VAPOR INJECTION TECHNOLOGY FOR HEAT-PUMP AIR CONDITIONER OF ELECTRIC BUS IN AN EXTREME ENVIRONMENT

by

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In order to solve the problems of a heat pump air conditioner for an electric bus under extreme conditions, such as the large compression ratio, the high discharge temperature, the reduced system performance, the frequent shutdown of the compressor due to the overheat protection, this paper proposes a vapor injection technology with an economizer, and carries out a theoretical analysis of the process to reveal the effect of the vapor injection technology on the performance of the air conditioner. The results show that at ultra-low temperature heating operating conditions, when the compressor speed increases from 2000 rpm to 5000 rpm, the heating capacity of the vapor injection system increases from 16.2% to 22.7%, and the heating performance coefficient increases from 2.8% to 14.2%.

Key words: *vapor injection technology, air conditioner, discharge temperature, heating performance coefficient*

Introduction

As an essential component of an electric bus, the air conditioner is directly related to the comfort of the internal environment of the electric bus and the safety of the whole vehicle. When the pure electric bus is running at a high temperature in summer and at a low temperature in winter, the exhaust temperature and exhaust pressure of the heat pump air conditioning system might be too high, and the viscosity of lubricating oil is reduced, and the performance of the system is attenuated, this condition will seriously affect the mileage and application range of the pure electric bus.

In view of the outstanding problems of air conditioners in extreme environments, many scholars have conducted extensive technical research, and much achievement has been obtained. Dutta *et al.* [1] carried out research on the vortex-type refrigeration compressor with liquid injection theoretically and experimentally. Ying *et al.* [2] established a mathematical model for gas-liquid two-phase mixture pressurization in the working process of the liquid injection scroll compressor and obtained the change law of the State parameters of the two-phase mixture with different liquid injection volumes. Fan and Wang [3] applied an imported liquid refrigerating method in a high temperature air conditioner. Jia *et al.* [4] studied theoretically and experimentally the characteristics of the air source heat pump, which was used to improve the air source heat pump motor insulation, and suggested a liquid injection bypass

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method to analyze the high temperature environment. Fei *et al.* [5] studied the effect of different spray volumes on the performance parameters of the heat pump water heater. Cui *et al.* [6] carried out theoretical analysis and experimental research on the system performance of the bus air conditioner under the maximum operating refrigeration condition with the application of the liquid injection technology and an economizer. He *et al.* [7] introduced a middle heat exchanger to a high temperature heat pump system and elucidated the effect of the vapor injection technology on the performance of the high temperature heat pump system. Xu and Wang [8] applied the vapor injection technology in the heat pump system under the condition of refrigeration and low temperature. Heo *et al.* [9] compared and analyzed the influences of four kinds of different forms of vapor injection on the performance of air-source heat pump systems. Hu *et al.* [10] established a mathematical model for analysis of the high temperature air conditioner based on the flash steam refrigerating technology, furthermore, the performance of R22 single-stage compression, R22 flash steam cooling in different outdoor environments was analyzed and compared. Yin [11] had a complete theoretical analysis of the high temperature refrigeration for an electric vehicle air-conditioning system with the jet economy. Chai *et al.* [12] carried out an experimental performance test of R134a refrigeration/heat pump system with flash steam under medium and low temperature conditions. Tang *et al.* [13] studied the performance of a quasi-two-stage scroll compressor for an electric bus air conditioner experimentally under five different ambient temperature conditions by using the vapor injection enthalpy-adding technology.

The results of the cited literature show that the liquid injection technology and the vapor injection technology can effectively reduce the discharge temperature of the compressor in extreme conditions, which is of great significance for ensuring the safe and reliable operation of the air conditioning system. However, the research of the liquid injection technology has not obviously improved the high compression ratio of the compressor at the extreme ambient temperature. The research of the vapor injection technology is mainly focused on the household and commercial low temperature heating field of single refrigerant. There is little research on the air conditioning field in a bus with refrigerant R407C. In this paper, the system performance of an electric bus air conditioner under extreme conditions is theoretically analyzed and experimentally studied by using vapor injection technology with an economizer.

