A Investigation of the Performance of Two-Stage Compression Refrigeration/Heat pump System with Dual-cylinder Rolling Piston Compressor

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Abstract. A thermodynamically analytical model on the two-stage compression refrigeration/heat pump system with vapor injection was derived. The optimal volume ratio of the high-pressure cylinder to the low-pressure one has been discussed under both cooling and heating conditions. Based on the above research, the prototype was developed and its experimental setup established. A comprehensive experiment for the prototype has been conducted, and the results show that, the favorable value of the volume ratio is between 0.70 and 0.85. The value of the relative intermediate pressure between 1.1 and 1.2 is suitable for the two-stage system; Comparing with the single-stage cycle, the discharge temperature decreases about 2.3°C～4.8°C under cooling conditions and 4.1°C～9.0°C under heating conditions.

Key Words: two-stage compression, heat pump, vapor injection, rolling piston compressor

Introduction

According to the “Montreal Protocol on Substances that Deplete the Ozone Layer”, the developing countries will freeze the production of hydrochlorofluorocarbons (HCFCs) in 2030. This put forward a new challenge for China, as a main producer of HCFCs. The production of air-conditioner in China ranks first in the world, and more than 90% of the household air-conditioner used R22 as refrigerant, so it is urgent to replace refrigerant R22 in China (Zeng 2007). Alternative refrigerants for HCFCs can be generally divided into three categories: the first one is hydrofluorocarbons (HFCs) refrigerants, the second is natural working substance, and the third is hydrocarbons (HC) refrigerants (Wang and Chen 2008). Among the alternative refrigerants for R22, R410A is a favorable one used in household air-conditioner. And its advantages are that it has excellent heat transfer performance and flow characteristics, and the refrigeration capacity is about 1.4 times higher than that of R22 under the condition of the same displacement of the compressor (Chuan 2007). However, its disadvantages are that the pressure is about 1.6 times higher than that of R22, which needs higher sealing performance in the compression chamber (Li and Zhong 2003). The development of the dual-cylinder rolling piston compressor is a good way to solve this problem, and also improved its efficiency of the compressor (Jian et al 2011).

The energy consumption of heating and air-conditioner for home in China accounts for about 14.85% of the total national energy consumption. In order to make air-conditioner more efficiency and more saving energy, the National Standard, GB12021.3-2010 named the minimum allowable value of the energy efficiency and energy efficiency grades for room air conditioners, has been implemented in China, which will greatly improved the efficiency of air-conditioner. The refrigeration system with economizer has been widely used in air-conditioner as it can be significantly improved the coefficient of performance. There are two types of economizer, sub-cooler and flash-tank. Compared with the sub-cooler system, there are three reasons to explain that the refrigerating system with flash tank as economizer can effectively improve the performance and COP. The experimental investigations had verified the heating capacity of the flash tank system is higher than that of the sub-cooler system (Ma and Zhao 2008, Wang et al 2008). And the flash tank system owns relatively low costs and simple component, and very approaches to the two-stage compression cycle so that the vapor entering the auxiliary inlets of the compressor is more close to the saturated state

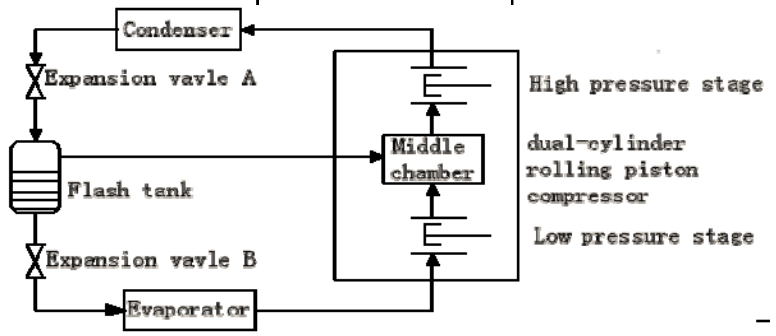
(Zhao et al 2006). The heat pump with economizer coupled with scroll compressor is considered to be the most economical and effective improvements against the insufficient heat capacity and poor reliability when running in cold regions (Martin 2002). The flash tank system coupled with scroll compressor, with favorable heating performance, was considered to be a suitable thermodynamic cycle for small-sized air source heat pump in cold regions. Scroll compressor is mainly used in the air-conditioner with higher capacity ranging from 7.5kW to 15kW, but rolling piston compressor, with relatively low cost and better performance, usually occupied in small size air-conditioner below 5kW. The two-stage rolling piston compressor in CO2 trans-critical cycle was developed and studied in detail and the motion and forces of the compressor had been analyzed according to the designed parameters (Tian et al 2010). Zhang Qian et al. suggested that the two-stage compression cycle with dual-cylinder rolling piston compressor could effectively reduce the discharge temperature and the compression ratio, and improve the COP, when R32 is used as refrigerant (Zhang et al 2011). (Bertsch et al 2008) verified the air-source heat pump system with two-stage compression and one-grade throttling, R410A as refrigerant, can improve the performance of air-conditioner.

