Theoretical and Experimental Study on Rotary Compressor with Double Vapor Injection and Its System

H J Wei\textsuperscript{1, 2, 3}, J Xu\textsuperscript{1, 2, 3}, B Yu\textsuperscript{* 3}, O X Yang\textsuperscript{1, 2, 3}, S G He\textsuperscript{3}, H H Zhao\textsuperscript{3} and Y P Hu\textsuperscript{1, 2, 3}

\textsuperscript{1} State Key Laboratory of Air-conditioning Equipment and System Energy Conservation, ZhuHai, Guangdong 519070, China
Phone: +86-756-8587795, Fax: +86-756-8668386
\textsuperscript{*}E-mail: jvcanjian@163.com
\textsuperscript{2} Guangdong State Key Laboratory of Refrigeration Equipment and Energy Conservation Technology, ZhuHai, Guangdong 519070, China
\textsuperscript{3} Gree Electric Appliances, Inc. of Zhuhai, ZhuHai, Guangdong 519070, China

Abstract. This paper takes three kinds of double vapor injection compression cycle as the research object, that is, the two-stage compression cycle with double vapor injection on low-pressure cylinder and intermediate of two cylinders (Cycle A), the two-stage compression cycle with double vapor injection on high-pressure cylinder and intermediate of two cylinders (Cycle B), and the three-stage compression cycle with double vapor injection (Cycle C). According to the law of conservation of mass and energy, the thermodynamic analysis model of each cycle was established, and the influence of saturation temperature of double vapor injection on the performance of each cycle was analyzed. The theoretical analysis results indicated that the refrigeration capacity of cycle A, cycle B and cycle C increase by 12.5\%, 3.5\% and 4.6\%, respectively, and COP of that increase by 3.09\%, 2.23\% and 3.22\%, respectively, compared with that of single stage compression cycle. Based on cycle A, a single-machine two-stage rotary compressor with double vapor injection and its heat pump air conditioning are made. The experimental results showed that, the heating capacity of cycle A can increase by 14.6\% when the heating COP is equivalent, and the heating COP can increase by 4.4\% when the heating capacity is equivalent, compared with the two-stage compression cycle with single vapor injection, under the –15°C ambient temperature heating condition.

1. Introduction
In recent years, with the advancement of “coal to clean energy” project in North China, the vapor injection compression technologies have developed rapidly [1]. The quasi two-stage compression
technology with vapor injection on compression chamber was first studied and applied in scroll compressor [2]. In order to solve the problems of insufficiency and back-flow of vapor injection, some technical solutions have been developed for rotary compressor, such as “vapor injection structure on blade [3]”, “vapor injection ports on the mid plate between two cylinders [4]” and “end-plate injection structure with check valve [5]”. Therefore, the heating capacity and COP of heat pump air conditioning in low temperature environment are improved to a certain extent. The two-stage compression technology has the characteristics of small compression ratio and small power loss during. The two-stage rotary compressor based on the optimal volume ratio [6] and vapor injection valve technology [7] has excellent heating performance and is widely used in air source heat pump.

All the above technologies belong to single vapor injection technology, while the double vapor injection technology can further improve the performance of refrigeration cycle in theory. Jung D et al. simulated and evaluated the performance of multi-stage heat pumps using mixtures [8]. Mathison M et al. established a mathematical model to evaluate the effect of the number of injection ports on the performance of vapor injection compression cycle [9]. Theoretical and experimental study of the double vapor injection scroll compressor shows that the capability and COP have been improved compared with the single vapor injection [10-11]. In this paper, according to the working characteristics of the rotary compressor, the thermodynamic models of three kinds of double vapor injection compression cycle were established to simulate their performance. Based on the optimum cycle mode, a single-machine two-stage rotary compressor with double vapor injection and its heat pump air conditioning are made, and the performance was tested at low temperature conditions.

2. Analysis models

2.1 Rotary Compressor with Double Vapor Injection and Its System

In order to realize double vapor injection function on rotary compressor, either two-cylinder structure or three-cylinder structure can be used. When the compressor is of two-cylinder structure, one vapor injection port is set between two cylinders, and the other can be on the compression chamber of any cylinder. When the compressor is of three-cylinder structure, the two vapor injection ports are set between the two adjacent cylinders. Considering the high cost performance requirements of small air source heat pump, the main form of vapor injection cycle is the system with flash tank [12]. In order to avoid back-flow and insufficiency of vapor injection, it is better to start vapor injection immediately after the compression process [4, 6]. Based on the above characteristics of vapor injection on cylinder of rotary compressor, Figure 1 shows the schematic diagram and pressure-enthalpy diagram of three kinds of double vapor injection compression cycle.