Analysis of the theoretical process of vapor injection technology

Figure 1 is the expression of the theoretical cycle of the vapor injection technology on the pressure-enthalpy diagram. After the condenser (State 3') enters the economizer and is cooled and subcooled (State 3), the refrigerant is divided into two parts: one passes through the main road expansion valve that the refrigerant is throttled (State 5) and enters the evaporator and absorbs heat (State 1), and it is compressed to State 7 by the compressor, then it is mixed with the saturated gas refrigerant (State 6) of the vapor injection port (State 8), it will continue to be compressed by the compressor to State 2, and finally enters the condenser to release heat and condense. The other part of the refrigerant is returned to the economizer by throttling the supplementary expansion valve (State 4), and the economizer absorbs the heat of

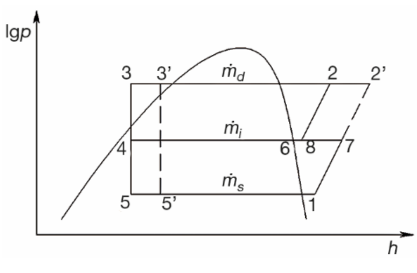

Figure 1. Representation of the theoretical cycle of the vapor injection technology on the pressure enthalpy diagram

the refrigerant (State 3'), which enters the economizer and then is evaporated, and the saturated gaseous refrigerant (State 6) passes the vapor injection port that is on the compressor (in the compressor's intermediate compression chamber), then it enters the compressor and mixes with the other superheated refrigerant and is compressed into State 7 (State 8), finally, it is compressed to State 2 through the compressor.

According to the law of conservation of mass, we have the following equation:

$$\dot{m}_i + \dot{m}_s = \dot{m}_d \tag{1}$$

where $\dot{m}_d$ [kgs^{-1}] is the mass-flow rate of condenser refrigerant, $\dot{m}_i$ [kgs^{-1}] – the mending refrigerant mass-flow rate, and $\dot{m}_s$ [kgs^{-1}] – the mass-flow of evaporator refrigerant.

According to the principle of conservation of energy [14, 15], the balance equation of the economizer is given by:

$$\dot{m}_i (h_6 - h_4) = \dot{m}_d (h_{3'} - h_3) \tag{2}$$

By eqs. (1) and (2), we have:

$$\dot{m}_s (h_{5'} - h_5) = \dot{m}_i (h_6 - h_{3'}) \tag{3}$$

where h_3 [kJkg^{-1}] is the enthalpy of the refrigerant at the inlet of the main expansion valve, $h_{3'}$ [kJkg^{-1}] – the enthalpy of the refrigerant at condenser outlet, h_4 [kJkg^{-1}] – the enthalpy value of refrigerant at the outlet of the repair expansion valve, h_5 [kJkg^{-1}] – the enthalpy value of refrigerant at the outlet of the main expansion valve, $h_{5'}$ [kJkg^{-1}] – the enthalpy value of refrigerant at the outlet of expansion valve without vapor injection, and h_6 [kJkg^{-1}] – the enthalpy value of refrigerant at the compressor inlet.

– The refrigeration capacity of the system is given by:

$$\begin{aligned} Q_l &= \dot{m}_s (h_1 - h_5) \\ &= \dot{m}_s (h_1 - h_{5'}) + \dot{m}_s (h_{5'} - h_5) \\ &= \dot{m}_s (h_1 - h_{5'}) + \dot{m}_i (h_6 - h_{3'}) \end{aligned} \tag{4}$$

where Q_l [kW] is the refrigerating capacity of medium pressure air replenishment system.

Compared with that without vapor injection system, the refrigeration capacity of the vapor injection system is given by:

$$\begin{aligned} \Delta Q_l &= Q_l - Q_l^{'} \\ &= \dot{m}_s (h_1 - h_5) - \dot{m}_s (h_1 - h_{5'}) \\ &= \dot{m}_s (h_{5'} - h_5) \\ &= \dot{m}_i (h_6 - h_{3'}) \end{aligned} \tag{5}$$

where ΔQ_l [kW] is the increased cooling capacity of medium pressure air replenishment system and Q'_l [kW] – the refrigerating capacity of without vapor injection system.

