ISSN: 2414-1895 DOI: 10.6919/ICJE.202504_11(4).0029

Design Analysis of Internal Heat Exchanger of Transcritical CO_2 Ice-Making System

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Abstract

This study investigates the internal heat exchangers in transcritical CO_2 refrigeration ice-making systems. A numerical model combining the distribution parameter method is developed to simulate heat transfer and pressure drop. Various pipe diameter combinations are compared to determine their required tube lengths and CO_2pressure drops on both high and low-pressure sides under given heat load conditions. Results indicate that Scheme 4 achieves the desired heat load with a shorter tube length (0.49 m) and lower high-pressure-side CO_2 pressure drop (3.6 kPa). This study provides a theoretical basis for optimizing internal heat exchangers to enhance the performance of transcritical CO_2 systems.

Keywords

Carbon Dioxide; Internal Heat Exchanger; Refrigeration Cycle.

1. Introduction

With the successful hosting of the 2022 Beijing Winter Olympics, more and more people are attracted to participate in ice and snow sports, and the scale of the events is getting bigger and bigger, which makes a number of international winter sports events successfully held in China. Therefore, the construction of modern and intelligent artificial ice rink is the guarantee for the smooth progress of ice and snow sports in China, and also promotes the prosperity and development of ice and snow sports in China [1].

As a heat return, heat exchanger device, not only the compressor suction temperature overheating state and prevent the compressor strike phenomenon, also can discharge supercritical CO_2 and the saturated steam of the evaporator outlet, can effectively reduce the outlet temperature of the refrigeration system, and prevent harmful flash steam, increase the air cooler outlet CO_2 over-cooling, reduce throttle loss, but also increase the suction temperature and pr enthalpy of the compressor, improve the circulation efficiency[2].Lu[3] conducted 3D simulation analysis of three different structural forms in the graphical user interface(GUI) of MATLAB software. The study shows that with the same number of ribs, increasing the rib thickness can improve the heat transfer efficiency and improve the overheat and supercooling degree. Ahcin Ziga[4] Scholars have assembled and analyzed the thermal efficiency, heat transfer and friction force of nine different internal heat exchangers, and evaluated the thermal hydraulic force of the oscillating flow shell tube internal heat exchanger for heat cooling.For the printed circuit board internal heat exchanger with supercritical carbon dioxide as the working medium, Zhang[5] adopted the method of combining numerical simulation and concomitant optimization. Compared with the traditional system, the heat channel heat transfer performance of the same pressure drop is improved by 24.3% and 29.3%, respectively, and the heat transfer performance of the cold channel is improved by 21.2% and 17.3%, respectively. Xiao[6] designed the open and closed switching heat pump drying system with an internal heat exchanger. It shows that the optimal SMER value can be obtained by using the heat pump drying system with open and closing switching type 2, thus improving the drying efficiency. Under different design conditions, Liu[7] increased the cooling capacity of the evaporator after installing the internal heat exchanger, thus effectively improving the performance of the automobile air conditioning system.Wu[8] designed a transcritical CO_2 cooling system with high and low temperature heat recovery. Under different outlet temperature and pressure conditions, the system operation efficiency and performance have been improved, but it also increases the total power consumption of the system. Liu et al.[9]set up a set of transcritical CO_2 parking air conditioning system with an internal heat exchanger, and studied the influence of the air cooler outlet temperature and exhaust pressure on the system performance of the system in the enthalpy difference room.

In this paper, the powerful calculation function of MATLAB software, using the distribution parameter method, respectively to the numerical model of heat transfer and pressure drop, with the cross critical CO_2 system performance simulation calculation. The results show that the mathematical model can reflect the changes in the actual operation process, and lays a foundation for further optimization design and optimal control.

2. Modeling and Methodology

In order to strengthen the heat transfer, this paper adopts the counter-current casing heat exchanger and the distribution parameter method, assuming that the heat transfer process inside the internal heat exchanger is in a stable state, and the flow in the tube is simplified to a one-dimensional flow problem. The pressure drop model of the internal heat exchanger is shown in Fig.1.

