Thermal Science and Engineering 2025, 8(1), 11347.
https://doi.org/10.24294/tse11347

Article

Volume calculation and heat transfer performance simulation of a helical tube with cryogenic helium under intermittent flow condition

Wenhui Cui[1,2], Zhaoxue Zhang[1,2], Meimei Zhang[1], Han Zhou[3], Pengtong Meng[3], Qian Wang[3], Linghui Gong[1,2,*]

[1] Key Laboratory of Cryogenic Science and Technology, Technical Institute of Physics and Chemistry, Chinese Academy of Sciences, Beijing 100190, China
[2] University of Chinese Academy of Sciences, Beijing 100049, China
[3] Zhongshan Institute of Advanced Cryogenic Technology, Zhongshan 528400, China
*** Corresponding author:** Linghui Gong, lhgong@mail.ipc.ac.cn

CITATION

Cui W, Zhang Z, Zhang M, et al. Volume calculation and heat transfer performance simulation of a helical tube with cryogenic helium under intermittent flow condition. Thermal Science and Engineering. 2025; 8(1): 11347.
https://doi.org/10.24294/tse11347

ARTICLE INFO

Received: 16 January 2025
Accepted: 13 February 2025
Available online: 24 February 2025

Abstract: The intermittent flow cold storage heat exchanger is one of the most important components of the pulse tube expansion refrigerator based on the reverse Brayton cycle. In the experimental system, the volume and heat transfer of the helical tube play a decisive role in the stable operation of the whole experimental system. However, there are few studies on heat transfer in a helical tube under helium working medium and intermittent flow conditions. In this paper, a process and method for calculating the volume of a helical tube are proposed based on the gas vessel dynamics model. Subsequently, a three-dimensional simulation model of the helical tube was established to analyze the heat transfer process of cryogenic helium within the tube. The simulations revealed that the temperature of helium in the tube decreases to the wall temperature and does not change when the helical angle exceeds 720°. Moreover, within the mass flow rate range of 1.6 g/s to 3.2 g/s, an increase in the mass flow rate was found to enhance the heat transfer performance of the helical tube. This study provides a reference for the selection and application of a helical tube under intermittent flow conditions and also contributes to the experimental research of inter-wall heat exchanger and pulse tube expansion refrigerators.

Keywords: helical tube; refrigeration; intermittent flow; heat transfer; helium

1. Introduction

In recent years, cryogenic technology has become more and more important in many fields, such as electronic superconductivity and large scientific devices. However, there is no corresponding refrigeration system that can provide a cooling capacity of 10-100W in the liquid helium temperature zone [1]. Because of its advantages of high reliability, low vibration, and simple structure, the pulse tube expansion refrigerator based on the reverse Brayton cycle is a new type that is expected to meet this demand [2,3]. The compressed gas is aspirated, expanded, cooled, and vented by an expander controlled by cold valves alternately opened, and then enters the heat exchanger to transfer the cold energy [2]. Among them, as one of the core components, the inter-wall heat exchanger has a non-negligible effect on the performance of the refrigerator. Jia et al. [4] and Sun et al. [5] found through calculation and experimental research that especially the efficiency of the wall heat exchanger seriously affected the minimum temperature and efficiency of the refrigerator. In the corresponding experimental equipment, the helical tube heat exchanger is an important structure to provide reliable and stable working conditions. Due to the advantages of compact structure, good heat transfer performance, strong pressure capacity, etc., it has been widely used in refrigeration, power generation, and other fields [6,7]. More broadly, such intermittent flow operating conditions are common in many refrigerators, such as Stirling refrigerators [8], but few relevant studies exist [9]. Therefore, it is very vital and universal to study the heat transfer performance of a helical tube under intermittent flow conditions.

Pan et al. [9] numerically simulated the heat transfer and pressure drop of a helical tube with a rectangular cross-section under oscillating conditions, proving that it has excellent operating performance at high frequency and high speed, so it is more suitable for Stirling-type machines. Wang et al. [10] studied the low-temperature heat transfer phenomenon of nitrogen flowing in the helical tube under the combined influence of pseudo-critical conditions, floating lift, and helical tube curvature, to design heat exchangers in liquid air energy storage. The results show that when the fluid temperature is lower than the pseudo-critical temperature, the buoyancy effect predominates, whereas the centrifugal effect caused by coil curvature predominates in the opposite condition. Cha et al. [11] studied helical tube heat exchangers used in hydrogen refueling stations and found that they had better heat transfer performance than straight tube heat exchangers due to the influence of secondary flow. And when the distance between the coils is uniform, the heat transfer will be enhanced.

However, there are few studies on heat transfer performance and parameter analysis of helium working medium in the helical tube, especially in intermittent flow conditions. Therefore, based on the experimental system of the key intermittent flow wall heat exchanger in the pulse tube refrigerator, this paper calculates the gas storage volume of the indispensable helical tube and analyzes the heat transfer simulation of the helical angle, mass flow, and other parameters to provide a reference for the application of the helical tube in helium heat exchange and intermittent flow conditions.

