# STUDY ON QUASI TWO-STAGE COMPRESSION CYCLE CHARACTERISTICS OF THE WATER HEATER OF THE AIR SOURCE HEAT PUMP IN COLD AREA

by

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*The water heater of an air source heat pump has the disadvantages of high exhaust temperature and weak heating performance in a cold area. The quasi two-stage compression cycle technology can effectively improve its operation characteristics in the cold area. In this paper, a rotor compressor with a medium pressure air supply is used to develop the water heater with R410A as the circulating working fluid. The system's heating performance under low temperature environments is experimentally studied. The results show that the exhaust temperature of the water heater system with the medium pressure air supply is lower than those without the air-supply system.*

Key words: *heat pump water heater*, *quasi two-stage compression, low temperature, heating performance*

## Introduction

In recent years, the winter haze weather frequently occurred in northern China. The coal-burning system, such as the coal-fired boiler heating and the bulk coal heating, is one of the important reasons for generating haze weather. The air source heat pump takes clean air as the cold and heat sources and realizes the energy saving for both refrigeration and heating based on the inverse Carnot circulation principle [1, 2]. As a form of air source heat pump, the water heater of the air source heat pump has the advantages of stable operation, high efficiency, energy-saving, flexible combination, and convenient installation. The application of the water heater is in line with the strategy of environmental protection, energy-saving, and sustainable development in China [3]. When the air source heat pump is operating in the northern cold area in winter, the outdoor environment temperature is low. The evaporation temperature becomes low, the suction specific volume of the compressor and the compressor exhaust temperature increase, the unit refrigerant heating capacity, the system heating performance coefficient, and the economic performance decrease [4].

In view of the shortcomings of air source heat pumps in a cold area, such as weak heating performance and unable to meet the requirements of heating, some scholars have found that the quasi-double-stage compressed heat pump system has a better heating effect when it is running in the cold area. Because of its high cost-performance ratio, it will be wide-

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ly used in air source heat pump systems. Ma and Zhao [5] experimentally studied the air source heat pump system with a flasher and found that the heat pump system with a flasher has a better heating effect than the heat pump system with a sub-cooler. Heo, *et al.* [6] conducted an experimental study on the heating performance of an air source heat pump system with an air compressor and an intermediate air supply in a low temperature environment. The results showed that the heating capacity and energy efficiency of the air source heat pump system with intermediate air supply increased greatly compared with the system without an air supply. Roh and Kim [7] experimentally studied the heating performance of the air supply source heat pump system with R410A as a refrigerant and found that the refrigerant injection rate was significantly affected by the intermediate air supply pressure, and the higher the intermediate air supply pressure, the greater the heating capacity and energy efficiency of the system. Dutta, *et al.* [8] carried out an experimental study on quasi-two-stage scroll refrigeration compression, indoor return air rate, and indoor air volume on the heating performance of the electric vehicle heat pump system with propane refrigerant. Liu, *et al.* [9] studied the influence of outdoor environment temperature of an electric vehicle heat pump air conditioning system. The results show that the COP of the maximum heating capacity of the system is still more than 1.7 even at –20 °C, and the heating capacity can be further improved by increasing the return air ratio. Qin, *et al.* [10, 11] also studied the heating performance of an electric vehicle heat pump air conditioning system. Tang, *et al.* [12] conducted experimental research on the application of quasi double-stage compression cycle technology to pure electric vehicle air conditioners. Ou, *et al.* [13] also conducted experimental research on the application of quasi double-stage compression cycle technology to air source heat pumps. He and Li [14] recommended a straightforward method for thermal problems.

Based on the quasi-double-stage compression cycle theory, this paper uses a rotor compressor with air supply to develop an air source heat pump water heater with R410A as the circulating working fluid. The heating performance of the system in cold regions is experimentally studied, which provides theoretical and data reference for further improving the system and developing new products with high efficiency.

## Test system

The medium pressure air-supply air source heat pump water heater system is composed of a rotor compressor, water-side plate heat exchanger, air-side tube fin heat exchanger, DC brushless motor, four-way reversing valve, electronic expansion valve, economizer (intermediate plate heat exchanger), liquid reservoir, *etc.* the heating principle of the system is shown in fig. 1. The system includes two cycles, the right side is the heat pump system cycle, and the left side is the user side hot water heat exchange cycle. The theoretical cycle of the medium-pressure supply air-supply principle is shown in fig. 2.