– The power consumption of the system compressor is given by:

$$W_l = \dot{m}_s (h_7 - h_1) + \dot{m}_d (h_2 - h_8) \tag{6}$$

where W_l [kW] is the compressor power of medium pressure air supply system, h_1 [kJkg^{-1}] – the enthalpy value of refrigerant at the compressor intake port, h_2 [kJkg^{-1}] – the enthalpy value of refrigerant at the outlet of expansion valve without vapor injection, h_7 [kJkg^{-1}] – the enthalpy value of refrigerant in compressor medium pressure exhaust, and h_8 [kJkg^{-1}] – the refrigerant enthalpy value of compressor air supply mixing port.

It is assumed that the isentropic efficiency of State 7 to State 2' is the same as the isentropic efficiency from State 8 to State 2, comparison with that without vapor injection system shows that the power consumption of the vapor injection system is given by:

$$\begin{aligned}\Delta W &= W_l - W_l^{'} \\ &= \dot{m}_s(h_7 - h_1) + \dot{m}_d(h_2 - h_8) - \dot{m}_s(h_{2'} - h_1) \\ &= \dot{m}_i(h_2 - h_8)\end{aligned} \tag{7}$$

where ΔW [kW] is the increased compressor power for medium pressure air supply system, W'_l [kW] – the compressor power of without vapor injection system, and $h_{2'}$ [kJkg^{-1}] – the refrigerant enthalpy value at the exhaust port without vapor injection compressor.

By analyzing eqs. (6) and (7) which show that the vapor injection compressor in the compression process can be approximately for two-stage compression process, the first level compression from State 1 to State 7, the compressor's power consumption is $\dot{m}_s$ ($h_7 - h_1$), the secondary compression is a mixed compression at the State 8 to the State 2, the power consumption of the compressor is $\dot{m}_d$ ($h_2 - h_8$), which is different from the vapor injection compressor power's consumption, that is $\dot{m}_s$ ($h_{2'} - h_1$). It is obvious that the vapor injection compressor can greatly increase the power consumption for $\dot{m}_i$ ($h_2 - h_8$), compensating for compression in the compressor power consumption.

– The heating capacity of the system is given by:

$$\begin{aligned}Q_r &= \dot{m}_d(h_2 - h_{3'}) \\ &= (\dot{m}_i + \dot{m}_s)(h_2 - h_{3'})\end{aligned} \tag{8}$$

where Q_r [kW] is the heating capacity of the medium pressure air supply system.

Compared with that without the vapor injection system, the heating capacity of the vapor injection system can be expressed:

$$\begin{aligned}\Delta Q_r &= Q_r - Q_r^{'} \\ &= (\dot{m}_i + \dot{m}_s)(h_2 - h_{3'}) - \dot{m}_s(h_{2'} - h_{3'}) \\ &= \dot{m}_i(h_2 - h_{3'}) - \dot{m}_s(h_{2'} - h_2)\end{aligned} \tag{9}$$

where ΔQ_r [kW] is the increased heating capacity for medium pressure air supply system and W'_l [kW] – the heating capacity of the without vapor injection system.

By analyzing eqs. (8) and (9), which show that the vapor injection system increases the exhaust mass-flow of the compressor, however, the enthalpy difference between the inlet and the outlet in the condenser is reduced, so the change of heat production is difficult to be intuitively judged from the enthalpy map, this is because its exhaust temperature is lower than that without the vapor injection system.

According to the First law of thermodynamics, the heat is equal to the sum of the refrigerating capacity and the power of the compressor. As the system refrigeration capacity and the power consumption of the compressor are increased, the heat production will increase, and the added value ΔQ_r is $\dot{m}_i$ ($h_2 - h_8$) + $\dot{m}_s$ ($h_{5'} - h_5$).

