INFLUENCE OF COMPRESSOR SPEED ON THE PERFORMANCE OF LOW PRESSURE VAPOR-INJECTED REFRIGERATION SYSTEMS

by

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For the problems of high compressor discharge temperature and system performance decay during the operation of cold storage, a parallel flow heat exchanger based low pressure make-up gas refrigeration system experimental bench was designed and built, and the changes of refrigeration system performance were analyzed under different compressor speeds. The results show that when the compressor speed increases from 2500 rpm to 4500 rpm, the compressor discharge temperature increases, the refrigeration capacity increases by 39.53% and the compressor power increases by 38.89%, in addition, as the speed increases, the system COP shows a trend of first increasing and then decreasing, with the best value of 2.71 at 3500 rpm.

Key words: *refrigeration systems, low pressure vapor-injected, compressor speed, refrigeration performance*

Introduction

Nowadays science and technology have been developing rapidly, and we are now coming into a fast-paced century. At the same time, the demand for daily necessities such as food and drugs is also increasing, so the cold chain transportation and preservation of food and drugs are also requiring an extremely high requirement. This is why research into the optimization of the system structure, stability and energy efficiency of refrigeration system is currently the focus of society and the academic community [1-3].

The compressor is the core component of the refrigeration system and its speed has a great influence on its performance. The disadvantages in high exhaust temperature, decay performance and high energy consumption have seriously hindered the further application of the refrigeration system [4].

Yang *et al*. [5] established an experimental platform for a dual-mode refrigeration system to study the energy-saving characteristics of a multifunctional cold storage refrigeration system. The experimental study investigated the frequency conversion characteristics and control strategies of the two modes, and the results showed that the optimal COP and the optimal compressor frequency existed as the compressor frequency of the low temperature stage increased, and the optimal frequency increased as the evaporating and condensing temperatures decreased. Jin *et al*. [6] conducted an experimental study of cold storage with ammonia

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screw refrigeration compressors, the results showed that when the temperature of the cold storage is higher than –28 °C, the comprehensive performance of the cold storage with single-stage compression refrigeration system is better, while when the temperature of the cold storage is lower than –28 °C, the cold storage with two-stage compression refrigeration system is better. Li *et al*. [7] proposed a hybrid refrigeration system based on dual compressors for refrigerated transport vehicles with PV refrigeration in order to solve the problem of maintaining refrigeration after parking, and established a mathematical model for simulation, which showed that the fuel consumption of refrigerated vehicles was reduced by about 7.06% compared with that of the conventional system. A medium pressure charge gas type heat pump air conditioning system with economizer for pure electric buses was developed by Li *et al*. [8] to solve the problem of performance degradation of heat pump air conditioning system for electric vehicles in high and low temperature environments. Chen [9] introduced the make-up air enthalpy technology for rolling rotor compressors and scroll compressors, explained their advantages and disadvantages, and found that the combination of intermediate make-up air technology and rotor type compressors can solve the problems existing in conventional air source heat pumps. Qin *et al*. [10] conducted experiments on two scroll compressors with different injection holes and concluded that the performance of the compressor with three sets of make-up air holes was better than that of the compressor with a single pair of make-up air holes. Kim *et al.* [11] compares the performance characteristics of liquid, steam and two-phase injection heat pumps with R410A scroll compressors and analyses the optimum injection quality of two-phase injection heat pumps in relation to injection pressure, compressor frequency and outdoor temperature. Now the artificial neural network [2, 12-17] is a hot research direction for the cold chain transportation.

The cited literature has analyzed and researched the mode optimization of refrigeration systems, the selection of compressors and the technology of vapor-injected, but there has been less research ono the effect of compressor speed on the performance of low pressure vapor-injected refrigeration systems, and this paper examines and analyses the compressor speed to lay the foundation for optimizing the performance of existing refrigeration systems.