2. Calculation about the gas storage volume of the helical tube

The experimental system for the intermittent flow cold storage heat exchanger is shown in **Figure 1**. The gas pressure, temperature, and mass flow rate in the pipeline undergo constant fluctuations over time as a result of the alternating opening and closing of solenoid valves a and b.

During the hot half cycle, with solenoid valve a opened and solenoid valve b closed, high-pressure helium enters the end-closed helical tube via solenoid valve a, leading to a continuous increase in pressure and temperature. Conversely, in the cold half cycle, solenoid valve a is closed while solenoid valve b is opened, causing the gas pressure within the helical tube to be released, resulting in a temperature reduction as low-pressure helium flows back to the compressor for pressurization. The next cycle commences when solenoid valve a is reopened and solenoid valve b is closed, and the process repeats. Importantly, to ensure the continuous and stable operation of the cycle under various experimental conditions, the selection of the appropriate helical tube is crucial.

In the intermittent flow condition, it is necessary to calculate the gas storage volume of the helical tube. The gas container dynamics model was adopted [12,13], where the duration of the hot half cycle and the cold half cycle is divided into several phases. When the time interval is small enough, it can be regarded as a steady state in each period. In each time, the filling process and venting process are considered reversible adiabatic processes, regardless of heat transfer loss. Helium is assumed to behave as the ideal gas. The compressor supply gas pressure is denoted as P_H, and the compressor return gas pressure is denoted as P_L.

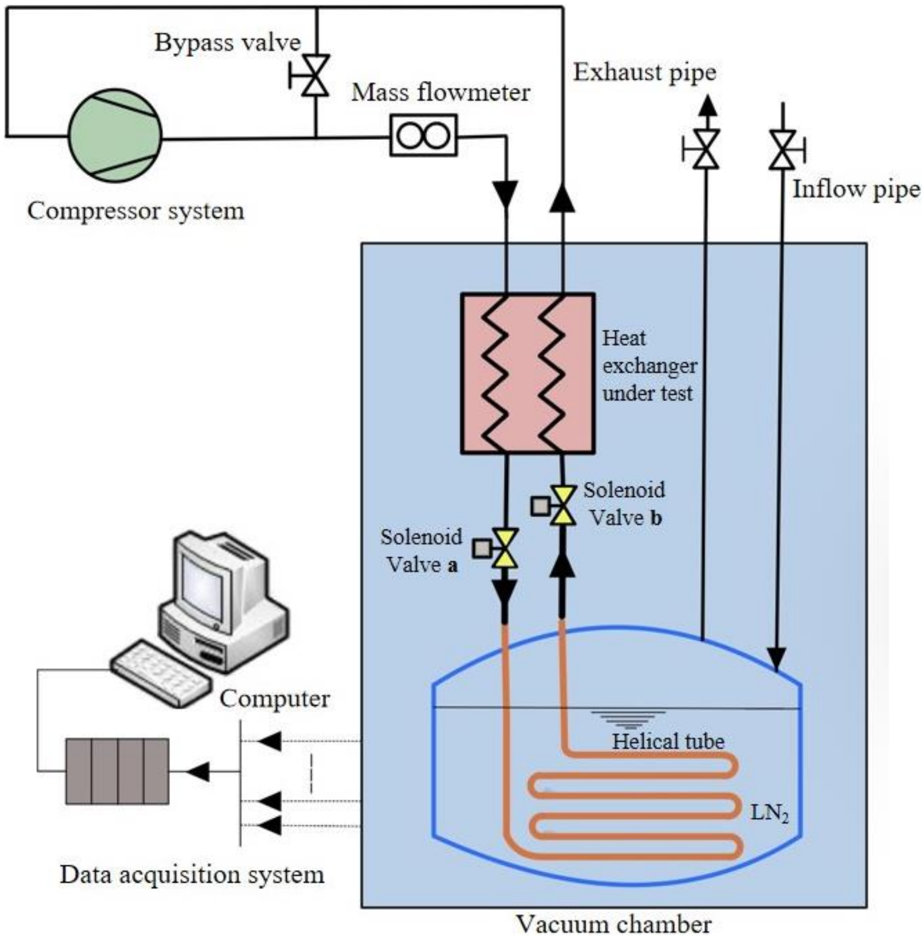


Figure 1. The flow chart of the experimental platform for intermittent flow cold storage surface heat exchanger.

2.1. Venting process

When venting, solenoid valve *b* is opened and solenoid valve a is closed. The isentropic expansion is assumed to occur during the venting process, represented by the process equation:

$$\frac{p}{\rho^{\gamma}} = \mathrm{C} \tag{1}$$

where γ is the gas-specific heat ratio, C is a constant.

Neglecting friction loss and potential energy changes, the mechanical energy conservation equation for the compressed gas can be simplified as:

$$\int vdp + \frac{u^2}{2} = 0 \tag{2}$$

In summary, the instantaneous velocity of the gas is:

$$u = \sqrt{\frac{2\gamma}{\gamma - 1}(p_i^{\frac{\gamma-1}{\gamma}} - p_L^{\frac{\gamma-1}{\gamma}})} \tag{3}$$

where p_i is the gas pressure at time t_i, Pa.