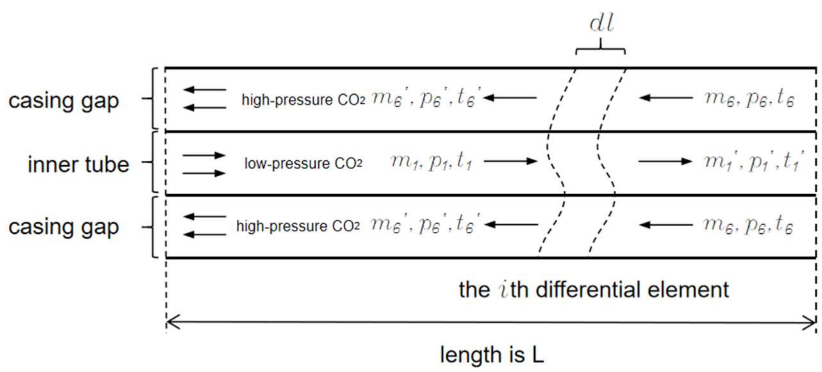

Fig. 1 One-dimensional flow model of the tube internal heat exchanger

In internal the heat exchanger, CO_2 temperature continues to rise, the heat from the high-pressure side CO_2 to the low-pressure side CO_2, and high-pressure side CO_2 outlet temperature and high-pressure side CO_2 inlet temperature should have certain temperature difference to ensure good heat exchange effect, heater outlet CO_2 temperature press selection

$$\Delta t_{hr} = t_{hr,CO_2,h,in} - t_{hr,CO_2,h,out} \tag{1}$$

Where Δt_{hr} is the high-pressure-side CO_2 inlet temperature, ℃; $\Delta t_{hr,CO_2,h,in}$ is the CO_2 outlet temperature on the high-pressure-side, ℃; $\Delta t_{hr,CO_2,h,out}$ is the heat transfer temperature difference, ℃.

In the absence of heat loss, the heat released by the high pressure CO_2 gas during the heat exchange process shall be equal to the heat absorbed by the low pressure CO_2 gas. The heat balance formula in the heat recovery process

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$$m_1(h_1'-h_1) = m_2(h_6 - h_6') \tag{2}$$

The calculation formula of total heat transfer coefficient is as follows

$$k = \frac{1}{\frac{1}{\overline{h}_{hr,CO_2}}\frac{d_{hr,CO_2,outer}}{d_{hr,CO_2,inner}} + \frac{d_{hr,CO_2,outer}}{2\lambda_{cu}}\ln\frac{d_{hr,CO_2,outer}}{d_{hr,CO_2,inner}} + \frac{1}{\overline{h}_{hr,h,CO_2}} + R_{dw}} \tag{3}$$

Where $d_{hr,CO_2,outer}$ is the internal heat exchanger CO_2 tube outer diameter, m; λ_{cu} is the thermal conductivity of copper, W/(m·K); R_{wg} is the dirt thermal resistance, m^2·K/W.
The calculation formula of the log-average temperature difference is as follows

$$\Delta t = \frac{(t_{hr,CO_2,in} - t_{hr,CO_2,out}) - (t_{hr,CO_2,in'} - t_{hr,CO_2,out''})}{\ln(\frac{t_{hr,CO_2,in} - t_{hr,CO_2,out''}}{t_{hr,CO_2,out} - t_{hr,CO_2,in'}})} \tag{4}$$