The theoretical calculation formula is:

– Evaporating-side cooling capacity:

$$Q_c = \dot{m}_1 (h_4 - h_1)$$

where $Q_c$ [kW] is the evaporator heating capacity, $\dot{m}_1$ [kgs$^{-1}$] – the mass-flow of the main refrigerant, $h_4$ [kJkg$^{-1}$] – the enthalpy of the refrigerant at the inlet of the evaporator, and $h_1$ [kJkg$^{-1}$] – the enthalpy of the refrigerant at the outlet of the evaporator.

– Heat exchange of economizer:

$$Q_e = \dot{m}_2 (h_7 - h_6)$$

where $Q_e$ [kW] is the heat exchange of economizer, $\dot{m}_2$ [kgs$^{-1}$] – the mass-flow rate of refrigerant in an auxiliary circuit, $h_6$ [kJkg$^{-1}$] – the enthalpy of the refrigerant at the inlet of economizer, and $h_7$ [kJkg$^{-1}$] – the enthalpy of the refrigerant at the outlet of economizer.

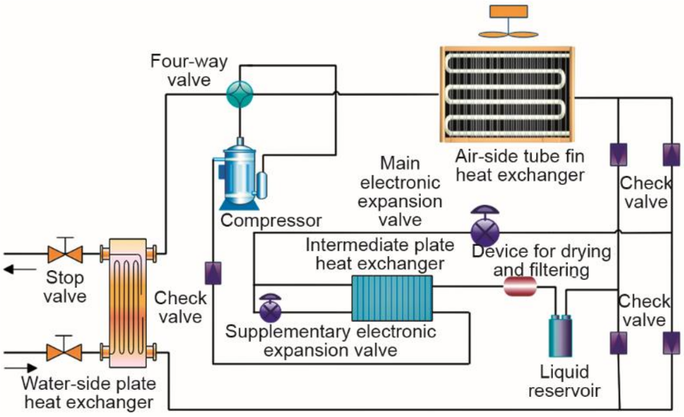


**Figure 1. System heating principle**

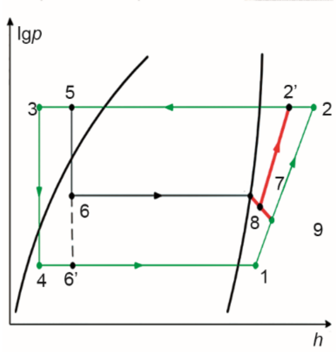


**Figure 2. Theoretical cycle of the principle of medium-pressure air-supply increasing enthalpy**

– Heating capacity of condensation side:

$$Q_h = (\dot{m}_1 + \dot{m}_2)(h_{2'} - h_3)$$

where $Q_h$ [kW] is the heating capacity of condenser, $h_2$ [kJkg$^{-1}$] – the enthalpy of the refrigerant at the inlet of condenser, and $h_3$ [kJkg$^{-1}$] – the enthalpy of the refrigerant at the outlet of the condenser.

– Compressor power consumption

$$W = \dot{m}_1(h_9 - h_1) + (\dot{m}_1 + \dot{m}_2)(h_{2'} - h_8)$$

where $W$ [kW] is the power of the compressor, $h_9$ [kJkg$^{-1}$] – the enthalpy of low pressure stage compression in the compression chamber before the air supply valve is opened, and $h_8$ [kJkg$^{-1}$] – the enthalpy of the refrigerant in the auxiliary circuit mixed with the refrigerant in the compression chamber after the air supply valve is opened.

– Relative amount of air supply:

$$R_m = \frac{\dot{m}_2}{\dot{m}_1 + \dot{m}_2}$$

where $R_m$ [kgkg$^{-1}$] is the relative amount of air supply.

– Air supply pressure ratio:

$$R_p = \frac{P_2 - P_8}{P_2 - P_1}$$

where $R_p$ [%] is the air supply pressure ratio, $P_2$ [kPa] – the compressor exhaust pressure, $P_8$ [kPa] – the pressure after the air supply valve opens mixed gas of the refrigerant, and $P_1$ [kPa] – the suction pressure of the compressor.

– System refrigeration performance coefficient:

$$COP_c = \frac{Q_c}{W} = \frac{\dot{m}_1(h_4 - h_1)}{\dot{m}_1(h_9 - h_1) + (\dot{m}_1 + \dot{m}_2)(h_{2'} - h_8)}$$

where $COP_c$ is the refrigeration performance coefficient.