– The refrigeration efficiency ratio is given by:

$$EER = \frac{Q_l}{W} = \frac{\dot{m}_s(h_1 - h_5)}{\dot{m}_s(h_7 - h_1) + \dot{m}_d(h_2 - h_8)} \tag{10}$$

where EER is the refrigeration energy efficiency ratio of the medium pressure air supply system.

Compared with that without the vapor injection system, the refrigeration efficiency ratio of the vapor injection system is given by:

$$\begin{aligned}\Delta EER &= EER - EER' \\ &= \frac{\dot{m}_s(h_1 - h_5)}{\dot{m}_s(h_7 - h_1) + \dot{m}_d(h_2 - h_8)} - \frac{\dot{m}_s(h_1 - h_{5'})}{\dot{m}_s(h_{2'} - h_1)} \\ &= \frac{\dot{m}_i[(h_6 - h_{3'}) - (h_2 - h_8)EER']}{\dot{m}_s(h_{2'} - h_1) + \dot{m}_i(h_2 - h_8)}\end{aligned} \tag{11}$$

where ΔEER is increased refrigeration efficiency ratio for medium pressure air supply system and EER – the refrigeration efficiency ratio of the without a vapor injection system.

– The thermal performance coefficient is given by:

$$COP = \frac{Q_r}{W} = \frac{\dot{m}_s(h_1 - h_5)}{\dot{m}_s(h_7 - h_1) + \dot{m}_d(h_2 - h_8)} \tag{12}$$

where COP is the thermal performance coefficient of the medium pressure air supply system.

Compared with that without the vapor injection system, the thermal performance coefficient of the vapor injection system is given by:

$$\begin{aligned}\Delta COP &= COP - COP' \\ &= (EER + 1) - (EER' + 1) \\ &= \frac{\dot{m}_i\left[(h_6 - h_{3'}) - (h_2 - h_8)EER'\right]}{\dot{m}_s(h_{2'} - h_1) + \dot{m}_i(h_2 - h_8)}\end{aligned} \tag{13}$$

where ΔCOP is the increased thermal performance coefficient for medium pressure air supply system and COP' – the thermal performance coefficient of the without a vapor injection system.

Equations (10)-(13) show that the vapor injection technology increases the refrigerating capacity, the heating capacity, and the power of the system as well. The system's cooling efficiency ratio and the heating performance coefficient increase, which depend on the difference between ($h_6 - h_{3'}$) and ($h_2 - h_8$) EER'. Only when the difference is greater than zero, the cooling energy efficiency of the vapor injection technology and the heating performance coefficient will increase, but the value of EER' for the vapor injection system under extreme conditions is relatively low. Now the vapor injection technology will improve EER and COP of the system significantly.

Experimental research on the performance of the vapor injection technology

Based on the theoretical process analysis of the vapor injection technology, a performance test table was set-up. The data collection points of the experimental table are shown in fig. 2. The experimental testing process was conducted in the standard enthalpy difference laboratory. According to QC-T656-2000 standard for *Performance Requirements of Automobile Air conditioning Refrigeration Unit*, QC-T657-2000 standard for *Test method of Automobile Air conditioning Refrigeration Unit*, GBT12782-2007 standard for *Performance Requirements and Test Methods for Automobile Heating*, GBT21361-2008 standard for *Automotive Air Conditioner* and GB7725-2004 standard for *Room Air Conditioner*. In the process of the experiment, the compressor speed was set to 2000 rpm, 3000 rpm, 4000 rpm, and 5000 rpm. The low temperature environmental conditions of the test were given as: the dry bulb temperature outside the vehicle was –20 ℃, the dry bulb temperature inside the vehicle was 20 ℃, and the wet bulb temperature was 15 ℃.

