EXPERIMENTAL STUDY ON PERFORMANCE OF MEDIUM-PRESSURE AIR-SUPPLY HEAT PUMP AIR CONDITIONING SYSTEM FOR PURE ELECTRIC BUS

by

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In order to solve the problem of serious attenuation of the performance of heat pump air conditioning system for pure electric bus at high temperature in summer and at low temperature in winter, a heat pump air conditioning system for medium-pressure air-supply pure electric bus with economizer is developed. The performance of the system working at from +50 °C to –20 °C ambient temperature and at different compressor speed is studied. The results show that the exhaust temperature of the compressor can be reduced greatly.

Key words: *heat pump air conditioning, medium pressure air supply, system performance, pure electric bus*

Introduction

The fast development of new energy vehicles is extremely important to alleviate the pressure of oil shortage, to reduce automobile exhaust emissions and to realize the sustainable development of automobile industry, among all new energy vehicles, the pure electric bus plays an important role in future [1, 2]. As an important part of pure electric bus, the heat pump air conditioning system directly affects its travelling mileage [3, 4]. Hosoz *et al.* [5] converted traditional automobile air-conditioning systems into heat pump systems, but the heating capacity was poor and an auxiliary heating had to be used. Xie *et al.* [6] compared the performance of electric vehicle heat pump air conditioning systems using different working fluids and compressors, and proposed several design solutions that are suitable for China's electric vehicle heat pump air conditioning systems. Li *et al.* [7] designed a vapor compression cooling and heating dual-mode heat pump air-conditioning system for pure electric vehicles, but the system was greatly affected by the external environment. Under harsh winter conditions, the heating capacity of the system can not be guaranteed.

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 heat pump air conditioning system are too high, the viscosity of lubricating oil is reduced, and the performance of the system is attenuated, which seriously affects the mileage and application range of pure

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electric bus. In view of the prominent problems of heat pump air conditioning system at the high temperature in summer and at the low temperature in winter, there are three solutions in open literature: two-stage compression cycle heat pump technology, cascade cycle heat pump technology, and quasi-two-stage compression cycle heat pump technology [8-10]. Both the two-stage compression cycle heat pump system and the cascade cycle heat pump system need two compressors, which not only increases the cost of the heat pump air conditioning system, but also is not conducive to the lightweight development of the heat pump air conditioning system. However, the quasi-two-stage compression cycle heat pump system adds an air-supply loop to the ordinary heat pump cycle, and the added components are less, so it is more suitable for the heat pump air conditioning system of pure electric bus. Through the air-supply technology, the exhaust temperature of the compressor is reduced and the performance of the heat pump system is improved [11].Wang *et al.* [12] used R410A as refrigerant to study the effect of air-supply technology on the performance of heat pump air conditioning system. Dutta *et al.* [13] carried out an experimental study on the quasi-two-stage compression cycle characteristics of scroll compressors, and Tang *et al.* [14] carried out an experimental study on the application of air supply technology to the air conditioning system of pure electric vehicles. Li *et al.* [15] simulated the heat transfer performance of evaporator in the air-supply system.

In this paper, R410A is used as refrigerant, and a micro-channel parallel flow heat exchanger is used in both internal and external heat exchangers, the performance of heat pump air conditioning system for pure electric bus under extreme conditions of ultra-high temperature in summer and of ultra-low temperature in winter is studied by using medium-pressure air-supply technology with economizer.

Medium-pressure air-supply heat pump air conditioning system for pure electric bus

The principle of the medium-pressure air-supply heat pump air conditioning system for pure electric bus is shown in fig. 1. Figure 2 shows the theoretical cycle principle of medium-pressure air supply on the pressure enthalpy diagram.

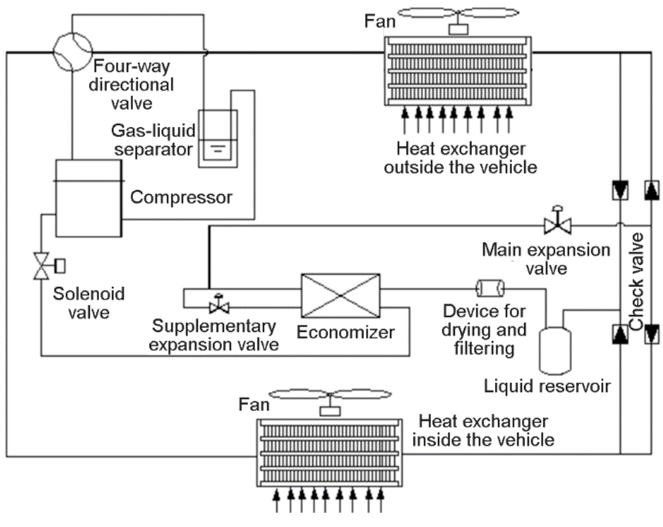

Figure 1. Schematic diagram of medium-pressure air-supply heat pump air conditioning system for pure electric bus

