flow rate, g/s. $T_{out}$ is outlet temperature, K; $T_{in}$ is inlet temperature, K; $T_{wall}$ is wall temperature, K.

The Colburn factor $j$ and friction factor $f$ are quantitative indexes of heat transfer, flow resistance respectively. Eqs. (5) and (6) are applied to calculate the Colburn factor $j$ and Friction factor $f$:

$$j = \frac{h}{\rho u C_p} Pr^{2/3}$$

(6)

$$f = \frac{\Delta \rho D_h}{2 \rho u^2 L}$$

(7)

Where $Pr$ is Prandtl number.

2.2. CFD model and validation

The computational domain is meshed with hexahedral grids using Ansys ICEM CFD. In order to obtain faster computation speed, only the locations with significant flow changes such as velocity boundaries are meshed by concentrated grid density. The value of the dimensionless distance $y^+$ is always maintained at less than 1.

The velocity inlet and pressure outlet condition are set respectively. On the surface of upper and lower parting sheet constant temperature is set. The differential temperature between wall and inlet temperature is 10 K. The inlet temperature is 300 K and 20 K. The periodic boundary conditions are applied to both sides in the computational model. No slip wall and coupled thermal boundary condition are adopted at the interface between fluid and solid. The material of solid domain is set as Al 6061, and thermal radiation and nature convection are neglected. Because of the strong disturbance caused by fluid passing thorough fin arrays, the Re-Normalisation Group (RNG) $k$-epsilon model is adopted in this paper. The convergent criteria is that the normalized residuals are less than $1 \times 10^{-6}$ for flow equation and $1 \times 10^{-8}$ for the energy equation.

The grid independence test is carried out by adopting air. For sinusoidal corrugated fins, the selected fin type for grid independence test at Reynolds number of 2000 is 11.5-$3/8W$, with $h=9.525$ mm, $s=2.21$ mm, $r=0.254$ mm, $W=9.525$ mm, $A=0.99$ mm. Figure 2 shows the variation of Colburn
factor $j$ and friction factor $f$ with different grid number. It is found that since the $1 \times 3000$ grid number, the friction factor $f$ and Colburn factor $j$ no longer change significantly with the increase of grid number.

The number simulation results are compared with the air experiment results of Kays & London [4]. As shown in figure 3, Colburn factor $j$ and friction factor $f$ are in good agreement with the experimental data. The root mean square error (RMSE) between the numerical results of friction factor $f$ and the experimental data is 8.16%. The RMSE between the numerical results of Colburn factor $j$ and the experimental data is 5.84%.

3. Results and discussions
In order to compare the difference of the flow and heat transfer performance between the sinusoidal corrugated fin channel and the serrated fin channel at room temperature and low temperature, two
types of fin channels with same fin height $h$, fin spacing distance $s$ and fin thickness $t$ are simulated. The structural parameters of the two type fin channels are listed in table 1.

| Table 1. Parameters of sinusoidal corrugated fins and serrated fins |
|-----------------|----------------|-----------|-----------|-----------|-----------|-----------|-----------|
| Fin mode        | Fin type       | $h$/mm    | $s$/mm    | $t$/mm    | $l$/mm    | $A$/mm    | $W$/mm    | $D$/mm    |
| 95BW3003        | sinusoidal corrugated fins | 9.5       | 3.0       | 0.3       | -         | 1.5       | 15        | 4.17       |
| 95JC3003        | serrated fins  | 9.5       | 3.0       | 0.3       | 15        | -         | -         | 4.17       |

3.1. Flow resistance performance

![Figure 4. Friction factor $f$ comparison of sinusoidal wavy fins and serrated fins](image)