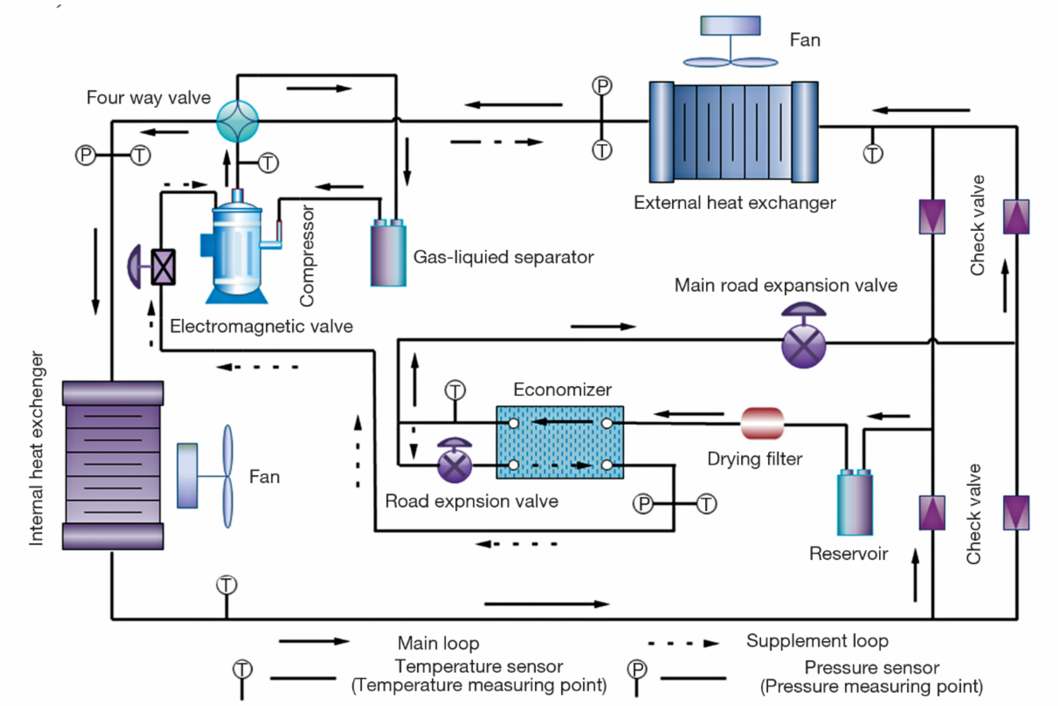

Figure 2. The data collection points of the experimental table

Analysis of experimental results

Figure 3 showed the effect of the vapor injection technology on the discharge temperature of the compressor with the change of the compressor speed. From fig. 3, the discharge temperature of the compressor gradually increased as the compressor speed increased, and at the same operating condition, the discharge temperature of the non-vapor injection was greater than the discharge temperature of vapor injection, and the discharge temperature difference increased gradually. Especially the discharge temperature decreased by 14.7% for the ultra-low temperature heating operation condition, the high discharge temperature of vapor injection (116.7 ℃), the high discharge temperature of vapor injection (99.6 ℃), and the high compressor speed (5000 rpm).

Figure 4 showed the influence of the vapor injection technology on the compressor power with the change of the compressor speed. Figure 4 showed that with the improvement of the compressor speed, the compressor power increased, and at the same operating conditions, the compressor power of the vapor injection was higher than that for the non-vapor injection, and the difference of the two rates gradually decreased. In the heating operation condition of ultra-low temperature, when the compressor speed reached 2000 rpm, the compressor power of the vapor injection was 12.7% higher than that of the non-vapor injection, but when the compressor speed was as high as 5000 rpm, the compressor power of the vapor injection was just 5.2% higher than that of the non-vapor injection.