The calculation formula of thermodynamic cycle is as follows:

– Heating capacity of heat pump system at the condenser side:

$$Q_\mathrm{h} = m_\mathrm{r}(h_2 - h_3)$$

– Compressor power:

$$W = m_o(h_7 - h_1) + m_\mathrm{r}(h_2 - h_8)$$

– Refrigerating capacity of heat pump system at the evaporator side:

$$Q_\mathrm{c} = m_o(h_1 - h_5)$$

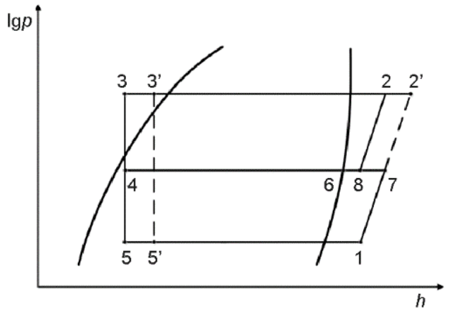

Figure 2. Representation of the cycle principle of medium-pressure air supply on pressure enthalpy diagram

– Refrigeration coefficient of heat pump system:

$$COP_\mathrm{c} = \frac{Q_\mathrm{c}}{W}$$

– Heating coefficient of heat pump system:

$$COP_\mathrm{h} = \frac{Q_\mathrm{h}}{W} = \frac{W + Q_\mathrm{c}}{W} = 1 + \frac{Q_\mathrm{c}}{W}$$

– Mass-flow rate of refrigerants for air supply:

$$\dot{m}' = \dot{m}_\mathrm{r} - \dot{m}_o$$

– Heat exchange capacity of economizer:

$$Q' = \dot{m}_o(h_{5'} - h_5)$$

where $\dot{m}_o$ [kgs^{-1}] is the mass-flow rate of refrigerants entering evaporator, $\dot{m}_\mathrm{r}$ [kgs^{-1}] – the mass-flow rate of compressor exhaust refrigerants, $\dot{m}'$ [kgs^{-1}] – the mass-flow rate of air-supply refrigerants.

Experimental process

According to the principle of quasi-two-stage compression cycle and the characteristics of air conditioning structure of pure electric bus, the experimental platform is built. The experiment is carried out in the standard enthalpy difference laboratory. According to QC-T656-2000 *Performance Requirements of Automobile Air conditioning Refrigeration Unit*, QC-T657-2000 *Test method of Automobile Air conditioning Refrigeration Unit*, GBT12782-2007 *Performance Requirements and Test Methods for Automobile Heating*, GBT21361-2008 *Automotive Air Conditioner,* and GB7725-2004 *Room Air Conditioner,* and other national and industry standards. In the process of the experiment, the compressor speed was set to 2000, 3000, 4000, and 5000 rpm. The high temperature environment conditions of the test were: the dry bulb temperature outside the vehicle was 50 °C, the wet bulb temperature was 36 °C, the dry inside the vehicle. The bulb temperature is 32.5 °C and the wet bulb temperature is 28 °C. The low temperature environmental conditions of the test are: the dry bulb temperature outside the vehicle is –20 °C, the dry bulb temperature inside the vehicle is 20 °C, and the wet bulb temperature is 15 °C. The charge of refrigerant R410A in heat pump air conditioning system is 10.81 kg.

Analysis of experimental results

Figure 3 is a curve graph of the exhaust temperature of compressor changing with the compressor speed. As seen from fig. 3, with the increase of the rotation speed of the compressor, the exhaust temperature of the compressor gradually increases, and at the same operating condition parameter point, the exhaust temperature of the non-air-supply system is greater than that of the medium-pressure air supply system. The exhaust temperature of the medium-pressure air supply system is 82.4 °C, falling by 15.8 °C. Figure 4 is a curve graph of the refrigeration capacity of system changing with the compressor speed. As seen from fig. 4, with the increase of the rotation speed of the compressor, the refrigeration capacity of the system gradually increases, and at the same operating condition parameter point, the refrigeration capacity of the non-air-supply system is less than that of the medium-pressure air supply system, When the rotation speed of the compressor is increased from 2000-5000 rpm, the refrigeration capacity of the medium-pressure air-supply system is increased by 1.4-6.5% compared with that of the non-air-supply system.