The friction factor $f$ comparison of those two types of fins is shown in figure 4. It can be seen that the variation trend of the friction factor $f$ with Reynolds number is the same at room temperature and low temperature, and both decrease with the increase of Reynolds number. Under the same Reynolds number at room temperature, the friction factor $f$ of the sinusoidal corrugated fin is higher than that of serrated fin, which indicates that the resistance characteristics of sinusoidal corrugated fin is higher at room temperature, and the difference of resistance characteristics between the two kinds of fins is almost unchanged at different Reynolds numbers. At low temperature, the friction factor $f$ of the serrated fin is higher than that of sinusoidal wavy fin when Reynolds number is small. With the increase of Reynolds number, the friction factor $f$ of the sinusoidal wavy fin begins to exceed that of serrated fin when Reynolds number is about 2000-3000. It indicates that at low temperature, when the fluid is in laminar flow region to transition region, the resistance characteristic of corrugated fin is small.

3.2. Heat transfer performance

The Colburn factor $j$ comparison of the two types of fins is shown in figure 5. It can be seen that the trend of Colburn factor $j$ with Reynolds number is the same at room temperature and low temperature, and both decrease with the increase of Reynolds number. The Colburn factor $j$ of the serrated fin is higher than that of the sinusoidal wavy fin at the same Reynolds number, no matter at room temperature or low temperature. The difference is more obvious in the laminar and transitional flow regions where Reynolds number is small. With the increase of Reynolds number, the difference
of Colburn factor $j$ between them becomes smaller and smaller. It also shows that the heat transfer ability of the serrated fin is better than that of the sinusoidal corrugated fin in different temperature regions. But when Reynolds number is large, this advantage is not obvious.

![Figure 5](image1.png)

**Figure 5.** Colburn factor $j$ comparison of sinusoidal wavy fins and serrated fins

### 3.3. $j/f$ Comparison

![Figure 6](image2.png)

**Figure 6.** $j/f$ factor comparison of sinusoidal wavy fins and serrated fins

The $j/f$ factor comparison of the two types of fins is shown in figure 6. It can be seen that the $j/f$ factor of the two types of fins decrease with the increase of Reynolds number at room temperature,
and the $j/f$ factor of the serrated fin is significantly higher than that of sinusoidal corrugated fin at the same Reynolds number. Combined with the analysis in sections 3.1 and 3.2, the $j/f$ factor of the serrated fin is higher and friction factor $f$ is lower at room temperature, that is, the heat transfer performance is good and the resistance is small. The comprehensive performance is better than that of the sinusoidal wavy fin. At low temperature, the $j/f$ factor of the two types of fins increase first and then decrease with the increase of Reynolds number. At the same Reynolds number, the $j/f$ factor of the serrated fin channel is still higher than that of sinusoidal corrugated fins, but the difference is not obvious when Reynolds number is small. Moreover, when the Reynolds number is low, the comprehensive performance difference between the two types of fins is small. At this time, although the serrated fin has strong heat transfer ability, the resistance is slightly larger. When the Reynolds number is high, the performance advantage of serrated fin begins to appear. Under almost the same heat transfer capacity, the resistance of serrated fin is less than that of the sinusoidal corrugated fin.

4. Conclusions
The flow and heat transfer performance of normal-temperature and low-temperature helium in corrugated wavy fin and serrated fin channels are simulated in this paper by using a validated three-dimensional numerical model. The influence of different temperature and Reynolds numbers on the fin performance are analyzed. The following conclusion are drawn:
1. The serrated fins overall performance is better than sinusoidal wavy fins no matter at room temperature and low temperature.
2. At cryogenic condition, when the Reynolds number is small, the overall performance difference between the two is very small.
3. As the Reynolds number increases, the overall performance advantage of the serrated fin gradually becomes larger.

References
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Acknowledgments
This work was supported by Comprehensive Research Facility for Fusion Technology Program of China (No. 2018-000052-73-01-001228), Anhui Provincial Nature Science Foundation (No. 1908085QE233), and the Science Foundation within the Institute of Plasma Physics, Chinese Academy of Sciences (No. DSJJ-18-04).