Although lots of researches focused on thermodynamic and dynamic performances of dual-cylinder rolling piston compressor, there are not much researches dealing with it used as a two-stage compressor with the working fluid of R410A in room air-conditioner. In this paper, the two-stage compression refrigeration/heat pump system coupled with dual-cylinder rolling piston compressor, in which a flash tank used as economizer and R410A as working fluid, would be studied in detail. By constructing the thermodynamically analytical model, the favorable volume ratio of the higher-pressure chamber to the lower pressure one (VRHL) of the dual-cylinder compressor would be discussed. The prototype was developed based the calculated results, and researched experimentally.

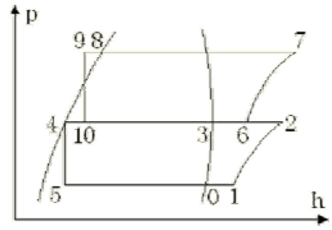
1 SYSTEM AND ANALYTICAL MODEL

The schematic chart and pressure-enthalpy diagram of the two-stage compression refrigeration/heat pump are shown in Figure 1. It mainly consist of the dual-cylinder compressor, condenser, expansion valve A (i.e. first stage throttling), flash tank, expansion valve B (i.e. second stage throttling) and evaporator. The middle chamber between the high pressure chamber and low pressure one is a mixing cave in which the discharge gas from the low pressure chamber is mixed with the saturated gas from the flash tank and cooled down. The superheated gas exhausted from the compressor flows into the condenser, in which it is cooled down and into sub-cooled liquid by rejecting heat to the cooling medium flowing through the condenser (i.e. process 7-9 in Figure 1b). And the liquid refrigerant flowing through expansion valve A is throttled into the two-phase mixture (process 9-10). The mixture enters into the flash tank and separates then into the saturated gas in the top volume (state 3) and the liquid in the bottom (state 4). The saturated gas is sucked into the middle chamber of the compressor and mixed with the discharge gas from the low pressure chamber, and eventually enters into the high pressure chamber. The liquid in flash tank becomes sub-cooled as the gas flashes continually, The liquid from the flash tank is throttled second when it flows through expansion valve B (process 4-5), and then enters into the evaporator. It becomes superheated gas when going through the evaporator and then sucked into the low-pressure chamber for first-stage compression (process 1-2). The discharge gas mixed with the saturated gas from the flash tank in the middle chamber (state2+state3=state6) and then sucked into the high-pressure chamber for the second-stage compression (process 6-7). The gas discharged from the compressor flows repeatedly into condenser and its working cycle has completed.

As the gas separated continuously out in flash tank, this lead to a higher sub-cooling degree of the liquid in the bottom, and the unit refrigerating capacity of the heat pump cycle increase in turn so that the heat exchanging process is improved. At the same time, the temperature of the saturated gas from the flash tank is lower than that of the discharged gas from the low-pressure chamber, which is helpful to improve the second-stage compression process and reduce the exhaust temperature of the compressor.



（a） Schematic chart



(b) Pressure-enthalpy diagram

Figure 1 Two-stage compression refrigeration/heat pump system

The main performances of the system can be obtained by analyzing the thermodynamic cycle shown in Figure 1 (b) according to the energy conservation law and the mass conservation law.