According to the continuity equation, the mass flow rate at time t_i can be obtained:

$$q_i = A\rho_i\sqrt{\frac{2\gamma}{\gamma - 1}(p_i^{\frac{\gamma-1}{\gamma}} - p_L^{\frac{\gamma-1}{\gamma}})} \tag{4}$$

In the process of venting, the gas density in the helical tube decreases continuously. According to the law of conservation of mass and the definition of density, the gas density in the helical tube after $\triangle t$ time can be obtained:

$$\rho_{i+1} = \rho_i - \frac{q_i}{V}\Delta t \tag{5}$$

where Δt is the time step, s; q_i is the mass flow rate of the venting process at time t_i, kg/s; ρ_{i} is the gas density at time t_i, kg/m^3; V is the gas storage volume of the helical tube, m^3.

Because venting in period is a reversible adiabatic process (Q = 0), the temperature equation in the helical tube can be derived using the first law of thermodynamics and the relationship between the internal energy and temperature of an ideal gas [14,15]:

$$\Delta U = Q - \int p dv \tag{6}$$

$$\Delta U = p\Delta v \tag{7}$$

$$\Delta U = c_v \Delta T \tag{8}$$

$$T_{i+1} = T_i - \frac{p_i}{(\rho_i^2 c_v)}(\rho_i - \rho_{i+1}) \tag{9}$$

where $\triangle U$ is the variation of gas internal energy, J/kg; Q is the exchange heat per kilogram of gas, J/kg; T_i is the gas temperature at time t_i, K; c_v is the constant volume specific heat capacity of gas, J/(kg·K).

According to the ideal gas equation of state, the gas pressure can be calculated:

$$p_{i+1} = \rho_{i+1} R_g T_{i+1} \tag{10}$$

where R_g is the gas constant of helium, J/(kg·K).

2.2. Filling process

When filling, solenoid valve a is opened and solenoid valve b is closed. Similarly, assuming the filling process is isentropic compression, the mass flow rate of inflation at time t_i:

$$q_i = A\rho_H\sqrt{\frac{2\gamma}{\gamma-1}\left(p_H^{\frac{\gamma-1}{\gamma}} - p_i^{\frac{\gamma-1}{\gamma}}\right)} \tag{11}$$

The gas density in the helical tube after $\triangle t$ time:

$$\rho_{i+1} = \rho_i + \frac{q_i}{V}\Delta t \tag{12}$$

The gas temperature in the helical tube:

$$T_{i+1} = T_i + \frac{p_i}{(\rho_i^2 c_v)}(\rho_{i+1} - \rho_i) \tag{13}$$

The gas pressure in the helical tube:

$$p_{i+1} = \rho_{i+1}R_g T_{i+1} \tag{14}$$

The calculation process of the gas storage volume of the helical tube is shown in **Figure 2**. It can be seen that the gas storage volume is closely related to the physical properties of helium, mass flow rate, and changing temperature and pressure in different periods; that is to say, it is more fundamentally affected by the key parameter of operating frequency.

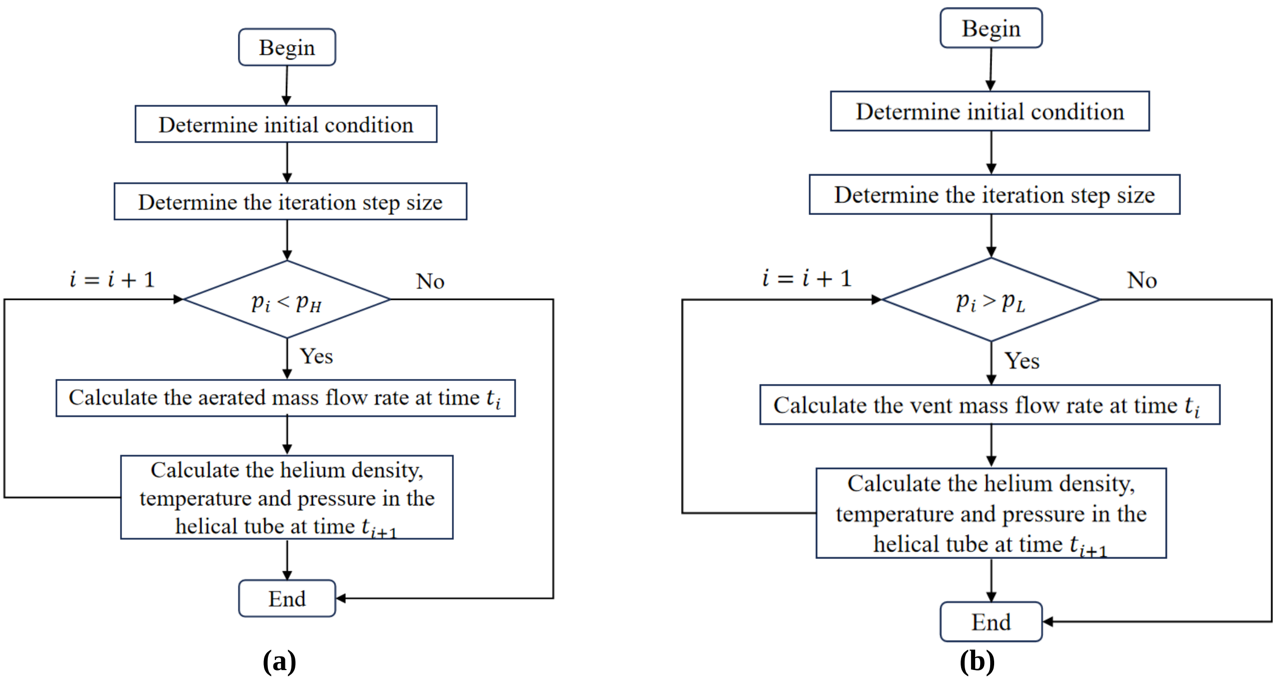
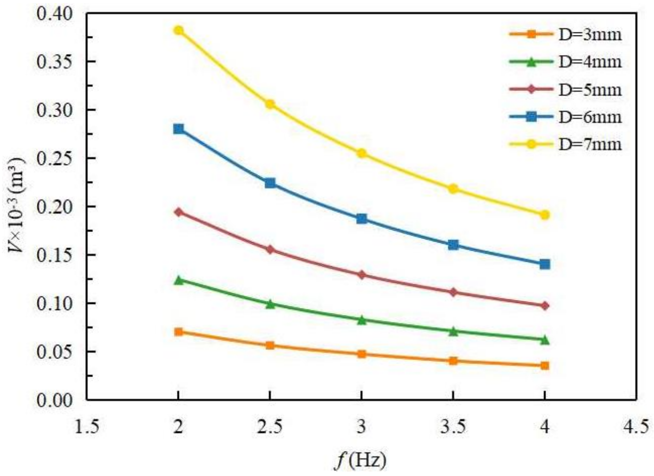