Where $t_{hr,CO_2,in}$,$t_{hr,CO_2,out}$, $t_{hr,CO_2,in'}$, $t_{hr,CO_2,out'}$ are the inlet and outlet temperature of CO_2 pipe on the low pressure side and the inlet and outlet temperature of CO_2 pipe on the highpressure side, °C.
Heat transfer area is calculated by using the following formula

$$\overline{A} = \pi d_{hr,CO_2,\ outer} \cdot L_{hr} \tag{5}$$

Where L_{hr} is the internal heat exchanger tube length, m.
Total heat transfer of the return heater is calculated as follows

$$Q_{hr} = k\overline{A}\Delta t \tag{6}$$

The supercritical CO_2 fluid changes the physical properties of the supercritical CO_2 fluid in the cooling process, which not only needs to study its heat transfer characteristics, but also needs to analyze its resistance inside the internal heat exchanger[10]. The calculation formula of the pressure drop is as follows

$$\Delta P_{hr} = \Delta P_{hr,f} + \Delta P_{hr,m} + \Delta P_{hr,l} + \Delta P_{hr,h} \tag{7}$$

Where $\Delta P_{hr,f}$ is the friction resistance pressure drop along the distance, kPa; $\Delta P_{hr,l}$ is the accelerate the drag pressure drop, kPa; $\Delta P_{hr,m}$ is the local resistance pressure drop, kPa; $\Delta P_{hr,h}$ is the static column power pressure drop, kPa.

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3. Results and Discussion

In this paper, the copper pipes of different pipe diameter are combined, and five pipe diameter combination schemes are listed as shown in Table 1. The inner pipe flows through the supercritical CO_2 fluid and adopts the spiral double or three channels in parallel, and the outer pipe flows through the cooling water high pressure CO_2, which can not only increase the heat transfer area but also enhance the disturbance reinforcement heat transfer.

Table 1. Table of combination of internal heat exchanger

Designation	Outer tube	Inside tube	Internal tube form
Scheme1	*Φ*16mm×0.75mm	*Φ*7mm×0.75mm	two channel
Scheme2	*Φ*16mm×0.75mm	*Φ*6mm×0.75mm	three channel
Scheme3	*Φ*19.05mm×0.75mm	*Φ*7mm×0.75mm	two channel
Scheme4	*Φ*19.05mm×0.75mm	*Φ*6mm×0.75mm	three channel
Scheme5	*Φ*26mm×0.75mm	*Φ*6mm×0.75mm	three channel

Fig. 2 shows that with the increase of pipe length, the actual heat production also increases, and the five schemes all break the design condition of 0.36kW. However, the low-pressure side CO_2 pressure drop is different. In the case of determining the outer tube diameter, the inner tube flow of the double and three channels is the same, and the increase of flow rate leads to a significant increase in pressure drop. Scheme 1 and Scheme 3 achieve the required heat production at the tube length of 0.45m and 0.75m. The heat growth rate of the three-channel scheme 4 and scheme 5 decreases with the increase of the diameter of the outer pipe. The schemes 2, 4 and 5 meet the heat capacity in the pipe length 0.28m, 0.49m and 1.14m respectively.

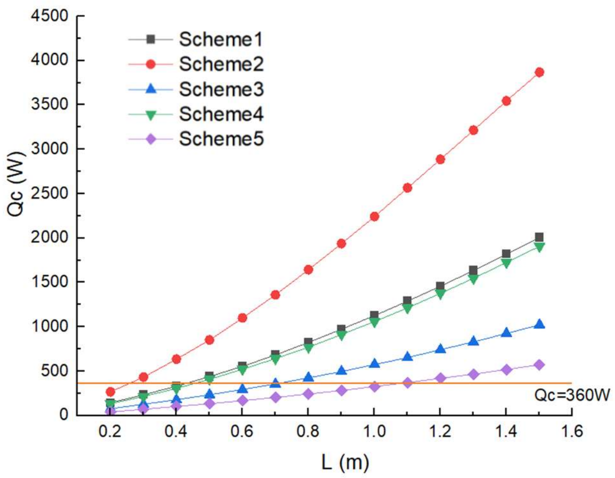

Fig. 2 The actual heat production of the internal heat exchanger changes with the tube length