Principle of low pressure vapor-injected refrigeration system

The low pressure vapor-injected refrigeration system is to reduce the superheat of the refrigerant when it enters the compressor by mixing the not fully saturated refrigerant after the charge valve with the saturated refrigerant at the evaporator outlet, resulting in the temperature reduction at the compressor suction port and the compressor discharge port to make the system run safely and stably. The air supply port for low pressure vapor-injected refrigeration system is in front of the compressor. The main components of the system are the compressor, the parallel flow heat exchanger, the intercooler, the filter drier, the oil separator, the gas-liquid separator, the main electronic expansion valve, the supplementary electronic expansion valve and other necessary auxiliary devices. The high temperature and high pressure gaseous refrigerant from the compressor is separated from the lubricating oil by the oil separator and then the refrigerant is diverted into two micro-channel parallel flow heat exchangers outside the reservoir, the refrigerant is indirectly heat exchanged with the air outside the reservoir through the two micro-channel parallel flow heat exchangers outside the reservoir respectively and becomes low temperature and high pressure liquid refrigerant, then it is mixed in the two micro-channel parallel flow heat exchangers outside the reservoir and flows into the liquid storage, where the water is separated from the liquid storage. The refrigerant from the intermediate heat exchanger is divided into two ways, the refrigerant from the intermediate heat exchanger flows into the main electronic expansion valve and becomes low temperature and low pressure liquid refrigerant after throttling and pressure reduction, the low temperature and low pressure liquid refrigerant enters the parallel flow heat exchanger of the micro-channel in the store and becomes low temperature and low pressure gaseous refrigerant. The liquid refrigerant is separated from the liquid refrigerant by the gas-liquid separator and then enters the compressor. The other way flows into the supplementary electronic expansion valve and then returns to the intermediate heat exchanger with the undivided refrigerant to become low temperature low pressure gaseous refrigerant, and then mixes with the refrigerant flowing from the main road and enters the compressor, completing a complete refrigeration cycle.

As shown in figs. 1 and 2, the principle of the low pressure vapor-injected cycle is: the refrigerant compressed by the compressor into a high temperature and high pressure state is discharged from the discharge port (state 2), then enters the condenser for condensation and exotherm (state 2 to state 3), and the cooled refrigerant is divided into the main circuit and the auxiliary circuit, the circulating mass of the main circuit passes through the economizer and the auxiliary circuit, the auxiliary circuit absorbs heat to make the main circuit subcooled to state 7, the subcooled state. The circulating mass then passes through the main circuit expansion valve throttling down to state 8, then evaporates in the evaporator (state 8 to state 9). At the same time, the circulating mass in the auxiliary circuit enters the auxiliary circuit expansion valve throttling (state 7 to state 4), then passes through the economizer for evaporation (state 4 to state 5), the auxiliary circuit circulating mass enters the low pressure charge port (state 5 to state 6) after throttling by the throttling valve, then mixes with the main circuit circulating mass to state 1. Mixing to state 1, the main and auxiliary circulating masses are mixed and compressed by the compressor and then discharged from the exhaust port.

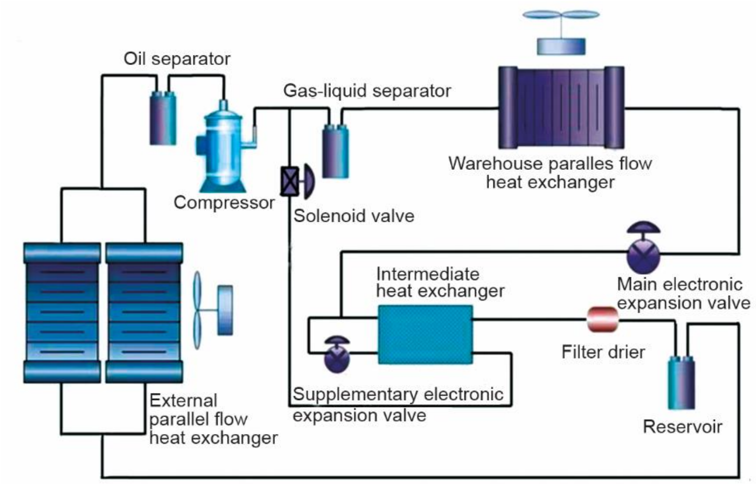

Figure 1. Schematic diagram of the low pressure vapor-injected refrigeration system

The formula for calculating the low pressure vapor-injected refrigeration system is as follows.

– Refrigeration system evaporator cooling capacity:

$$Q_C = \dot{m}_c (h_9 - h_8)$$

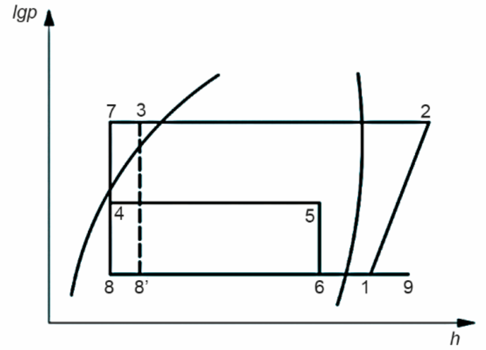

Figure 2. Theoretical cycle pressure enthalpy diagram for low pressure vapor-injected

where Q_C [kW] is the cooling capacity, $\dot{m}_r$ [kgs^{-1}] – the refrigerant mass-flow rate at the evaporator inlet, h_9 [kJkg^{-1}] – the enthalpy of refrigerant at the outlet of the evaporator, and h_8 [kJkg^{-1}] – the enthalpy of refrigerant inlet to the evaporator.