Figure 2. Helical tube gas storage volume calculation process: **(a)** Filling process; **(b)** venting process.

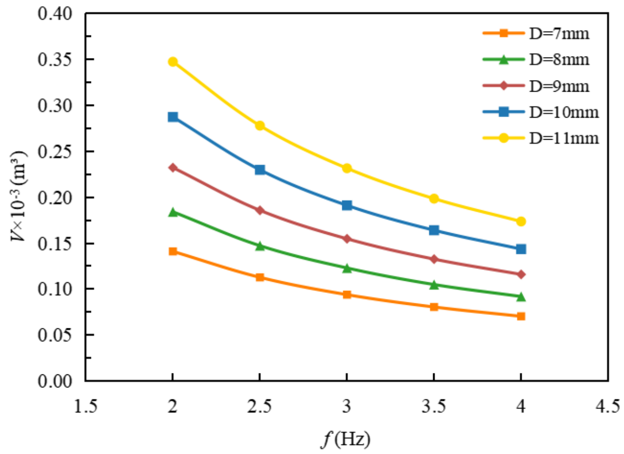
2.3. Calculation results

Given the supply gas pressure in the compressor of 2 MPa and return gas pressure of 0.5 MPa, ignoring the flow resistance through the heat exchanger, it is calculated that under different solenoid valve diameters, the gas storage volume of the helical tube changes with the working frequency of the solenoid valve, as shown in **Figure 3**.

As the operating frequency of the solenoid valve increases, the gas storage volume required by the helical tube becomes smaller. At the same operating frequency, the larger the diameter of the solenoid valve, the larger the gas storage volume required by the helical tube. At the same time, because the inflation process is a stable gas supply from a high-pressure gas source, and the pressure in the helical tube during the gas release process is decreasing, the average mass flow rate of the inflation process is greater than that of the gas release process, so for the helical tube with a given gas storage volume, the solenoid valve b should be larger than the valve diameter of the solenoid valve a.



(a)