The unit refrigerating capacity

$$q_0=h_1-h_4 \tag{1}$$

The unit heating capacity

$$q_k= h_7-h_9 \tag{2}$$

The compression work for the low-pressure chamber

$$w_L =h_2-h_1 \tag{3}$$

The compression work for the high-pressure chamber

$$w_H =h_7-h_6 \tag{4}$$

The cooling coefficient of performance

$$\mathrm{COP_r}=\frac{Q_0}{W}=\frac{q_{mL}\cdot q_0}{q_{mL}\cdot w_L+q_{mH}\cdot w_H}=\frac{h_1-h_4}{\dfrac{h_2-h_1}{\eta_{\mathrm{iL}}\eta_{mo}\eta_m\eta_{VL}}+\dfrac{h_2-h_4}{h_6-h_9}\times\dfrac{h_7-h_6}{\eta_{\mathrm{iH}}\eta_{mo}\eta_m\eta_{VH}}} \tag{5}$$

The heating coefficient of performance

$$\mathrm{COP_h}= \mathrm{COP_r}+1 \tag{6}$$

VRHL:

$$\zeta=\frac{V_{h,H}}{V_{h,L}}\times\frac{n_L}{n_H}=\frac{q_{mH}v_{Hin}/\eta_{VH}}{q_{mL}v_{Lin}/\eta_{VL}}\times\frac{n_L}{n_H}=\frac{n_L\eta_{VL}}{n_H\eta_{VH}}\times\frac{(h_2-h_4)v_6}{(h_6-h_9)v_1} \tag{7}$$

Indicated efficiency is calculated by the follow formula (Guo Q.T, and Wu J.F. 1994),

$$\eta_i=1-0.6[1-(p_2/p_1)^{-0.3}] \tag{8}$$

Where, p_2 stands for discharge pressure, and p_1 stands for suction pressure. The electrical efficiency

$$\eta_{el}=\eta_i\eta_{mo}\eta_m \tag{9}$$

for the compressor takes value of 0.55, and the volumetric efficiency takes value of 0.9 (Ma et al 2003). It is assumed that the volumetric efficiency for the low-pressure chamber is equal to the one for the high-pressure chamber. In addition, the same rotation speed for the low-pressure chamber and for the high-pressure chamber is considered as their rolling pistons are driven by one shaft.

The cooling operation conditions for the calculations were assumed that the evaporation temperature keeps at 5°C while the condensation temperatures are 35°C, 40°C, 45°C, respectively. The heating ones were that the condensation temperature keeps at 40℃ while the evaporation temperatures are -5°C and -10°C, respectively. The superheating degree keeps at 10°C and the sub-cooling degree 5℃ for all of the above conditions. The variations of COP_r and COP_h with VRHL are simulated according to the formula (1)～(7).

The varying curves of COP_r and COP_h with VRHL for the two-stage cycle coupled with dual-cylinder rolling piston compressor are shown in Figure 2. There are two numbers following the symbol of COP_r or COP_h in the curve denotations, which represent the operation conditions. The first number stands for the evaporating temperature, and the second number stands for the condensing temperature. For example, the denotation, COP_r 5～35, expresses the COP_r curve when the evaporating temperature is 5°C and the condensing temperature is 35°C. From Figure 2, it can be seen that COP_r rises and then decreases with increase of VRHL, and approaches its maximum value when VRHL is between 0.70 and 0.85. And COP_h keeps nearly constant with increase of VRHL. Therefore, considering the cooling and heating performance of the system in balance, the favorable VRHL is between 0.70 and 0.85. And the VRHL value of 0.80 has selected for the developed prototype.