– System heating performance coefficient:

$$COP_{\mathrm{h}} = \frac{Q_h}{W} = \frac{\dot{m}_1 (h_4 - h_1)}{\dot{m}_1 (h_9 - h_1) + (\dot{m}_1 + \dot{m}_2)(h_{2'} - h_8)}$$

where $COP_{\mathrm{h}}$ is the heating performance coefficient.

## Test device

The experiment was carried out in the enthalpy difference Laboratory of science and Technology Park of Zhongyuan University of technology. In this test, the inverter air-jet enthalpy-increasing rotor compressor WHP32900AEKTQ9JK produced by Shanghai Highly was selected and the speed range is 900~6600 rpm. The water-side heat exchanger adopts P80Hx38/1P-SC-M brazed plate heat exchanger produced by Shurepu. The air-side tube-fin heat exchanger is vertically arranged in an L shape, the length of the copper tube is 960 mm, the number of processes is 9, the outer area of the fin is 77.6 square meters, the average area of the copper tube is 3.58 square meters, and two fans are matched on the central axis of the heat exchanger. The test device is equipped with 5 pressure and 13 temperature measuring points. The pressure is measured by the pressure sensor Emerson PT5-50M/T, 11 T-type thermocouples are arranged at the unit measuring points with tin foil tape and heat conductive agent to measure the temperature of the refrigerant circuit, insert two PT100 platinum resistors into the inlet and outlet pipes of the unit under test to measure the temperature of the inlet and outlet water. Figure 3 shows the test device.

**Figure 3. Test device**

## Operating condition of the test

According to GB/T10870-2014 *Vapor compression cycle chilled water (heat pump) unit performance test method*, GB/T25127.1-2010 *low ambient temperature air source heat pump (chiller) unit Part I* requirements are formulated as shown in tab. 1. In the test process, the compressor speed is 5000 rpm. The circulating water flow is 16.69 $m^3$/h, the superheat setting the value of the main circuit electronic expansion valve is 5 K, the setting value of the auxiliary circuit electronic expansion valve is 20 K.

## Analysis of test results

Figure 4 shows the variation curve of compressor exhaust temperature with outdoor ambient temperature, it can be seen from the figure that the exhaust temperature of the system increases with the decrease of the outdoor temperature. Under the same outdoor temperature conditions, the exhaust temperature of the air supply is lower than that without the air supply. When the outdoor temperature drops from 7 ℃ to –25 ℃, the exhaust temperature of the compressor in the air supply system decreases by 5.1 ℃, 13.9 ℃, 14.2 ℃, 11.7 ℃, 12.3 ℃, and 9.8 ℃ respectively, compared with the non-air-supply system. Especially in the outdoor ultra-low temperature of –15 ℃ and –25 ℃, the exhaust temperature of the air supply system

**Table 1. Operating condition of the test**

| Operating condition | Heat source side (air side) | | Use side (hot water side) | |
|---|---|---|---|---|
| | Dry bulb temperature [°C] | Wet bulb temperature [°C] | Initial water temperature [°C] | End water temperature [°C] |
| Nominal operating condition | 7 | 6 | 9 | 55 |
| Variable operating condition | 2 | – | 9 | 55 |
| Low temperature operating condition | –5 | –8 | 9 | 55 |
| Low temperature operating condition | –10 | – | 9 | 55 |
| Ultra low temperature operating condition | –15 | – | 9 | 55 |
| Ultra low temperature operating condition | –25 | – | 9 | 55 |

is higher than 100.0 °C, while the exhaust temperature of the air supply system is lower than 96.0 °C, which ensures the compressor's performance safe and stable operation.

Figure 5 shows the variation curve of heating capacity temperature with outdoor ambient temperature, and it can be seen from the figure that the heating capacity decreases with the decrease of outdoor temperature. Under the same outdoor temperature conditions, the heating capacity of the air supply is increased compared to without the air supply. When the outdoor temperature drops from 7 °C to –25 °C, the heating capacity of the air supply system increases by 6.2% to 15.5% compared to the non-air-supply system. Among them, when the outdoor environment temperature is –5 °C, it is observed that the opening degree of the supplementary expansion valve is 89%, and the opening degree is larger, which leads to an increase in the mass-flow of refrigerant entering the condenser, at this time, the heating capacity increases the most.