It can be seen from Figure 1-(a), Figure 1-(c) and Figure 1-(e) that saturated vapors 7 and 10 with different pressures are produced by three expansion valves and two flash tanks in each refrigeration cycle, and the two differences are compression structure and location of vapor injection ports. In cycle A, a two-stage rotary compressor is used, and the double vapor injection ports are located on low-pressure cylinder and intermediate of two cylinders respectively. In cycle B, the double vapor injection ports are located on high-pressure cylinder and intermediate of two cylinders respectively. In cycle C, a three-stage rotary compressor with double vapor injection ports is used.
Figure 1. Schematic diagram and $p$-$h$ diagram of double vapor injection compression cycle

Take Figure 1-(b) as an example to illustrate the change process of refrigerant state. After the suction gas 1 starts to be compressed, the vapor 10 from the 2nd stage flash tank is injected into the compression chamber of low-pressure cylinder, and then the gas is continuously compressed from the mixed state 13 to the state 2 and discharged. The vapor 7 from the 1st stage flash tank is injected...
between the two cylinders and mixed with the exhaust of the low-pressure cylinder to state 3. Then it is sucked into the high-pressure cylinder, compressed to state 4 and discharged into the condenser.

2.2 Model of vapor injection on cylinder
Taking the process vapor injection in cycle A as an example, the model of vapor injection on cylinder is analyzed. The vapor injection flow is a large pressure difference injection process, which can be regarded as a constant-volume adiabatic process [12]. The mass conservation equation, energy conservation equation and state equation can be expressed as:

\[
\begin{align*}
\dot{m}_{13} &= \dot{m}_1 + \dot{m}_{10} \\
\dot{m}_{13}u_{13} &= \dot{m}_1u_1 + \dot{m}_{10}h_{t0} \\
(T_{13}, p_{13}, s_{13}) &= f(v_{13}, u_{13})
\end{align*}
\]

(1)

Where, \( \dot{m}_1 \), \( \dot{m}_{10} \) and \( \dot{m}_{13} \) are the suction mass, the vapor injection mass and the refrigerant mass in the compression chamber after vapor injection in each rotation cycle, respectively, \( h \) is the specific enthalpy, \( u \) is the internal energy, \( T \) is the temperature, \( p \) is the pressure, \( s \) is the specific entropy, and \( v \) is the specific volume. The relative vapor injection mass \( \alpha_{10} \) should meet the following requirements:

\[
\begin{align*}
\alpha_{10} &= \frac{m_{10}}{m_1 + m_{10}} \\
\alpha_{10} \leq x_9 \\
p_{13} &\leq p_{10}
\end{align*}
\]

(2)

Where, \( x_9 \) is the dryness of two-phase refrigerant behind the 2nd stage expansion valve. When \( \alpha_{10} < x_9 \), the refrigerant \( \alpha_{11} \) in front of the 3rd stage expansion valve is the two-phase refrigerant, so its specific enthalpy can be calculated as follows:

\[
h_{11} = \frac{h_9 - \alpha_{10}h_{t0}}{1 - \alpha_{10}}
\]

(3)

Process 1-13 is a pressure mutation process, and the technical work is as follows:

\[
w_{1-13} = \int vdp = (p_{13} - p_1) \frac{v_1 + v_{13}}{2}
\]

(4)

2.3 Thermodynamic models of three cycles
The process 2/7-3 of cycle A, the process 10/13-2 of cycle B and the double vapor injection of cycle C all belong to vapor injection on the intermediate of two cylinders, and their models can be referred to the relevant literature [12]. Assuming that the compression process is isentropic and the throttling process is adiabatic, the thermodynamic models of three cycles are established.

The specific refrigeration capacity and COP of the three cycles can be expressed as follows:

\[
q_c = h_1 - h_{12}
\]

\[
COP = \frac{q_c}{w}
\]

(5)

(6)

The specific technical work of cycle A, cycle B and cycle C are shown in equation (7), equation (8) and equation (9), respectively.

\[
w = (p_{13} - p_1) \frac{v_1 + v_{13}}{2} + \frac{h_2 - h_{t1}}{1 - \alpha_{10}} + \frac{h_4 - h_3}{(1 - \alpha_{10})(1 - x_6)}
\]

(7)

\[
w = h_{13} - h_1 + (p_3 - p_2) \frac{v_1 + v_{13}}{2} + \frac{h_4 - h_3}{(1 - x_9)(1 - \alpha_7)}
\]

(8)
\[ w = h_{13} - h_1 + \frac{h_2 - h_{14}}{1 - x_9} + \frac{h_4 - h_3}{(1 - x_6)(1 - x_7)} \]  

(9)

Where, \( \alpha \) is the relative vapor injection mass on high-pressure cylinder of cycle B.