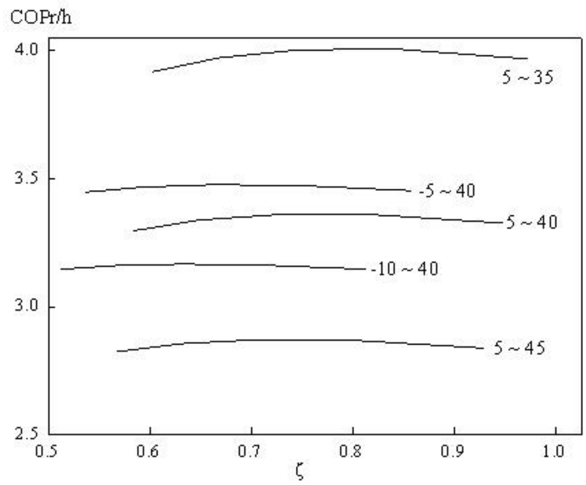


Figure 2 The variation of COP with VRHL

2 EXPERIMENTAL SET UP

The testing system, as shown in Figure 3, is in accordance with Chinese National Standards, GB/T 15765—2006 (Hermetic motor-compressors for room air conditioners) and GB/T 5773—2004 (The method of performance test for positive displacement refrigerant compressors). As mentioned above, the VRHL of the dual-cylinder rolling piston compressor for the experiments is 0.80. The testing conditions is reached by adjusting the temperature of chilled water to control the evaporation temperature, the temperature of cooling water to control the condensation temperature and the opening of the manually-driven expansion valves to control the suction pressure and intermediate pressure, respectively. The shut-off valve in the middle path between the flash tank and the mixing chamber is used to switch on/off the vapor injection into the mixing chamber. If it is switched off, the testing system for the prototype becomes a single-stage compression cycle from a two-stage compression cycle with vapor injection. In the measurements, the measured pressures and temperatures are transmitted to the data acquisition instrument by the pressure transducer and thermal resistance of PT100, respectively. The flow rate of the chilled water and cooling water are measured by the turbine flow meters. The main specifications of the sensors and instruments are shown in Table 1 and the main specifications of the compressor are listed in Table 2.

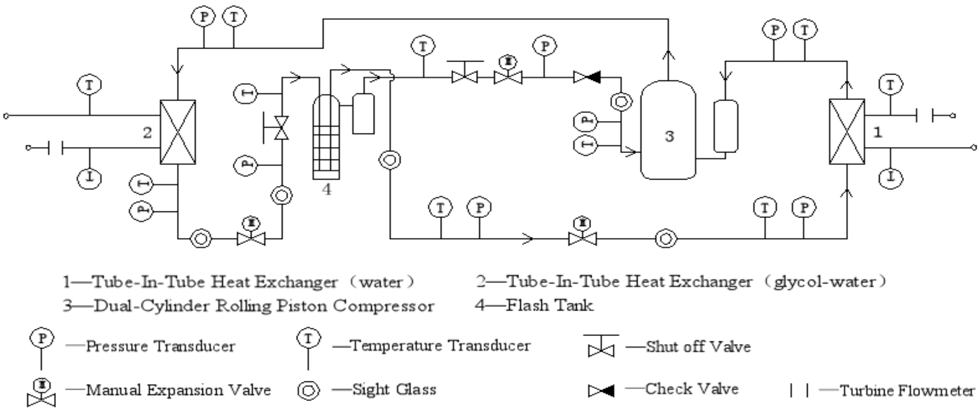


Figure 3 Measuring system for the prototype

Table 1 Main specifications of the sensors and the testing equipments

Sensor	Accuracy	Full Scale	Model
Temperature	±0.2 □	-	Pt100
Pressure transducer	±0.2% of full scale	2.5/4.5 MPa	Huba
Flowrate meter	±0.2% of full scale	2.5 kg·s-1	LZB-50
Ampere meter	±0.2% of full scale	-	DZFC-1
Watt meter	±0.2% of full scale	10 kW	DZFC-1
Data logger	±0.2% of full scale	-	HP34970A

Table 2 Specific performance and structural parameters of compressors.

Parameters	Low compressor	High compressor
Height of the compressor	31.25 cm	24.32 cm
Diameter of the compressor	12.96 cm	11.5 cm
Height of the cylinder	2.2 cm	1.6 cm
Diameter of the cylinder	5.4 cm	4.2 cm
Diameter of the piston	4.73 cm	3.32 cm
Width of the vane	0.47 cm	0.32 cm
Suction angle	0.416 rad	0.698 rad
Exhaust angle	5.917 rad	5.661 rad
Stroke volume	11.73 cm^3	8.896 cm^3

The prototype was tested according to the experimental conditions shown in Table 3. The parameters of the conditions mainly include the evaporating temperature or pressure, condensing temperature or pressure, superheating degree, sub-cooling degree and intermediate pressure etc. Firstly, the single-stage cycle was experimented when the shut-off valve in the middle path was closed. After that, the shut-off valve was fully opened for testing the two-stage cycle while the expansion valve in the middle path was regulated to control the intermediate pressure. After the system operating steadily, the data were recorded by data acquisition instrument, and it was much cared that the discharge temperature exceeded the limit of 130°C or not.