Fig. 2 shows that the required pipe length of Scheme 2 is the shortest, but Fig. 3 shows that the CO_2 pressure drop on the high-pressure side changes significantly. Due to the small diameter of the outer pipe, the flow rate increases. The CO_2 pressure drop on the high-pressure side of schemes 1 and 2 increased greatly. The CO_2 pressure drop on the low-pressure side is relatively high to optimize the flow rate distribution and improve the heat transfer efficiency. Although it seems to be contradictory with the pressure drop characteristics of the high-pressure side, the higher heat change does require

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greater flow rate, which leads to the demand for larger pressure drop on the low-pressure side. Therefore, the scheme 4 can use a shorter pipe to meet the heat production and import and export temperature, but also can achieve lower high-pressure-side CO_2 pressure drop and pipe length is moderate.The design parameters of the internal heat exchanger are shown in Table 2.

Table 2. Design parameters of the internal heat exchanger

Type	Material quality	Pipe diameter/mm	Length/m	Pressure drop/kPa
Reverse flow type double-pipe heat inter-changer	copper pipe	outer tube: Φ19.05×0.75 inside tube: Φ6.00×0.75 (three channels)	0.49	low-pressure side CO_2: 46.4 high-pressure side CO_2: 3.6

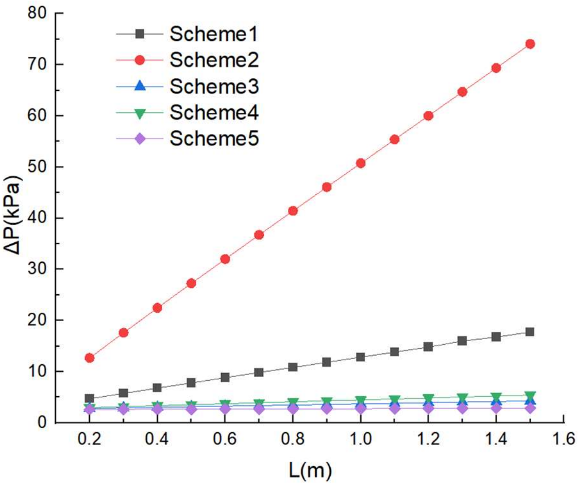

Fig. 3 Change of CO_2 pressure drop on the high-pressure side of internal heat exchanger with tube length

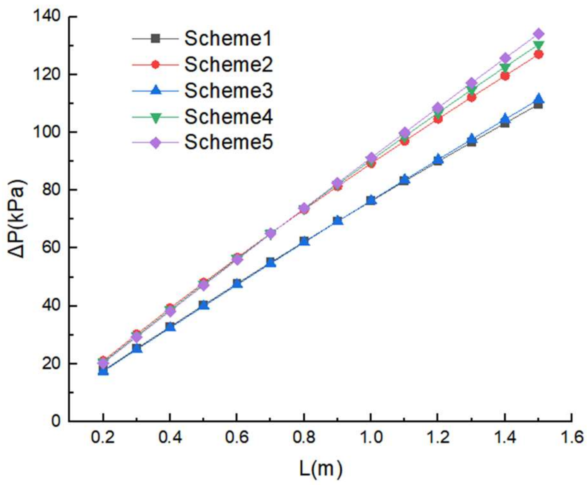

Fig. 4 Change of CO_2 pressure drop on the low-pressure side of internal heat exchanger with tube length

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4. Conclusion

This study investigates the internal heat exchangers in transcritical CO_2 refrigeration ice-making systems. A numerical model combining the distribution parameter method is developed to simulate heat transfer and pressure drop. The following conclusions can be drawn from this article:

(1) Among the different pipe diameter combinations, Scheme 2 requires the shortest tube length to meet the heat load, but it results in a significant increase in the high-pressure-side CO_2 pressure drop.

(2) Scheme 4 can meet the heat load requirements with a relatively short tube length (0.49 m) while achieving a lower high-pressure-side CO_2 pressure drop (3.6 kPa), offering a better overall performance.

(3) Optimizing the pipe diameter combination and tube length of internal heat exchangers can effectively reduce system pressure drop and improve heat exchange efficiency, providing valuable insights for the design of transcritical CO_2 refrigeration systems.

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