3. Discussions on simulation results

In this paper, R32 is used in the double vapor injection compression cycles. According to the above analysis model, the theoretical performance of each cycle under typical low temperature heating condition was calculated. The parameters of the working condition are as follows: evaporation temperature is -20°C, condensation temperature is 45°C, superheat of suction gas is 10K, and subcooling of liquid is 7K. Figures 2, 3 and 4 show the map of COP varying with the saturation temperature of double vapor injection in three cycles respectively.

![Figure 2. COP varies with the saturation temperature at cycle A](image1)

![Figure 3. COP varies with the saturated temperature at Cycle](image2)

It can be seen from Figure 2 that the COP of cycle A first increases and then decreases with the saturation temperature of VI-LPC (vapor injection on low-pressure cylinder) and VI-ITC (vapor injection on the intermediate of two cylinders). The reason is that the relative vapor injection mass decreases with the increase of saturation temperature, and the refrigeration capacity and technical work decrease with the decrease of relative vapor injection mass. Therefore, COP reaches the maximum at a certain temperature. When the saturation temperature of VI-LPC is between -16°C and -13°C and the saturation temperature of VI-ITC is between 10°C and 20°C, COP is in the best range, and it’s better in the region near evaporation temperature. The main reason is that there is extra technical work loss of vapor injection on cylinder compared with that on the intermediate of two cylinders. According to equation (4), the smaller the difference between \( p_{13} \) and \( p_1 \) is, the lower the technical work loss will be. However, when the saturation temperature of VI-LPC is too low, the gas separated by the 2nd stage flash tank cannot be completely injected into the cylinder. According to equation (3), the refrigerant in front of the 3rd stage expansion valve does not reach the saturated liquid state, which makes the refrigeration capacity and COP decrease rapidly.

As can be seen from Figure 3, COP is higher in the area where the saturation temperature of VI-HPC (vapor injection on high-pressure cylinder) is close to the saturation temperature of VI-ITC. The reason is consistent with the above analysis. In this area, the smaller the difference between \( p_3 \) and \( p_2 \)
is, the lower the technical work loss will be. When COP is greater than 3.2, the saturation temperature area of cycle A is larger than that of cycle B.

![Figure 4](image1.png) COP varies with the saturation temperature at cycle C

![Figure 5](image2.png) Increase of refrigeration capacity and COP compared with SSCC

It can be seen from Figure 4 that the effects of FVI-ITC (the first vapor injection on the intermediate of two cylinders) and SVI-ITC (the second vapor injection on the intermediate of two cylinders) on cycle C are balanced, and the COP contour presents a regular ellipse. Compared with Figures 2, 3 and 4, the highest COP of cycle A and cycle C is approximate. For the problem of how to choose the double vapor injection cycle, Figure 5 gives the comparison of the optimal refrigeration capacity and COP at the optimum COP point among these cycles.

It can be seen from Figure 5 that the refrigeration capacity and COP improvement of cycle A, cycle B and cycle C are all greater than that of TSCC (the two-stage compression cycle with single vapor injection). The refrigeration capacity of cycle A, cycle B and cycle C increase by 12.5%, 3.5% and 4.6% respectively, and COP of that increase by 3.09%, 2.23% and 3.22% respectively, compared with that of SSCC (single stage compression cycle). Compared with cycle C, the COP increase of cycle A is slightly lower, but the refrigeration capacity increase is more significant. Because the second vapor injection of cycle A is VI-LPC, when $p_{10}$ is slightly higher than the suction pressure, vapor injection can be realized. However, cycle B and C are both VI-ITC, so the vapor pressure is affected by cylinder volume ratio. When the COP is optimal, the vapor pressure is greater than that of cycle A, so the enthalpy in front of evaporator of cycle A is even lower. At the same time, considering the complexity of compressor structure, cycle A is a better choice.