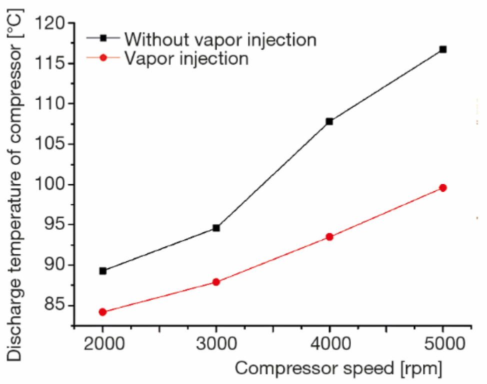

Figure 3. Variation of exhaust temperature of compressor with compressor speed

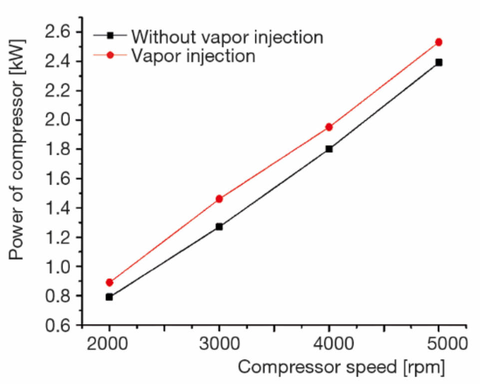

Figure 4. Variation of compressor power of system with compressor speed

Figure 5 showed the influence of the vapor injection technology on the heating capacity of the system with the change of the compressor speed. Figure 5 showed that at the ultra-low temperature heating operation condition, with the improvement of the compressor speed, the heating capacity of the system increased gradually, and in the same working condition parameter point, the heating capacity of the non-vapor injection system is less than that of the vapor injection system, and the difference value increased gradually, when the compressor speed increased from 2000 rpm to 5000 rpm, the heating capacity of vapor injection system increased by more 16.2%-22.7% than that of the non-vapor injection system. It can be seen that the vapor injection technology has significantly reduced the heat decrement of the heat pump air conditioning under the low temperature environment.

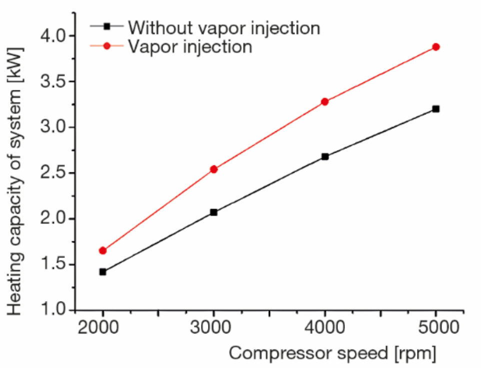

Figure 5. Variation of heating capacity of system with compressor speed

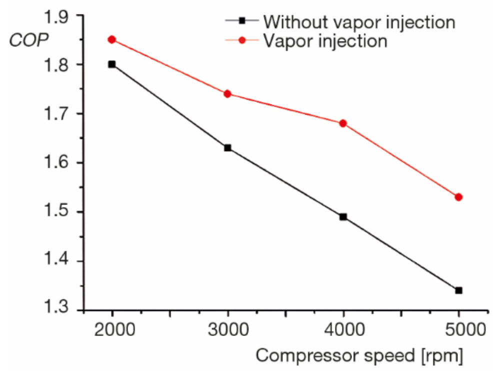

Figure 6. Variation of system *COP* with compressor speed

Figure 6 showed the influence of the vapor injection technology on the heating performance coefficient, *COP*, with the change of the compressor speed. Figure 6 showed that at the ultra-low temperature heating operation condition, with the improvement of compressor speed, the heating performance coefficient, *COP*, decreased gradually, and in the same working condition parameter point, the *COP* of the vapor injection system was more than that for the non-vapor injection system. With the increase of compressor speed, the difference value increased gradually. When the compressor speed increased from 2000 rpm to 5000 rpm, the COP of the vapor injection system increased by 2.8%-14.2%.

Conclusion

We can make the following important conclusions.

- Compared with the non-vapor injection heat pump air conditioners [16, 17], the vapor injection technology can significantly reduce the compressor discharge temperature, making the system safe and reliable operation.
- In the ultra-low temperature heating operation conditions, compared with the non-vapor injection heat pump air conditioners, when the compressor speed increased from 2000 rpm to 5000 rpm, the heating capacity of the vapor injection system increased by 16.2%-22.7%.
- In ultra-low temperature heating operating conditions, compared with the non-vapor injection heat pump air conditioners, when the compressor speed increased from 2000 rpm to 5000 rpm, the heating performance coefficient COP of the vapor injection increased by 2.8%-14.2%. An optimal condition [18, 19] can be obtained theoretically, which will be carried out in the future.