– Compressor power:

$$W = \dot{m}_r (h_2 - h_1)$$

where W [kW] is the compressor power, $\dot{m}_r$ [kgs^{-1}] – the refrigerant mass-flow rate at the compressor discharge port, and h_1 [kJkg^{-1}] – the enthalpy of refrigerant at the suction port of the compressor.

– Refrigeration system cooling factor:

$$\mathrm{COP_c} = \frac{Q_c}{W}$$

– Vapor-injected refrigerant mass:

$$\dot{m}' = \dot{m}_r - \dot{m}_c$$

Intermediate heat exchanger heat exchange:

$$Q' = \dot{m}_r (h_3 - h_7)$$

where Q' [kW] is the heat exchange of the intermediate heat exchanger, h_3 [kJkg^{-1}] – the inlet refrigerant enthalpy of the intermediate heat exchanger, and h_7 [kJkg^{-1}] – the enthalpy of refrigerant at the outlet of the intermediate heat exchanger.

Experimental equipment

The piston compressor GEAHCX34e/380-4S produced by Blog is used for this real, the heat exchanger inside the store is made by Colin with the number of flat tube rows of 52. The heat exchanger outside the store is made by Colin with the number of flat tube rows of 52. The intermediate heat exchanger is made by Jiangsu Weiyi with the model number B3-014-20D-3.0, the electronic expansion valve is made by E^2 V-24 produced by Carle from Italy. The pressure sensor is made by Emerson from the USA with the model numbers PT5-30M/T and PT5-07M/T. The pressure transducers are PT5-30M/T and PT5-07M/T made by Emerson, USA.

This experimental program according to the *cold storage design code*, GB/T25129-2010 *air cooler for refrigeration*, JB/T11967-2014 *refrigeration and air conditioning equipment condenser with micro-channel heat exchanger*, and other national standards under the specification. The ambient temperature inside the store is set at –2 °C and 3 °C, the ambient temperature outside the store is 32 °C, the superheat degree of the main valve is 6 K, the superheat degree of the supplementary valve is 25 K, the speed of the compressor is 2500 rpm, 3000 rpm, 3500 rpm, and 4000 rpm.

Table 1. Experimental working conditions of the refrigeration system

Test conditions	Ambient drybulb temperaturein the warehouse [℃]	Ambient dry bulb/wet bulb temperature outside the store [℃]	Main valve overheating degree [K]	Overheating of the fill valve [K]	Air replenishment mode
	–2	32/24	6	25	Low pressure make-up air

Analysis of experimental results

Figure 3 shows the effect of compressor speed on the compressor discharge temperature. As can be seen from the figure, the discharge temperature of the compressor increases with its speed, from 68.5 °C to 79.2 °C when the compressor speed is increased from 2500 rpm to 4500 rpm in vapor injected mode, an increase of 5%. When the compressor speed is increased from 2500 rpm to 4000 rpm in non-vapor injected mode, the discharge temperature increases from 86.8 ℃ to 94.1 ℃, an increase of 10.7%. The increase in compressor speed increases the work done by the compressor on the refrigerant, the discharge pressure increases and the discharge temperature increases.

Figure 4 shows the effect of the change in compressor speed on the system cooling capacity. As the compressor speed increases, the system cooling capacity gradually increases. This is because as the compressor speed increases, the suction and discharge capacity of the compressor increases, resulting in an increase in the system cooling capacity per unit time and a significant increase. When the compressor speed is increased from 2500 rpm to 4500 rpm in vapor injected mode, the cooling capacity increases from 12.97 kW to 21.45 kW, an increase of 39.53%%. The compressor speed has a large impact on the refrigeration capacity of the low pressure vapor-injected system.