4. Prototypes of compressor and air conditioning system

Based on cycle A, a two-stage rotary compressor with double vapor injection has been developed. On the basis of the original two-stage rotary compressor QXFT-B096, the vapor injection structure is set on low-pressure cylinder. The volume of the low-pressure cylinder is 9.6cm³, and the volume ratio of the high-pressure cylinder to the low-pressure cylinder is 0.68. On the basis of the original two-stage compression air conditioning KFR-35GW, a flash tank and an expansion valve are added. Figure 6 shows the three-dimensional model of the two-stage rotary compressor with double vapor injection, and Figure 7 the photo of the heat pump air conditioning.
As can be seen from Figure 6, the two vapor injection structures are as follows: (1) the vapor injection inlet G is connected with the working chamber of the low-pressure cylinder D through the vapor channel of the separation plate F to form the low-pressure vapor injection structure; (2) the inner chamber of the sub bearing is set between the low-pressure cylinder D and the high-pressure cylinder C, and the vapor injection inlet H is connected with the inner chamber to form the high-pressure vapor injection structure.
injection structure. For the low-pressure vapor injection structure, the position of the rolling piston changes with the eccentric movement of the crankshaft, and the vapor channel is blocked by the end face of the piston to close the vapor injection structure. For the high-pressure vapor injection structure, a valve is set between the inner chamber and the vapor injection inlet H, so as to avoid the back-flow of the gas from the inner chamber.

As can be seen from Figure 7, the air conditioning system uses three electronic expansion valves to adjust the saturation temperature of the double vapor injection, and uses two flash tanks to separate the double vapor with different temperature and pressure characteristics. An on-off valve is set between the gas outlet of the flash tank and the vapor inlet to open or close the vapor channel as required. In addition, a thin tube is connected to the vapor channel, which is used to access pressure sensor to monitor the vapor pressure.

5. Discussions on experimental results
The air enthalpy difference method is used to test the performance of the two-stage compression air conditioning system. The test condition is -15°C ambient temperature heating condition, the indoor dry-bulb and wet-bulb temperature are 20°C and 15°C respectively, and the outdoor dry-bulb temperature is -15°C. The heating performance of the air conditioning was tested at the compressor operating frequency of 110Hz and 95Hz, and the performance was compared with the single vapor injection two-stage compression system and single-stage compression system under the same condenser and evaporator. Figure 8 and Figure 9 show the variation of relative heating capacity and relative COP (compared with single-stage compression system) with the saturation temperature of double vapor injection two-stage compression air conditioning system.

As can be seen from Figure 8 and Figure 9, the maximum heating capacity of the two-stage compression system with double vapor injection is increased by more than 47%, and the maximum COP is increased by more than 10%. The reason why the experimental capacity is better than the simulated value is that the compression ratio and pressure difference of the low-pressure cylinder are greatly reduced, and the volume efficiency has been significantly improved. There is an optimal value of heating capacity with the variation of saturation temperature of VI-LPC. In the temperature range of
-12°C to -5°C, the refrigerant enthalpy at the inlet of the evaporator decreases with the decrease of the saturation temperature of VI-LPC, so the heating capacity increases. When the saturation temperature of VI-LPC is lower than -12°C, the heating capacity decreases because the vapor is not completely injected into the low-pressure cylinder. On the other hand, the heating capacity decreases with the increase of the saturation temperature of VI-ITC, because the exhaust temperature and the heating capacity decrease rapidly with the two-phase refrigerant injection.

The volume ratio of high-pressure cylinder to low-pressure cylinder is a constant value, so the state of the refrigerant entered by vapor injection inlet is affected by injection pressure [13]. When the injection pressure is too low, the separated gas of flash tank can not be completely injected. When the injection pressure is too high, some liquid can be injected. This is the reason why COP has an optimal value with the change of saturation temperature.

Table 1. Experimental results of different cycles at low ambient temperature heating condition

| Compressor type | Cycle mode | Operating frequency | Heating capacity | COP  | Discharge temperature |
|-----------------|------------|---------------------|------------------|------|-----------------------|
| QXF-B096        |            | 110Hz               | 2625W            | 1.84 | 82.6°C                |
| QXFT-B096       | TSCC       | 110Hz               | 3308W            | 2.05 | 74.8°C                |
| QXFTD-B096      | Cycle A    | 110Hz               | 3792W            | 2.03 | 68.8°C                |
| QXFTD-B096      | Cycle A    | 95Hz                | 3355W            | 2.14 | 66.1°C                |

Table 1 shows the experimental results of the optimal COP, heating capacity and discharge temperature under the set operating frequency. Compared with SSCC, the maximum heating capacity of cycle A is increased by 44.5%, and the COP is increased by 10.3%. Compared with TSCC, the heating capacity of cycle A can increase by 14.6% when the heating COP is equivalent, and the heating COP can increase by 4.4% when the heating capacity is equivalent. Since the discharge temperature of the cycle A can be reduced by 6°C to 8.7°C than TSCC, theoretically, the double vapor injection system can operate at lower ambient temperature (e.g. -30°C). However, due to the limitation of experimental conditions, there is no reliable experimental result yet, which will be the next research work.