In Table 3, $pm = \sqrt{p_k \cdot p_0}$, which expresses the geometric mean value of p_o, the evaporating pressure, and p_k, the condensing pressure. The intermediate pressure was set to 1.0pm, 1.1pm, 1.2pm and 1.3pm, respectively, in the experiments. As known, the discharge pressure of the low-pressure chamber will keep at certain value when the VRHL value of a dual-cylinder compressor is given. The vapor injecting process will work when the intermediate pressure is higher than the discharge pressure of the low-pressure chamber. In the experiments for the single-stage cycle, the pressure in the middle path after the shut-off valve can be approximately considered as the discharge pressure. It is found in the experiments that vapor injection exists when the intermediate pressure is equal to or higher than pm, which is slightly greater than the discharge pressure of the low-pressure chamber.

Table 3 Experimental conditions and corresponding intermediate pressure

Operation mode	Evaporating Temperature (□)	Condensing Temperature(□)	Superheating Degree (□)	Sub-cooling Degree (□)	Intermediate Pressure(bar)			
					pm	1.1pm	1.2pm	1.3pm
Cooling	5	35			14.14	15.5	17.0	18.5
		40			15	16.5	18.0	19.5
		45	10	5	16	17.5	19.0	20.5
Heating	-5	40			12.8	14.1	15.39	16.67
	-10	40			11.78	12.96	14.14	15.32

3 RESULTS AND DISCUSSIONS

The experimental results for the prototype were rearranged as the curves shown in Figure 4 ～Figure 9. For cooling conditions, the COP_r, cooling capacity and discharge temperature for the two-stage system coupled with dual-cylinder rolling piston compressor varying with the relative intermediate pressure, which is the ratio of the intermediate pressure to pm, are shown in Figure 4～Figure 6. The first number in the curve denotation means the evaporating temperature and the second number means the condensing temperature. For example, the denotation, 5-35, means the curve was obtained with evaporating temperature of 5℃ and condensing temperature of 35°C.

The variation of COP_r with the relative intermediate pressure is shown in Figure 4. It can be seen that COP_r increases firstly when the relative intermediate pressure rises, and COP_r gets the maximum value when it is around 1.1. When the relative intermediate pressure exceeding 1.1, COP_r decreases with increase of the relative intermediate pressure. COP_r goes up with decrease of the condensing temperature when the evaporating temperature keeps constant. COP_r gets the value of 3.81 when the evaporating temperature is 5°C and the condensing temperature 35°C.

The variation of cooling capacity with the relative intermediate pressure is shown in Figure 5. The varying trend of the cooling capacity is similar to that of COP_r. This because that the power input to the compressor rises slightly with increase of the relative intermediate pressure when the vapor injection works. And its maximum value appears when the relative intermediate pressure is around 1.1. The cooling capacity is 4.8kW when the evaporating temperature is 5°C and the condensing temperature 35°C.

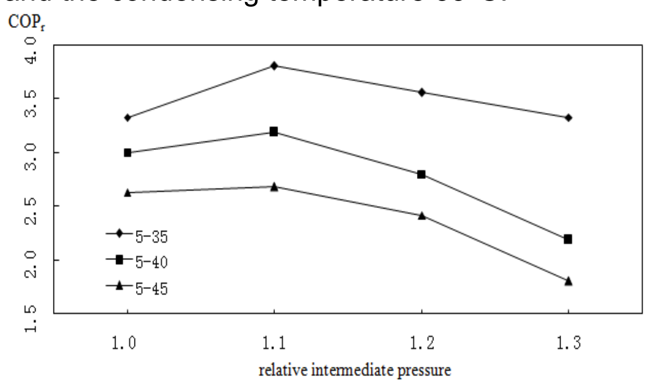


Figure 4 The variation of COP_r with relative intermediate pressure

The variation of the discharge temperature with relative intermediate pressure is shown in Figure 6. The discharge temperature decreases significantly with increase of relative intermediate pressure. In the experimental conditions, the discharge temperature decreases about 2.3°C～4.8°C whenever the relative intermediate pressure increases 0.1. Comparing with the single-stage cycle, the discharge temperature decreases within 1°C when the relative intermediate pressure is 1.0. When the evaporating temperature is 5°C and the condensing temperature is 45°C, the discharge temperature can reach 89.0°C at the relative intermediate pressure of 1.0 and go down to 85.6°C at the relative intermediate pressure of 1.1. The discharge temperature goes up with increase of the condensing temperature when the evaporating temperature keeps constant. In the range of the measured conditions, the discharge temperature decreases about 5.8°C～7.3°C at the same relative intermediate pressure whenever the condensing temperature rises per 5°C.