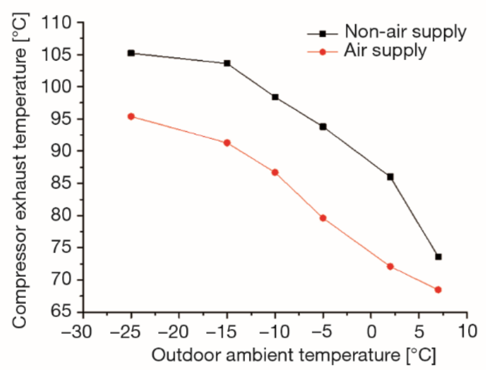


**Figure 4. Variation of compressor exhaust temperature with outdoor ambient temperature**

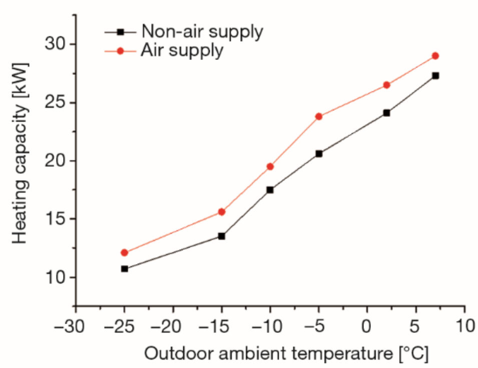


**Figure 5. Variation of heating capacity with outdoor ambient temperature**

Figure 6 shows the variation curve of compressor power with outdoor ambient temperature. It can be seen from the figure that the compressor power decreases with the decrease of outdoor temperature. Under the same outdoor temperature conditions, the power of the air-supply compressor is improved compared with that without the air supply. When the outdoor temperature drops from 7 °C to –25 °C, the compressor power of the air supply system is increased by 2.8% to 9.5% compared with without air-supply system.

Figure 7 shows the variation curve of system $COP_h$ with outdoor ambient temperature. It can be seen from the figure that the system $COP_h$ decreases with the decrease of outdoor temperature, under the same outdoor temperature, the $COP_h$ of air supply is higher than that of non air supply, when the outdoor temperature drops from 7 °C to –25 °C, the $COP_h$ of the air supply system increases by 3.3% to 9.6% compared without air-supply system. The reason for the increase in $COP_h$ of the air-supply increasing enthalpy system is the increasing range of heating capacity of the air-supply condenser is greater than that of the compressor due to air supply. The $COP_h$ is the ratio of the two, so the $COP_h$ of the air-supply system has been improved.

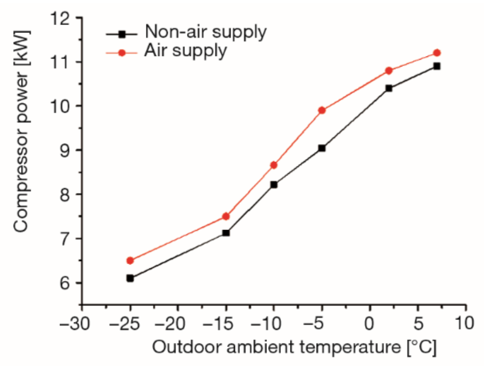


**Figure 6. Variation of compressor power with temperature ambient temperature**

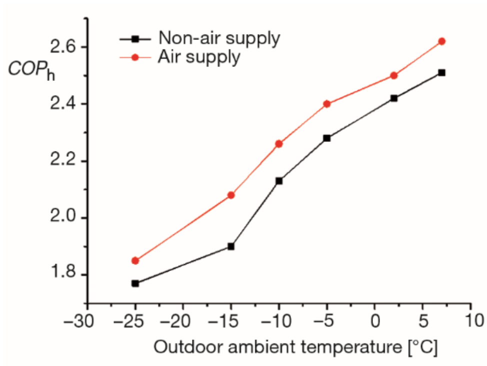


**Figure 7. Variation of $COP_h$ with outdoor outdoor ambient temperature**

## Conclusion

Based on the principle of increasing enthalpy of air supply, the heating performance of air-source heat pump water heaters with air-supply in the cold area is tested and researched, and the following main conclusions are as follows.

- In the cold area, the exhaust temperature of the air source heat pump water heater system with air supply is lower than that without the air supply. Under the outdoor ultra-low temperature of –15 ℃ and –25 ℃, the exhaust temperature of the non-air-supply system exceeds 100.0 ℃, while the exhaust temperature of the air supply system is lower than 96.0 ℃. At this time, the compressor can operate safely and stably.
- When the outdoor temperature drops from 7℃ to –25℃, compared without air-supply system, the heating capacity of the air supply system increases by 6.2-15.5%, the compressor power increases by 2.8-9.5%, and the $COP_h$ increases by 3.3%～9.6%.

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