The reasons why the experimental results are better than the simulated results in Figure 5 are as follows: (1) The volume ratio of the compressor prototype is larger than simulated value in Figure 5 (0.53), which is beneficial to increase the vapor injection mass flow, so as to improve the heating capacity; (2) The efficiency of the compressor is not considered in the simulation, but the two-stage compression and vapor injection process are beneficial to reduce the compression ratio and improve the isentropic efficiency, which is beneficial to improve COP.

6. Conclusions
In this paper, the thermodynamic performance of three kinds of double vapor injection compression cycles is simulated. A two-stage rotary compressor with double vapor injection and its system are developed. The influence of saturation temperature of double vapor injection on heating performance at low ambient temperature was studied.

Under typical low temperature heating conditions, the theoretical refrigeration capacity of cycle A, cycle B and cycle C are increased by 12.5%, 3.5% and 4.6% respectively, and the theoretical COP is
increased by 3.09%, 2.23% and 3.22% respectively, compared with single vapor injection two-stage compression cycle.

With the increase of saturation temperature of vapor injection, COP increased first and then decreased. The results show that there are differences in the optimal saturation temperature regions of COP among the three kinds of cycles. The region of cycle A is close to the evaporation temperature, the region of cycle B is close to the saturation temperature of VI-ITC, and the region of cycle C is affected by the equilibrium of double vapor temperature.

The experimental results showed that, the heating capacity of cycle A can increase by 14.6% when the heating COP is equivalent, and the heating COP can increase by 4.4% when the heating capacity is equivalent, compared with TSCC, under the −15°C ambient temperature heating condition.

References

[1] Ma G, Wang L, and Liu Y 2019 Technological progress of rolling piston compressor Refrigeration and Air-conditioning 19(2) 55-64 (in Chinese)

[2] Ma G Y, and Zhao H X 2008 Experimental study of a heat pump system with flash tank coupled with scroll compressor Energy and Buildings 40(5) 697-701.

[3] Liu X, Wang B, Li X, and Shi W 2016 Performance of Rotary Compressor with Vapor Injection Mechanism on Blade J. Refrigeration 37(2) 1-8 (in Chinese)

[4] Yan G, Jia Q, and Bai T 2015 Experimental investigation on vapor injection heat pump with a newly designed twin rotary variable speed compressor for cold regions Int. J. of Refrigeration http://dx.doi.org/doi:10.1016/j.ijrefrig.2015.10.024

[5] Wang B, Ding Y, and Shi W 2018 Experimental research on vapor-injected rotary compressor through end-plate injection structure with check valve Int. J. Refrigeration https://doi.org/10.1016/j.ijrefrig.2018.08.018

[6] Hu Y, Luo H, Wu J, Wei H and Yang O 2018 Research on two-stage rotary compressor with refrigerant injection for cold climate heat pump International Compressor Engineering Conference. Purdue Paper 2604

[7] Hu Y S, Wei H J, Yu B, et al 2019 Research on the vapor injection of two-stage rotary compressor IOP Conf. Series: Materials Science and Engineering 604 012072

[8] Jung D, Kim H K, and Kim O 1999 A study on the performance of multistage heat pumps using mixtures Int. J. Refrigeration 22 402-13

[9] Mathison M M, Braun J E, and Groll E A Performance limit for economized cycles with continuous refrigerant injection Int. J. Refrigeration 34 234-242

[10] Xu S, and Ma G Y 2015 Characteristic of Scroll Compressor Refrigeration System with Twice Vapor Injection J. Refrigeration 36(1) 40-4 (in Chinese)

[11] Kang D, Jeong J H, and Ryu B 2018 Heating performance of a VRF heat pump system incorporating double vapor injection in scroll compressor Int. J. Refrigeration https://doi.org/10.1016/j.ijrefrig.2018.09.027

[12] Huang H 2018 Technology and application of two stage compression variable volume ratio air source heat pump (Beijing: Machine Industry Press)

[13] Yu B, Wei H, and Yang O 2019 Effect of Intermediate Pressure on Performance of Single-machine Two-stage Rotary Compressor J. Refrigeration Technology 42(1) 43-7 (in Chinese)