RESEARCH ON QUASI-TWO-STAGE COMPRESSION CYCLE CHARACTERISTICS OF REFRIGERATION SYSTEM FOR COLD STORAGE

by

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In view of the current test methods and evaluation indicators for the performance of cold storage freezing systems, a cold storage experimental platform with a micro-channel heat exchanger is designed using the inject technology, and the power consumption and cooling performance are tested under different conditions of the internal and external environment temperatures, vapor injecting flux, compressor speed, and vapor inject pattern. The experimental results provide us with a useful source for the optimization or improvement of cold storage freezing systems.

Key words: *freezing system, experiment platform, vapor inject technology, cooling performance*

Introduction

With the rapid development of China's economics, people's living standards have been continuing to be improved, and the demand for fresh and frozen food has also been increasing. Foods such as grain, meat, fruits, vegetables, and seafood must be stored in a suitable low-temperature environment. At the same time, the storage of medicines, biological and chemical substances need a precise constant temperature and humidity environment, so it is of great significance to develop an efficient cold storage refrigeration system [1, 2]. The cold storage system is divided into a refrigeration system and a freezing system according to the difference in the temperature in the warehouse. The freezing system can provide a sub-zero temperature environment. When the system is operating under low temperature conditions, the refrigeration performance of the refrigeration system will be greatly reduced due to the large temperature difference between the outside and the inside and the low temperature inside the warehouse. At the same time, excessive exhaust temperature will often cause the compressor to turn on high-temperature protection, resulting in the system's unsafe operation [3, 4].

In response to these problems, much research was conducted [5-10], He and Li [11] suggested a neural network computation for thermal problems. In this paper, using R404A as the refrigerant, a low pressure air supply refrigeration system is designed based on a micro-channel parallel-flow heat exchanger. The refrigeration performance and energy efficiency are experimentally studied for optimization of refrigeration systems in cold storage.

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Experimental system

System principle

The design of the freezing system is based on the principle of quasi-two-stage compression refrigeration. The quasi-two-stage compression is to add a supplementary air circuit to the compressor to solve the problems of large pressure ratio and high exhaust temperature encountered in the single-stage compression. That is a supplementary circuit system is added to the quasi-two-stage compression freezing system. The refrigerant, after condensation and heat exchange, is separated from the main circuit electronic expansion valve, and a small part of the refrigerant enters the intermediate heat exchanger through the supplementary circuit electronic expansion valve. Heat exchange with the refrigerant flowing out of the condenser can increase the sub-cooling degree of the main circuit refrigerant. The supplementary circuit refrigerant is finally mixed with the main circuit refrigerant at the compressor suction port, reducing the compressor discharge temperature. The air supplement refrigeration cycle of this cold storage refrigeration system and the distribution of measuring points are shown in fig. 1.