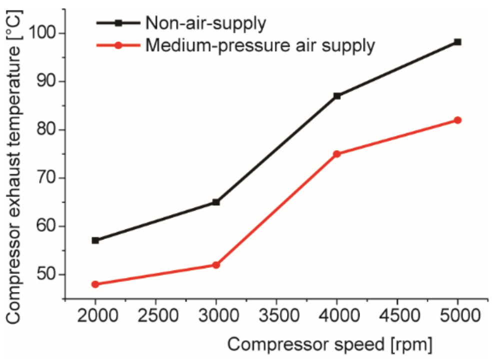

Figure 3. Variation of exhaust temperature of compressor with compressor speed

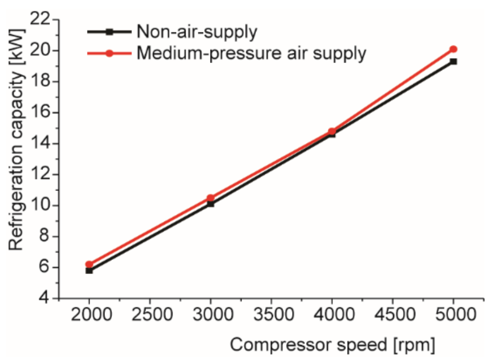

Figure 4. Variation of refrigeration capacity of system with compressor speed

Figures 5 and 6 show the curve graphs of system compressor power and system COP_c changing with the compressor speed, respectively. It can be seen from figs. 5 and 6 that with the increase of compressor speed, the compressor power of the system increases gradually, and the COP_c of the system decreases gradually, and at the same working condition parameter point, the compressor power and COP_c of the non-air-supply system are lower than that of the medium-pressure air supply system. When the compressor speed is increased from 2000 rpm to 5000 rpm, the compressor power of the medium pressure air supply system is 0.7~3.4% higher than that of the non-air-supply system, and the COP_c of the medium pressure air supply system is 0.3~3.3% higher than that of the non-air-supply system.

Figure 7 shows the curve diagram of the exhaust temperature of the compressor changing with the compressor speed. It can be seen from fig. 7 that with the increase of compressor speed, the exhaust temperature of compressor increases gradually, and at the same working condition parameter point, the exhaust temperature of non-air-supply system is higher than that of medium pressure air supply system, especially when the compressor speed is

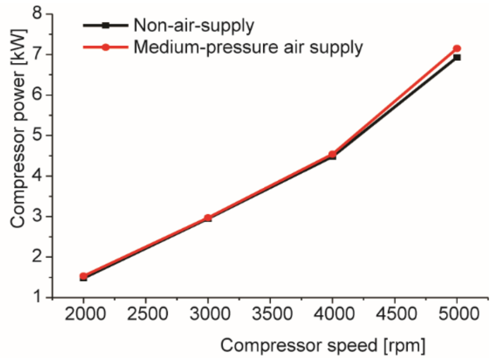

Figure 5. Variation of compressor power of system with compressor speed

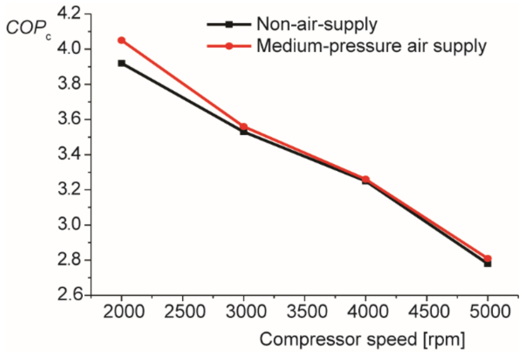

Figure 6. Variation of system COP_c with compressor speed

5000 rpm, the exhaust temperature of the non-air-supply system is up to 113.4 °C, while that of the medium pressure air supply system is 96.2 °C, falling by 17.2 °C than that of the medium pressure air supply system. Figure 8 shows the curve graph of heating capacity of the system changing with the speed of the compressor. It can be seen from fig. 8 that with the increase of compressor speed, the heating capacity of the system increases gradually, and at the same working condition parameter point, the heating capacity of the non-air-supply system is smaller than that of the medium pressure air-supply system, and when the compressor speed increases from 2000-5000 rpm, the heating capacity of medium-pressure air-supply system is 10.1~14.3% higher than that of non-air-supply system.