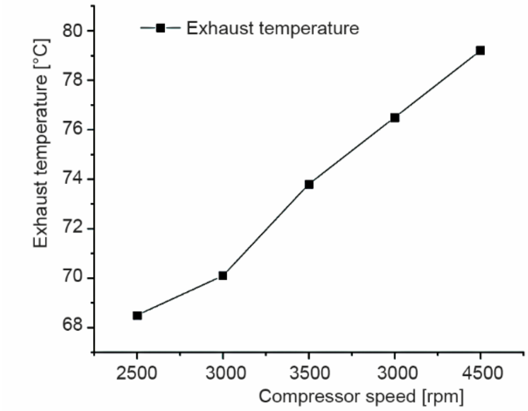

Figure 3. Effect of compressor speed on compressor discharge temperature

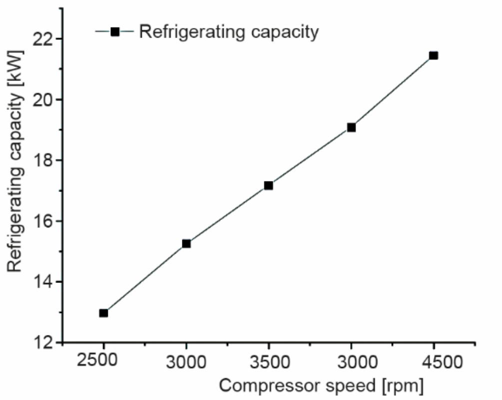

Figure 4. Effect of compressor speed on system cooling capacity

Figure 5 shows the effect of the change in compressor speed on the compressor power for the same previous operating conditions. As can be seen from the graph, the compressor power increases with increasing speed. This is due to the increase in compressor speed, which also increases the compressor discharge mass-flow rate. When the compressor speed is increased from 2500 rpm to 4500 rpm in the charge mode, the compressor power increases from 5.39 kW to 8.82 kW, an increase of 38.89%, indicating that the vapor injected system has little effect on the power of the compressor. The main reason is that although the vapor-injected system will be a small part of the refrigerant out, but in the entry before the main road and complementary road for convergence, although the refrigerant superheat at the import is reduced, and the total amount of refrigerant into the compressor remains unchanged, so the compressor power change is not significant.

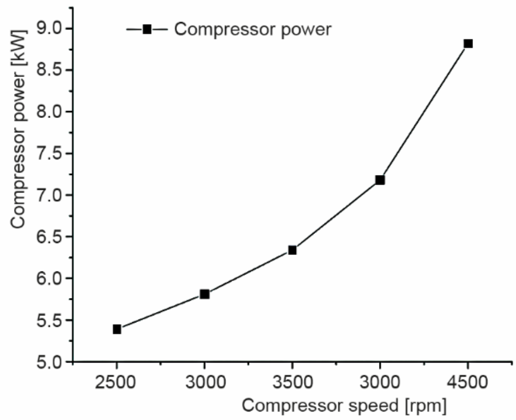

Figure 5. Effect of different speeds on compressor power

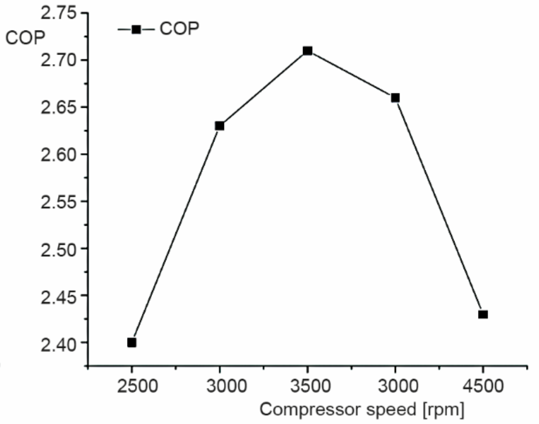

Figure 6. Effect of compressor speed on system COP

Figure 6 shows the impact on COP of the system with the change of compressor speed. It can be seen from fig. 4 that, with the increase of compressor speed, COP in the air supply system first increases and then decreases with the increase of compressor speed. When COP is at 2500 rpm, 3000 rpm, 3500 rpm, 4000 rpm, and 4500 rpm, COP vapor injected system is 2.4, 2.63, 2.71, 2.66, and 2.43, respectively, compared with the non-vapor injected system. It can be concluded that under the condition of vapor injected mode, the change with the compressor speed is more obvious, and the COP value is the maximum when the speed is 3500 rpm.

Conclusions

In order to investigate the effect of compressor speed on the refrigeration performance of a refrigerated system, an experimental bench was built for a refrigerated system based on a parallel flow heat exchanger with a low pressure vapor-injected mode. The effect of compressor speed in the vapor-injected system on compressor discharge temperature, system cooling capacity, compressor power and system COP was investigated and the following conclusions were drawn.

- Under different compressor speed, the change of refrigeration system cooling performance was analyzed. The results show that: when the compressor speed increases from 2500 rpm to 4500 rpm, the compressor discharge temperature increases, the refrigeration capacity increases by 39.53% and the compressor power increases by 38.89%.

- As the compressor speed increases the COP increases with the compressor speed and then decreases. Under the vapor-injected condition, the change with the compressor speed is more obvious at 3500 rpm when the maximum COP value is 2.71.

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