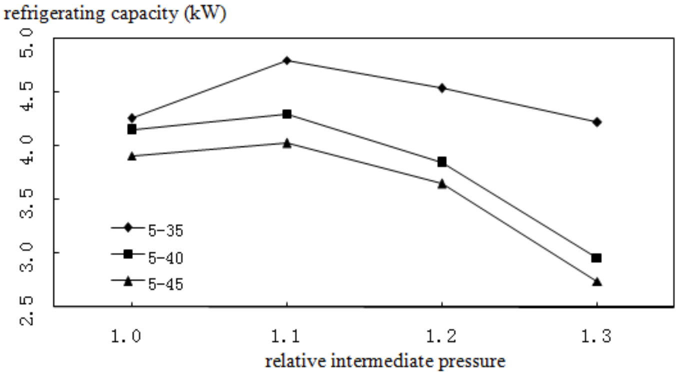


Figure 5 The variation of cooling capacity with relative intermediate pressure

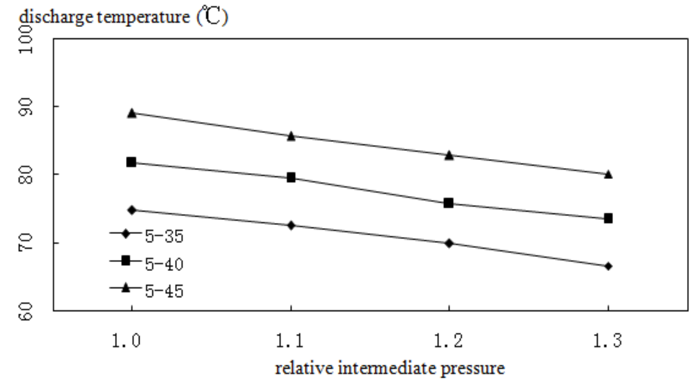


Figure 6 The variation of discharge temperature with relative intermediate pressure under refrigerating mode

For heating conditions, the COP_h, heating capacity and discharge temperature for the two-stage system varying with the relative intermediate pressure are shown in Figure 7～Figure 9.

The variation of COP_h with the relative intermediate pressure is shown in Figure 7. It can be seen that the varying trend of COP_h is similar to the one of COP_r. COP_h increases firstly when the relative intermediate pressure becomes great, and COP_h gets the maximum value when the relative intermediate pressure is around 1.1. After the relative intermediate pressure exceeding 1.1, COP_h decreases gradually with increase of the relative intermediate pressure, but the variation of COP_h is not as sharp as the one of COP_r. COP_h gets increased with evaporating temperature when the condensing temperature keeps constant. And COP_h is 3.39 when the evaporating temperature is -5°C and the condensing temperature is 40°C.

The variation of the heating capacity with relative intermediate pressure is shown in Figure 8. The varying trend of the heating capacity is similar to that of COP_h as the power input of the compressor rises slowly with increase of relative intermediate pressure. But the maximum value of the heating capacity appears when the relative intermediate pressure is around 1.2. The heating capacity is 4.76kW when the evaporating temperature is -5°C and the condensing temperature is 40°C.

The variation of the discharge temperature with relative intermediate pressure is shown in Figure 9. Similar to the cooling conditions, the discharge temperature decreases with increase of the relative intermediate pressure. And the greater the relative intermediate pressure is, the lower the discharge temperature is. In the range of the experimental conditions, the discharge temperature decrease about 4.1°C～9.0°C whenever the relative intermediate pressure increases 0.1. The discharge temperature, when the relative intermediate pressure is 1.0, approaches to the one of the single-stage cycle. The discharge temperature gets increased with reduction of evaporating temperature when the condensing temperature keeps constant. The discharge temperature decreases about 5.5°C～14.5°C whenever the evaporating temperature raises 5°C when the relative intermediate pressure keeps unchanged. This demonstrates that the vapor injection can attain more decrease of the discharge temperature under the heating conditions than that under the cooling conditions. When the evaporating temperature is -10°C and the condensing temperature is 40°C, the discharge temperature can reach 110.9°C at the relative intermediate pressure of 1.0 and go down to 104.6°C at the relative intermediate pressure of 1.1.