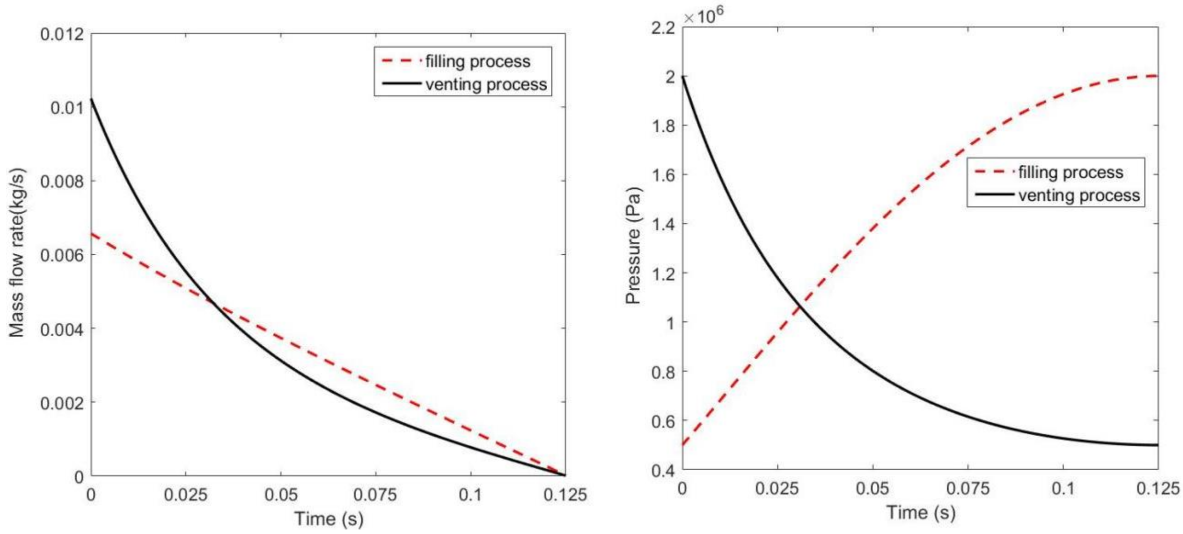


(b)

Figure 3. Under different solenoid valve diameters, the change of the gas storage volume of the helical tube with the working frequency: **(a)** Filling process; **(b)** venting process.

When the diameter of the solenoid value a is 5 mm and the frequency is 4 Hz, the gas storage volume of the helical tube is calculated to be 0.0975 L, and the diameter of the solenoid valve b is 8.23 mm.

The variation curve of helium mass flow rate and gas pressure in the helical tube with time is shown in **Figure 4**. The mass flow rate changes steadily from large to small to full, while the pressure in the tube increases rapidly and becomes stable. When the gas is discharged, the pressure in the tube decreases rapidly, which makes the mass flow rate decrease with a larger slope, unlike the high-pressure gas source when the gas is charged. Finally, the helical tube presents a dynamic cycle change condition.



(a) **(b)**

Figure 4. Variation curves of mass flow rate and pressure with time during the filling and venting process: **(a)** mass flow rate; **(b)** pressure.

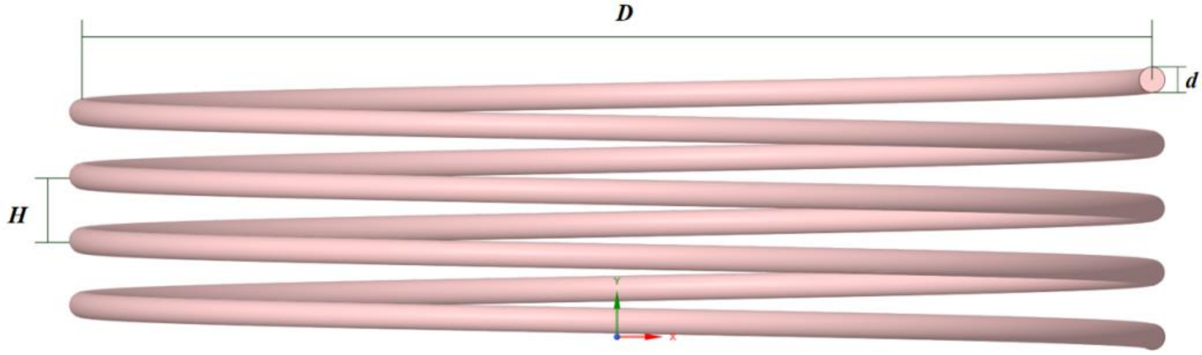
3. Simulation study on heat transfer characteristics of helical tube

3.1. Model and grid

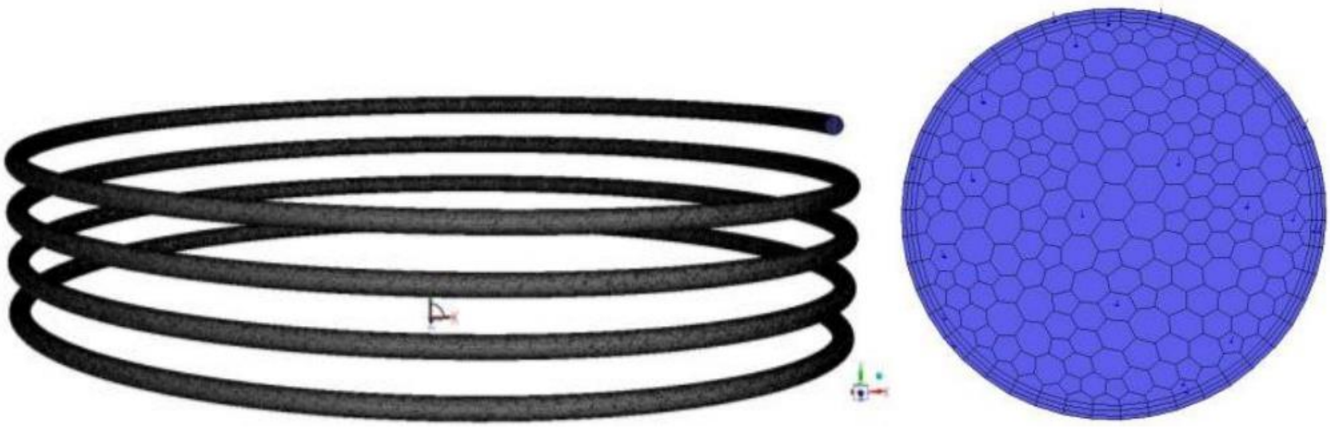
Modeling of the helical tube was achieved using SolidWorks, based on the volume calculations outlined in Section 2.3. The schematic diagram of the helical tube, as depicted in **Figure 5a**, illustrates key dimensions for the model, including a diameter (d) of 6 mm, a pitch (H) of 15 mm, a curvature diameter (D) of 250 mm, and a total length (L) of 3142.1 mm.