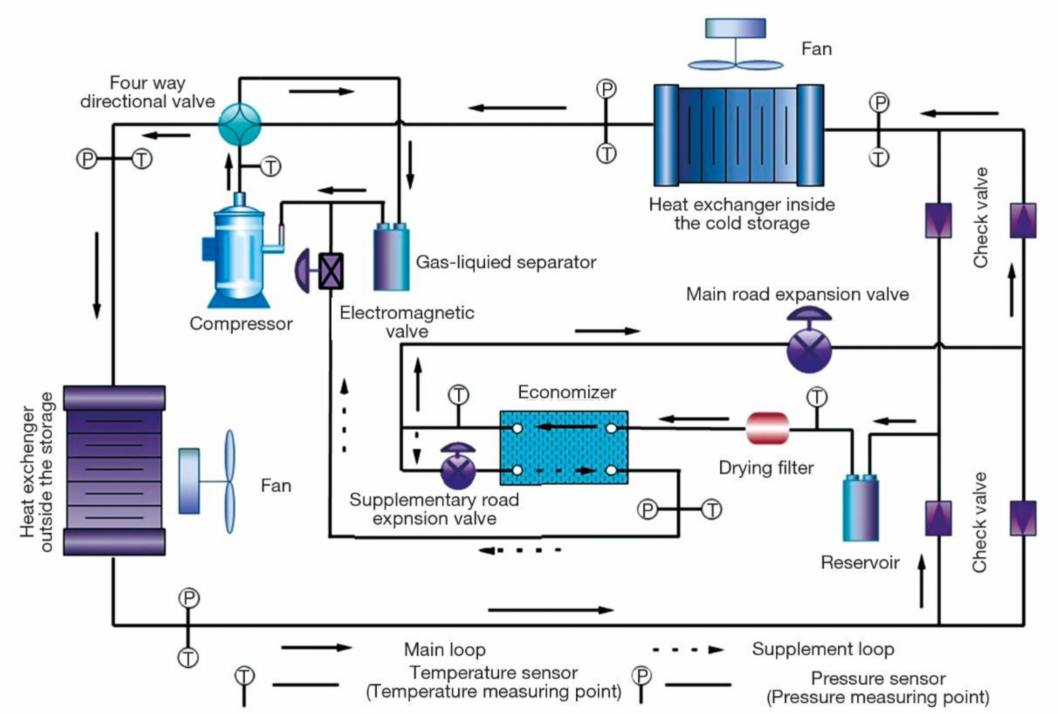

Figure 1. System flow and distribution diagram of measuring points

The principle of low-pressure air supply circulation

Figure 2 is the pressure enthalpy diagram of the low pressure air supply theoretical cycle. From the figure, it can be seen that 1→2 is the process of refrigerant being compressed into high temperature and high pressure gas in the compressor, and the state point 2 represents the state of the refrigerant at the compressor discharge port. The 2→3 means that the refrigerant changes from a high temperature and high pressure gaseous state to a low temperature and high pressure liquid state in the micro-channel parallel-flow heat exchanger outside the library to the state point 3. The 7→8 is the process of throttling and pressure reduction of the main circuit refrigerant by the main circuit electronic expansion valve. The 7→4 is the process of the supplementary circuit electronic expansion valve throttling and depressurizing the supplementary refrigerant, the main circuit refrigerant and the supplementary refrigerant are respectively throttled and depressurized to state point 8 and state point 4. The 3→7 means that the process of refrigerant heat exchange with the refrigerant in the supplementary circuit in the economizer, the refrigerant is sub-cooled to state point 7. At this time, the heat exchange process of the refrigerant in the supplementary circuit is 4→5. The refrigerant in the main circuit loop is sub-cooled by the economizer and then enters the micro-channel parallel-flow heat exchanger in the library and goes through the process 8→9 to the state point 9. The main loop refrigerant and the supplement loop refrigerant are mixed at the suction port to state point 1 and then enter the compressor for the next refrigeration cycle.

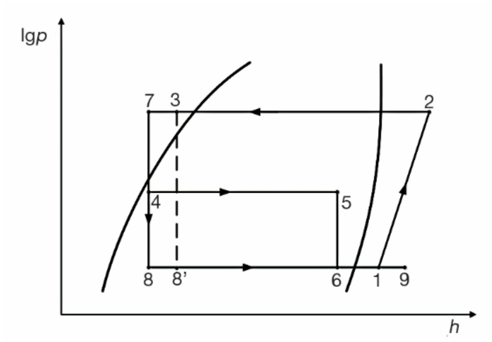

Figure 2. Low-pressure air supply theoretical cycle pressure enthalpy diagram

Experimental set-up

This experiment was carried out in the enthalpy difference laboratory of Zhongyuan University of Technology University Science Park. Combined with the system, this experiment uses R404A scroll compressors, parallel flow heat exchangers, economizers, and other equipment for cold storage. The main equipment parameters are shown in tab. 1.

Table 1. Test main equipment and main parameters

Equipment name	Specification model	Manufacturer
Compressor	Scroll type: 3CC171SA0M, applicable refrigerant: R404A, Maximum cooling capacity: 29.9 kW, displacement: 171.2 cm^3/rec	Matsushita
Heat exchanger inside the cold storage	Parallel flow type: 1200×586×36 mm, flat tube size: 1300×36×3 mm, quantity: 1	Self-made
Heat exchanger outside the cold storage	Parallel flow type: 1350×564×36 mm, flat tube size: 1300×36×3 mm, quantity: 1	Self-made
Main road expansion valve	Electronic expansion valve: E2V-24, maximum cooling capacity 21.7 kW, Driver EVD evolution, adjust the opening degree 10%~100%	Carel
Supplementary road expansion valve	Electronic expansion valve: E2V-14, maximum cooling capacity 7.7 kW, Driver EVD evolution, adjust the opening degree 10%~100%	Carel
Economizer	Flat plate type: B3-014-20D-3.0, design temperature –160 ℃~+200 ℃, Design capacity 10 kW, design pressure 3 Mpa	Wei Yi