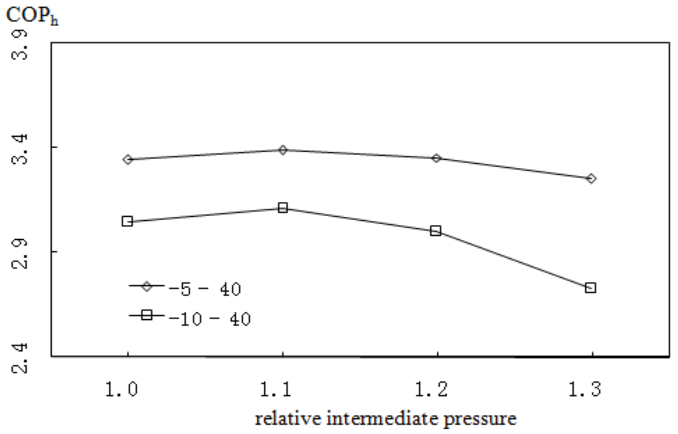


Figure 7 The variation of COP_h with relative intermediate pressure

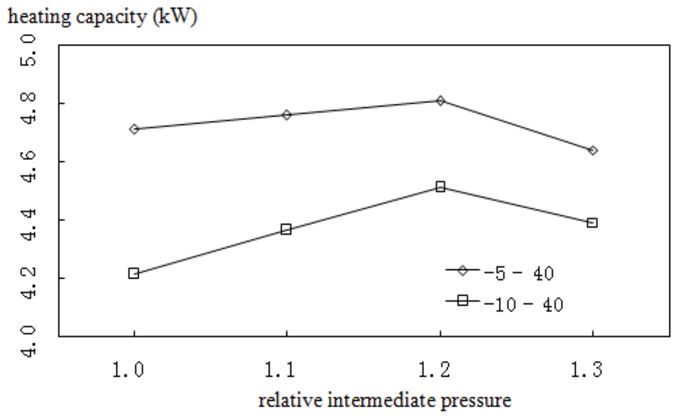


Figure 8 The variation of heating capacity with relative intermediate pressure

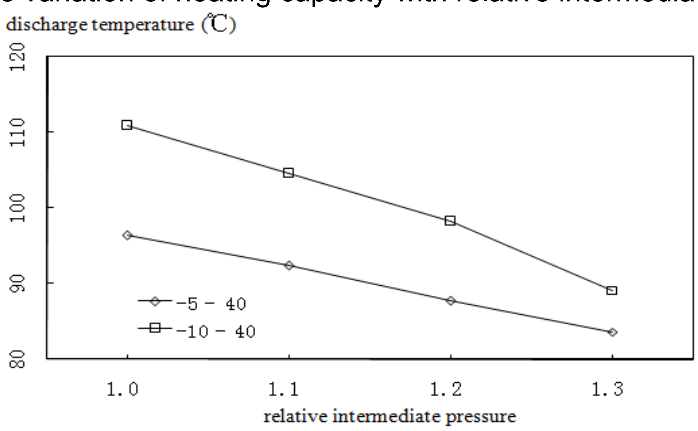


Figure 9 The variation of discharge temperature with relative intermediate pressure under heating mode

4 CONCLUSIONS

Through the theoretical analysis and experimental research on the two-stage compression refrigeration/heat pump system using the refrigerant of R410A coupled with dual-cylinder rolling piston compressor and flash tank, the conclusions can be drawn as follows.

1) The coefficient of performance of the two-stage compression refrigeration/heat pump system increases and then decreases when the volume ratio of the higher-pressure chamber to the lower-pressure one grows up, and the favorable value of the volume ratio is between 0.70 and 0.85.
2) The coefficient of performance and cooling/heating capacity increases and then decreases when the relative intermediate pressure rises. The maximum coefficient of performance and cooling capacity appear when the relative intermediate pressure is around 1.1, and the maximum heating capacity appears when relative intermediate pressure is around 1.2. So, the value of the relative intermediate pressure between 1.1 and 1.2 is suitable for the two-stage system.
3) The discharge temperature of the compressor decreases significantly with increase of the relative intermediate pressure. In the range of experimental conditions, the discharge temperature goes down about 2.3°C～4.8°C under cooling conditions and 4.1°C～9.0°C under heating conditions whenever the relative intermediate pressure rises 0.1.

5 ACKNOWLEDGEMENTS

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