The polyhedral mesh division method was utilized [16], resulting in a total of 2,272,600 grids, and the mesh independence was verified. For the flow field capture of complex geometry such as helical tubes, the polyhedral mesh partitioning method can not only generate high-quality unstructured mesh that can adapt to complex geometry and improve the calculation accuracy, but also ensure the simulation has high stability and convergence and reduce numerical errors [17].

The mesh division of the helical tube inlet is shown in **Figure 5b**.



(a)



(b)

Figure 5. Geometry and mesh of the numerical model: **(a)** geometric model; **(b)** meshing of helical inlet.

3.2. Boundary conditions and solution methods

The numerical simulation of the helical tube is carried out by Fluent software with specific boundary conditions. The inlet boundary is set as a mass flow inlet with a temperature of 82 K, and the outlet boundary is defined as a pressure outlet. Additionally, the wall conditions are specified as equal wall temperature and non-slip boundary [18], with a temperature of 77 K, a wall thickness of 1 mm, and red copper as the wall material.

The simulation incorporates the helium real gas model of the NIST database for physical parameters and employs the RNG k-ε turbulence model [19–21]. RNG k-ε model takes into account turbulent anisotropy and vortex flow conditions [22,23], thus improving the accuracy of dealing with high turbulence at the end of the cavity [24], so it is widely used in the helical tube flow heat transfer model, which corresponds to the intermittent flow operating conditions in this paper [25].

The governing equations for momentum, energy, turbulent kinetic energy, and turbulent dissipation rate are solved using the second-order upwind scheme [26]. The mass flow at the inlet and outlet of the helical tube was monitored to ensure the convergence of the numerical simulation.

3.3. Results and analysis

The temperature contours of the model for various helical angles are shown in **Figure 6**, with all contours indicating the right side as the inner direction of the helical tube.

It can be seen that the fluid near the tube wall is first cooled after the helium enters the helical tube. Due to the centrifugal force, the temperature inside the helical tube decreases faster. As the helical angle escalates, the fluid temperature at different positions in the tube continues to approach, and until the section of 720°, the fluid temperature at different locations is basically the same.

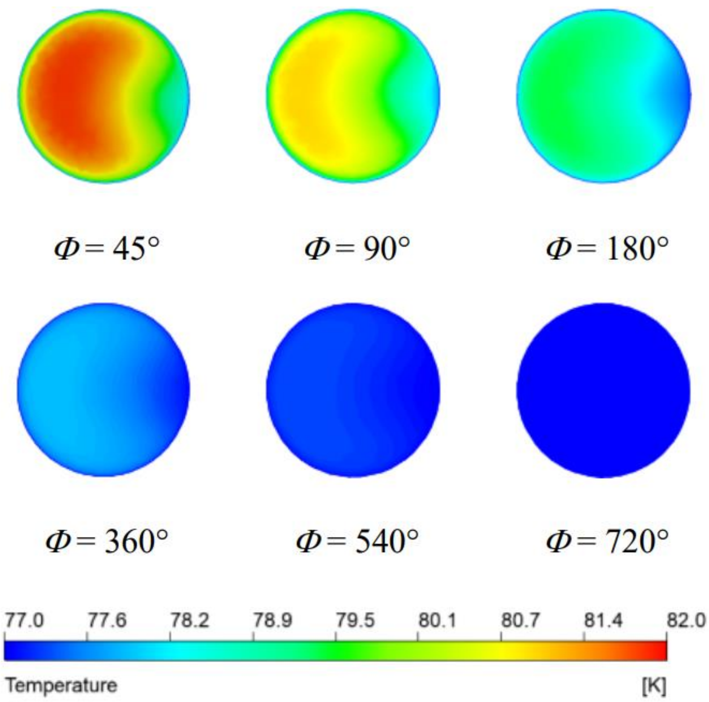


Figure 6. The contours of temperature at different helical angles.

Under different mass flow conditions, the variation curve of helium temperature with a helical angle is shown in **Figure 7**, and the variation curve of convective heat transfer coefficient with a helical angle is shown in **Figure 8**.

It can be seen that the smaller the mass flow rate, the faster the temperature reduction rate of helium, because at a lower mass flow rate, the heat exchange required to reduce the same temperature decreases so that the temperature can drop faster. When the helical angle is greater than 720°, it gradually becomes stable, and there is no significant difference in temperature among the three.

As can be seen from **Figure 8**, the heat transfer coefficient reaches its maximum at the entrance section and then decreases to stable. With the increase of mass flow rate, the heat transfer coefficient gradually increases, indicating that increasing mass flow rate within a certain range can improve the heat transfer performance of the helical tube under the condition that the outlet temperature of the helical tube is almost unchanged.