Experimental data processing

The temperature and pressure of each measuring point of the system can be measured and automatically recorded by thermocouples and pressure sensors. There is an air volume measurement box inside the library, which can measure the air volume and temperature of the evaporator. The relevant data is processed as follows:

Total cooling capacity:

$$Q_{\mathrm{c}} = \frac{q_{mi}(h_{\mathrm{a1}} - h_{\mathrm{a2}})}{V_n'(1+W_n)} \tag{1}$$

where Q_{c} [W] is the system cooling capacity, q_{mi} [$\mathrm{m^3s^{-1}}$] – the test air volume, h_{a1} [$\mathrm{Jkg^{-1}}$] – the test return air enthalpy value, h_{a2} [$\mathrm{Jkg^{-1}}$] – the test enthalpy of the supply air, V'_n – [$\mathrm{m^3kg^{-1}}$] – the humid air specific volume, and W_n – [$\mathrm{kgkg^{-1}}$] – the moisture content of the air entering the nozzle.

Compressor motor power consumption:

$$P_{\mathrm{el}} = UI\eta_{\mathrm{el}} \tag{2}$$

where P_{el} [W] is the compressor motor power consumption, u [V] – the compressor voltage, I [A] – the compressor current, and η_{el} – the electrical efficiency.

Coefficient of refrigeration performance:

$$COP_C = \frac{Q_{\mathrm{c}}}{\eta_{\mathrm{el}}} \tag{3}$$

Analysis of experimental results

According to GB-T30134-2013 *Cold Storage Management Code*, *Cold Storage Design Code 2010*, QBT 4681-2014 *Micro-Channel Heat Exchanger for Room Air Conditioner*, *etc*., the experimental conditions are formulated. In the experiment, the compressor speed is set to 4000 rpm, and the cold storage standard experimental conditions are used inside and outside the warehouse. The ambient temperature outside the warehouse is set to 16 °C, 25 °C, 32 °C, and 40 °C, and the ambient temperature inside the warehouse is set to 10 °C, the condensing air volume is set to 65%, the evaporating air volume is set to 100%, the main circuit electronic expansion valve is set to 5 K, and the supplementary circuit electronic expansion valve is set to 30 K.

The change of compressor discharge temperature with the ambient temperature outside the warehouse is shown in fig. 3. It can be seen from fig. 3 that as the temperature outside the warehouse continues to rise, the compressor discharge temperature continues to rise. Under the same conditions, the exhaust temperature of the air supplement system is lower than that of the non-air supplement system. When the temperature outside the warehouse rises from 16 °C to 40 °C, the exhaust temperature of the air supplement system drops by 6.7-11.7%, respectively, compared with the non-air supplement system. The low pressure air supply system has a significant effect on reducing the compressor discharge temperature. The temperature outside the warehouse reaches 40 °C, and the exhaust temperature without gas supplement is as high as 96.4 °C, and the discharge temperature exceeds the normal operation of the compressor of temperature range, but the exhaust temperature of the low pressure air supply system is only 88.5 °C. Therefore, the low pressure air supply system has an advantage in the operation of the compressor under the high temperature environment outside the warehouse. The main reason is that the supplementary circuit refrigerant and the main circuit overheated refrigerant are mixed at the compressor suction port, which reduces the enthalpy of the refrigerant in the compressor and the discharge temperature.

The change of the system cooling capacity with the ambient temperature outside the warehouse is shown in fig. 4. It can be seen from fig. 4 that the cooling capacity decreases with the increase of the ambient temperature outside the warehouse. Under the same conditions, the cooling capacity of the low pressure air supply system is increased by 7.1% to 18.3% compared with that of no supplemental air. The main reason is that when a low pressure air supply is used, the enthalpy difference of the refrigerant in the main evaporator increases, and the cooling capacity increases.