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References

[1] Dutta, A., *et al.,* An Investigation of the Performance of a Scroll Compressor Under Liquid Refrigerant Injection, *International Journal of Refrigeration, 24* (2001), 6, pp. 577-587
[2] Ying, X., *et al.,* Influence of Suction Jet on Scroll Compressor and System Performance, *Journal of Refrigeration, 36* (2015), 5, pp. 10-15
[3] Fan, B., Wang, H., Application and Theoretical Analysis of Spray Cooling in High Temperature Air Conditioning, *Refrigeration and Air-conditioning, 25* (2011), 2, pp. 175-178
[4] Jia, Q. L., *et al.,* Experimental Study on Improving the Performance of Air Source Heat Pump in High Temperature Conditions in Summer, *Refrigeration and Air Conditioning, 14* (2014), 8, pp. 113-118
[5] Fei, J. Y., *et al.,* Effect of Suction Jet on Performance of Air Source Heat Pump Water Heater, *Journal of Xi'an Jiaotong University, 42* (2008), 7, pp. 818-822
[6] Cui, S. Q., *et al.,* Experimental Research on the Application of Low Pressure Vapor Injection Technology to The Maximum Operating Refrigeration Conditions of Passenger Air-Conditioning, *Cryogenic and Superconductivity, 44* (2016), 10, pp. 72-76
[7] He, Y. G., *et al.,* Experimental Research on the Application of Supplemental Gas Technology in High Temperature Heat Pump, *Journal of Xi'an Jiaotong University, 49* (2015), 6, pp. 103-108
[8] Xu, X., Wang, Y. H., Refrigerant Injection for Heat Pumping/Air Conditioning Systems: Literature Review and challenges discussions, *International Journal of Refrigeration*, 34(2010), 2, pp. 402-415
[9] Heo, J., *et al.* Comparison of the Heating Performance of Air-Source Heat Pumps Using Various Types of Refrigerant Injection, *International Journal of Refrigeration, 34* (2011), 2, pp. 444-453

[10] Hu, W. J., *et al.*, Research on Flash Steam Cooling Technology and R134a for High Temperature Air Conditioners, *Fluid Machinery*, *43* (2015), 10, pp. 73-78
[11] Yin, H. Y., Experimental Research and Analysis of Low temperature Heat Pump Air Conditioning System for Electric Vehicles (in Chinese), *Refrigeration and Air-Conditioning*, (2016), 7, pp. 73-77
[12] Chai, Y. P., *et al.*, Experimental Study of R134a Quasi-Secondary Compression Refrigeration/Heat Pump System with Flash Tank Air-Injection (in Chinese), *Journal of Refrigeration*, (2017), 2, pp. 11-16
[13] Tang, J. C., *et al.*, Experimental Study on Quasi-Double Stage Compression Heat Pump Performance of Electric Vehicle Air-conditioner Using Scroll Compressor, *Journal of Refrigeration*, *39* (2018), 1, pp. 134-139
[14] Zhang, J. F., *et al.*, Intermediate Replenishment Technology of Scroll Compressor, *Refrigeration and Air-conditioning*, *12* (2012), 2, pp. 22-24
[15] Wu, Y. Z., *Refrigeration Principles and Equipment* (3rd Edition), Xi'an Jiaotong University Press, Xi'an, China, 2010, pp. 83-90
[16] Wang, X., *et al.*, The Modelling and Energy Efficiency Analysis of Thermal Energy Management Operation of Ground Source Heat Pump Air-Conditioning System, *Thermal Science*, *24* (2020), 5, pp. 3229-3237
[17] Fu, C., Zhao, X., Study on Energy Efficiency Evaluation and Influencing Factors of Ground Source Thermal Energy Management System Operation, *Thermal Science*, *24* (2020), 5, pp. 3319-2237
[18] He, J. H., A New Proof of the Dual Optimization Problem and Its Application to the Optimal Material Distribution of SiC/Graphene Composite, *Reports in Mechanical Engineering*, *1* (2020), 1, pp. 187-191
[19] Liu, X. Y., *et al.*, Optimization of a Fractal Electrode-Level Charge Transport Model, *Thermal Science*, *25* (2021), 3, pp. 2213-2220

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