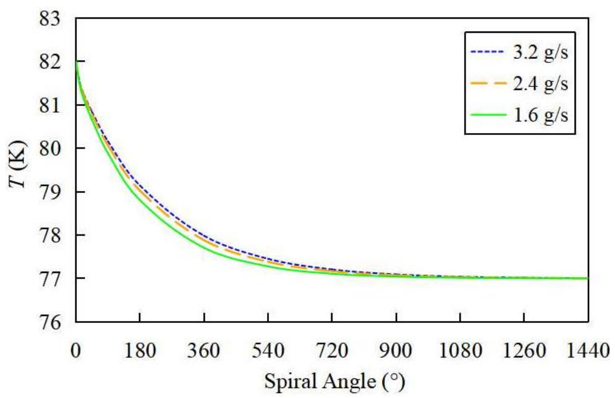


Figure 7. The variation curves of fluid temperature with helical angle at three different mass flows.

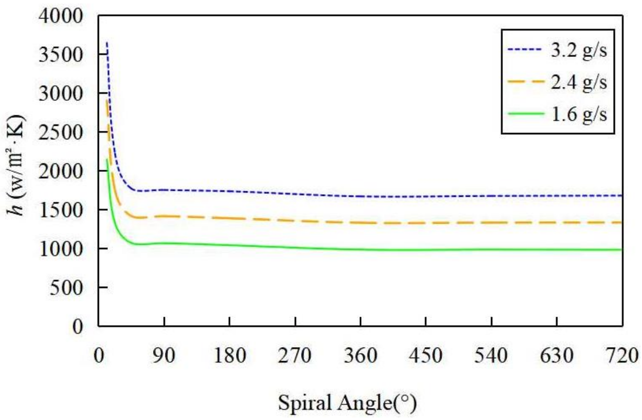


Figure 8. The variation curves of the convective heat transfer coefficient with helical angle at three different mass flows.

4. Conclusion

The volume of the helical tube in the experimental system of intermittent flow cold storage heat exchanger is calculated based on the gas container dynamics model. It is observed that the volume of the helical tube decreases as the working frequency of the solenoid valve increases. Conversely, the volume of the helical tube increases as the diameter of the solenoid valve increases. A three-dimensional numerical model was built to investigate the heat transfer characteristics of cryogenic helium in the helical tube. The study found that the flow of helium in the helical tube was influenced by centrifugal force, resulting in a temperature distribution with a radial gradient. It was observed that the temperature inside the helical tube decreased faster. When the helical angle exceeded 720°, the temperature of helium throughout the cross-section became uniform, reaching its lowest outlet temperature. The helium exhibits a faster rate of temperature reduction as the mass flow rate decreases within the range of 1.6 g/s to 3.2 g/s. Furthermore, increasing the mass flow rate can enhance the heat transfer coefficient. It provides a reference for the selection and application of a helical tube under intermittent flow conditions. Subsequently, the influence of helical tube curvature, section shape, and other parameters and the heat transfer law can be studied. Further, verification analysis will be carried out according to more practical and specific experimental results.

Author contributions: Conceptualization and methodology, WC and ZZ; software, WC; validation, MZ and LG; formal analysis, WC and HZ; investigation, PM; resources, QW; data curation, HZ; writing—original draft preparation, WC and ZZ; writing—review and editing, WC and ZZ; visualization, HZ and PM; supervision, QW, MZ and LG. All authors have read and agreed to the published version of the manuscript.

Acknowledgments: The authors would like to thank Zhongshan Advanced Cryogenic Technology Research Institute for their support in establishing the experimental platform.

Conflict of interest: The authors declare no conflict of interest.