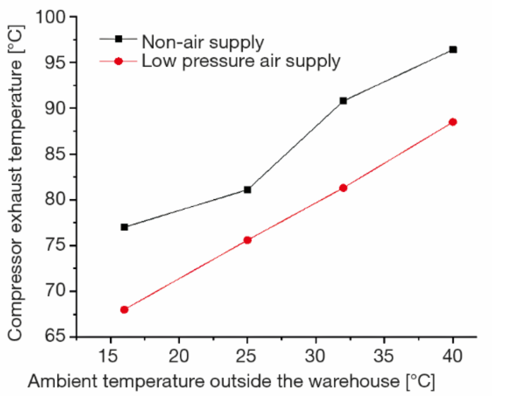

Figure 3. The compressor discharge temperature changes with the ambient temperature outside the warehouse

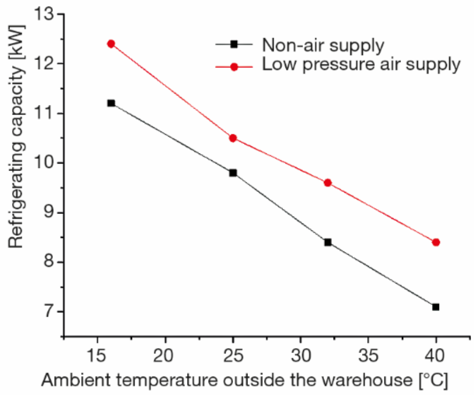

Figure 4. The system cooling capacity changes with the ambient temperature outside the warehouse

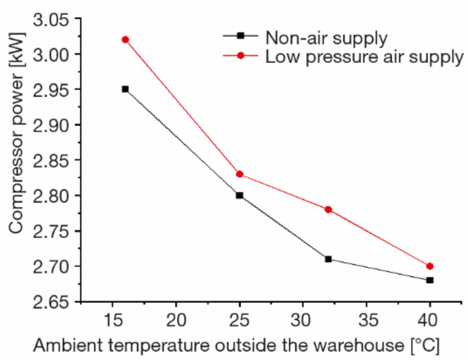

Figure 5. The compressor power changes with ambient temperature outside the warehouse

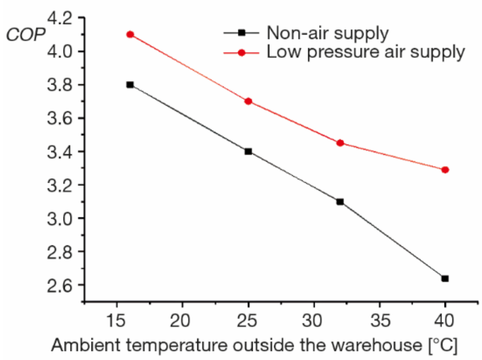

Figure 6. The system COP changes with the ambient temperature outside the warehouse

The change of compressor power with the ambient temperature outside the warehouse is shown in fig. 5. It can be seen from fig. 5 that the compressor power decreases with the increase of the ambient temperature outside the warehouse. Compared with the low pressure air supply system, the compressor power increased by 1.1-2.6%; the main reason is that the compressor suction volume remains unchanged. The increase in air supplement increases the volume of the compressor, so the compression power rises.

The change of system COP with the ambient temperature outside the warehouse is shown in fig. 6. It can be seen from fig. 6 that as the ambient temperature outside the warehouse increases, the COP decreases with the increase in temperature. Under the same conditions, a low pressure air supply is better than no supplementation. The air system COP increased by 7.1-24.6%. This is because the low pressure supplemental gas increases the enthalpy difference, increases the refrigeration capacity, and also increases the compressor power. The increase ratio of the refrigeration capacity is greater than the increased ratio of the compressor power, so the system COP is improved.

Conclusions

- Under the condition of changing the ambient temperature outside the warehouse, the refrigeration system for cold storage using a low pressure air supply system reduces the discharge temperature of the compressor by 6.7-17.7% compared with the compressor without a supplemental air system. The use of a low pressure air supply system can improve compression of stability of machine operation.
- At the same variable temperature outside the warehouse, the cooling capacity of the low-pressure air supply system increases by 7.1-18.3% compared with no supplement, the compressor power increases by 1.1-2.6%, and the system COP increases by 7.1-24.6%.

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