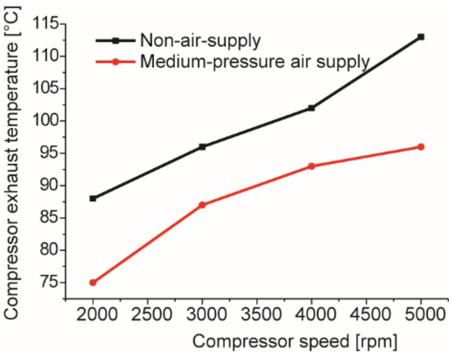

Figure 7. Variation of exhaust temperature of compressor with compressor speed

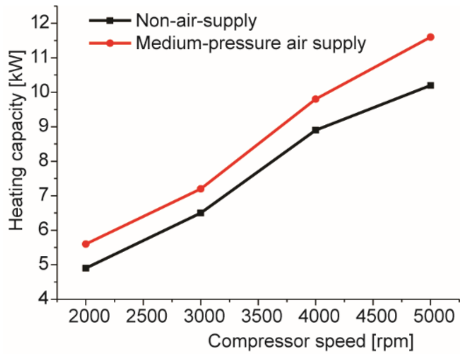

Figure 8. Variation of heating capacity of system with compressor speed

Figures 9 and 10 show the curve graphs of system compressor power and system COP_h changing with compressor speed, respectively. It can be seen from figs. 9 and 10 that with the increase of compressor speed, the compressor power of the system increases gradually, and the COP_h of the system decreases gradually, and at the same working condition parameter point, the compressor power and COP_h of the non-air-supply system are lower than those of the medium pressure air-supply system. When the compressor speed is increased from 2000-5000 rpm, the compressor power of the medium pressure air-supply system is 1.3~5.8% higher than that of the non-air-supply system, and the COP_h of the medium pressure air-supply system is 8.0~9.5% higher than that of the non-air-supply system.

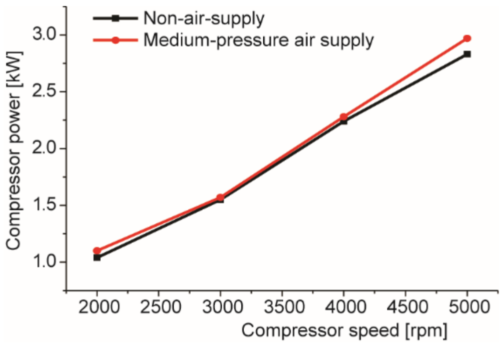

Figure 9. Variation of compressor power with compressor speed

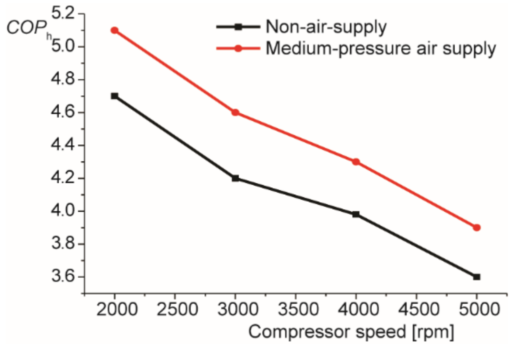

Figure 10. Variation of system COP_h with compressor speed

Conclusion

- In the extreme conditions of 50 °C and –20 °C, compared with the non-air-supply system, the medium-pressure air-supply technology can obviously reduce the exhaust temperature of the compressor.
- When the rotation speed of the compressor is increased from 2000-5000 rpm, compared with the non-air-supply system, the refrigerating capacity of the medium-pressure air-supply system is increased by 1.4~6.5%, and the compressor power is increased by 0.7~3.4%. The COP_c is increased by 0.3~3.3%.
- In the ultra-low temperature environment of –20 °C, when the compressor speed is increased from 2000-5000 rpm, compared with the non-air-supply system, the heating capacity of the medium-pressure air-supply system is increased by 10.1~14.3%, and the compressor power is increased by 1.3~5.8%. The COP_h is increased by 8.0~9.5%.

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