References

1. Hu J, Luo E. Progress in regenerative cryocoolers. Science & Technology Review. 2015; (02): 99–107. doi: 10.3981/j.issn.1000-7857.2015.02.015.
2. Jia Q, Gong L, Feng G, et al. Thermodynamic analyses and the experimental validation of the Pulse Tube Expander system. Cryogenics. 2018; 91: 118–124. doi: 10.1016/j.cryogenics.2018.01.005
3. Cui W, Gong L, Jia Q, et al. Numerical study of heat transfer characteristics of intermittent flow cold storage surface heat exchanger. IOP Conference Series: Materials Science and Engineering. 2024; 1301(1): 012036. doi: 10.1088/1757-899x/1301/1/012036
4. Jia Q. Theoretical study and experimental verification of pulse tube expander and jet expander [PhD thesis]. Technical Institute of Physics and Chemistry, Chinese Academy of Sciences; 2020.
5. Sun W, Dai W, Liang J, et al. Research on pulse tube refrigerator with low temperature switch valve. Cryogenics and Superconductivity. 2000; (3): 1–9. doi:10.16711/j.1001-7100.2000.03.001
6. Zhu L, Cai T, Chen X, et al. Gas-solid flow behavior and heat transfer in a spiral-based reactor for calcium-based thermochemical energy storage. Journal of Energy Storage. 2024; 99: 113481. doi: 10.1016/j.est.2024.113481
7. Zhang Y, Wang D, Lin J, et al. Development of a computer code for thermal–hydraulic design and analysis of helically coiled tube once-through steam generator. Nuclear Engineering and Technology. 2017; 49(7): 1388–1395. doi: 10.1016/j.net.2017.06.017
8. Choudhari M, Gawali BS, Patil JD. Oscillating flow heat transfer: a comprehensive review. Energy Sources, Part A: Recovery, Utilization, and Environmental Effects. 2022; 44(3): 7598–7619. doi: 10.1080/15567036.2022.2113932
9. Pan C, Zhang T, Wang J, et al. CFD study of heat transfer and pressure drop for oscillating flow in helical rectangular channel heat exchanger. International Journal of Thermal Sciences. 2018; 129: 106–114. doi: 10.1016/j.ijthermalsci.2018.02.035
10. Wang Y, Lu T, Liu X, et al. Heat Transfer of Near Pseudocritical Nitrogen in Helically Coiled Tube for Cryogenic Energy Storage. Energies. 2022; 15(8): 2752. doi: 10.3390/en15082752
11. Cha H, Choi Y, Kim S. Heat transfer performance of a helical heat exchanger depending on coil distance and flow guide for supercritical cryo-compressed hydrogen. Progress in Superconductivity and Cryogenics. 2022; 24(3): 62–67. doi: 10.9714/PSAC.2022.24.3.062
12. Bejan A. Advanced engineering thermodynamics, 4th ed. Newark: Wiley; 2016.
13. Shen W, Tong J. Engineering thermodynamics, 5th ed. China: Higher Education Press; 2016.
14. Rahman MA. Review on heat transfer augmentation in helically coiled tube heat exchanger. International Journal of Thermofluids. 2024; 24: 100937. doi: 10.1016/j.ijft.2024.100937
15. Kaushik S, Singh S, Panwar K. Experimental study of fluid flow properties in spiral tube heat exchanger with varying insert shape over spiral tube profile. Materials Today: Proceedings. 2023; 80: 78–84. doi: 10.1016/j.matpr.2022.10.117
16. Yan F, Lu H, Feng S. Numerical Simulation of Liquified Natural Gas Boiling Heat Transfer Characteristics in Helically Coiled Tube-in-Tube Heat Exchangers. Frontiers in Heat and Mass Transfer. 2024; 22(5): 1493–1514. doi: 10.32604/fhmt.2024.055324
17. Zhao Y, Mao Q. Experimental and numerical analysis of unsteady state conditions on thermal storage performance of a conical spiral shell-tube energy storage system. Journal of Energy Storage. 2024; 88: 111579. doi: 10.1016/j.est.2024.111579
18. Kazemi-Esfe H, Shekari Y, Omidvar P. Comparison of heat transfer characteristics of a heat exchanger with straight helical tube and a heat exchanger with coiled flow reverser. Applied Thermal Engineering. 2024; 253: 123772. doi: 10.1016/j.applthermaleng.2024.123772
19. Bi H, Yang J, Chen C, et al. Heat transfer and flow characteristics of intermittent oscillating flow in tube. Applied Thermal Engineering. 2023; 225: 120233. doi: 10.1016/j.applthermaleng.2023.120233
20. Wu J, Zhao J, Sun X, et al. Design method and software development for the spiral-wound heat exchanger with bilateral phase change. Applied Thermal Engineering. 2020; 166: 114674. doi: 10.1016/j.applthermaleng.2019.114674
21. Li S, Cai W, Chen J, et al. Numerical study on the flow and heat transfer characteristics of forced convective condensation with propane in a spiral pipe. International Journal of Heat and Mass Transfer. 2018; 117: 1169–1187. doi: 10.1016/j.ijheatmasstransfer.2017.10.080
22. Dong X, Wu Y, Zhang C, et al. Experimental and numerical study on heat transfer and flow characteristics of molten salt

nanofluids in spiral-wound tube heat exchanger. International Journal of Thermal Sciences. 2023; 191: 108343. doi: 10.1016/j.ijthermalsci.2023.108343
23. Han Z, Reitz RD. Turbulence Modeling of Internal Combustion Engines Using RNG κ-ε Models. Combustion Science and Technology. 1995; 106(4–6): 267–295. doi: 10.1080/00102209508907782
24. Zhang D, Zhang G, Wang H, et al. Numerical investigation of time-dependent cloud cavitating flow around a hydrofoil. Thermal Science. 2016; 20(3): 913–920. doi: 10.2298/tsci1603913z
25. Pan C, Zhou Y, Wang J. CFD study of heat transfer for oscillating flow in helically coiled tube heat-exchanger. Computers & Chemical Engineering. 2014; 69: 59–65. doi: 10.1016/j.compchemeng.2014.07.001
26. Wu J, Lu S, Wang C, et al. Numerical study on heat transfer characteristics of helically coiled grooved elliptical tube heat exchanger. International Journal of Heat and Fluid Flow. 2024; 110: 109622. doi: 10.1016/j.ijheatfluidflow.2024.109622