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ORIGINAL RESEARCH ARTICLE

Simulation of CO_2 heat pump air conditioning system for new energy vehicle

Yefeng Liu[1,2*], Yuqing Wang[1], Danping Tang[1]

[1] *School of Energy and Power Engineering, University of Shanghai for Science and Technology, Shanghai 200093, China.*

[2] *Shanghai Key Laboratory of Multiphase Flow and Heat Transfer in Power Engineering, Shanghai 200093, China.*
E-mail: yfliu209@163.com

ABSTRACT

The CO_2 heat pump air conditioning system of new energy vehicle is designed, and the vehicle model of CO_2 heat pump module and heat management system is established based on KULI simulation. The effects of refrigerant charge, running time and compressor speed on the heat pump air conditioning system is studied, and the energy consumption is compared with the PTC heating system and the CO_2 heat pump air conditioning system without waste heat recovery. The results show that the optimal charge for full-service operation is 750 g; increasing the compressor speed can increase the cooling capacity, so that the refrigerant temperature in the passenger compartment and battery inlet can quickly reach the appropriate temperature, but the COP_h, COP_c are reduced by 2.5% and 1.8% respectively. By comparing it with PTC heating and CO_2 heat pump air conditioning systems without waste heat recovery, it is found that the energy consumption of this system is only for the PTC heating systems 42.5%, without waste heat recovery carbon dioxide heat pump air conditioning system of 86.6%. It greatly saves energy, but also increased the waste heat recovery function, so that the system supply air temperature increased by 26%, improving passenger cabin comfort. This provides a reference for the future experimental research of CO_2 heat pump air conditioning and heat management system.

Keywords: New Energy Vehicles; CO_2 Heat Pump Air Conditioning; Thermal Management System; Energy Consumption Comparison; KULI Simulation

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1. Introduction

The thermal management of new energy vehicles is mainly divided into air conditioning systems and thermal management systems according to demand. The air conditioning system includes two parts: refrigeration and heating, which are responsible for the comfort of the passenger compartment; the thermal management system includes battery thermal management, motor electronic control thermal management and other equipment cooling, which is responsible for the safe travel of the car. Air conditioning systems can be divided into 3 categories according to the heating method: first, the air conditioning system using PTC (Positive Temperature Coefficient) electric heater heating, PTC point heating method can be divided into water heating PTC heating and air heating PTC heating two kinds, of which plumbing PTC electric heater can follow the air conditioning system structure of fuel vehicles, and the application is more common[1]. The second is the use of waste heat recovery heating air conditioning system, the drive motor, power battery generated heat recovery to the passenger compartment heating, in order to save energy, improve the mileage; the third is the use of heat pump heating air conditioning system[2], heat pump air conditioning system on the basis of household air conditioning developed, but its operating conditions are worse, automotive air conditioning needs to run under different working conditions throughout the year, ambient temperature, light intensity, driving speed, Many factors, such as body structure, can greatly affect the operating characteristics of automotive air conditioners[3]. To this end, many domestic and foreign car companies such as Tesla[4], BMW[5], BYD, etc. are vigorously studying heat pump air conditioning and making practical progress, successfully applying it to new energy vehicles.

CO_2 as a kind of environmental friendly refrigerant, its ODP = 0, GWP = 1, non-toxic and non-flammable, low price, excellent thermo-dynamic properties, high safety level, making it used in automotive air conditioning has great advantages. As early as 2003, Deson[6] researched and developed a set of CO_2 automotive heat pump air conditioners on the basis of the original R134a automotive air conditioner, and built an experimental test bench to study its heating performance, and the test results showed that the heating effect of CO_2 automotive heat pump air conditioners was 30% higher than that of the R134a system. Tamura *et al.*[7] designed a CO_2 heat pump air conditioning system for commercial vehicles and added waste heat recovery to serve as an auxiliary heat source during the dehumidification process. Comparing the experimental method with the R134a system, the results show that under the same conditions, the COP_h of the system is 1.13 times that of the R134a system after using waste heat, and Chen *et al.*[8] developed a set of CO_2 The automotive air conditioning system, and through the simulation of its performance transient and steady-state simulation, laid the foundation for the domestic CO_2 automotive air conditioning research; Liu *et al.*[9] designed and built a CO_2 automotive air conditioner and conducted detailed experimental research on it, and obtained the parameters of wind speed, exhaust pressure, compressor speed and so on The influence of COP obtains the operating characteristic curve of CO_2 heat pump air conditioner; Wang *et al.*[10] designed an energy-efficient CO_2 heat pump air conditioning system using three heat exchangers in view of the problem that the heat pump heating effect is not significant in low temperature environment. The results of the study show that the heating COP of the system in the low temperature environment can reach 3.15, which has a very good heating effect.

On the basis of the predecessors, this paper establishes a set of pure electric car CO_2 heat pump module and heat management system through KULI13 software, mainly including compressor, expansion valve, heat exchanger, passenger compartment, battery motor and vehicle heat management model, and studies and analyzes the performance of passenger compartment in cooling, heating and other working conditions when the system is applied to new energy vehicles, which is the CO_2 experimental study of heat pump air conditioning and thermal management system provides a reference basis.

2. Modeling of CO_2 heat pump module and thermal management system

First, according to the manufacturer's different pressure ratio of the compressor volume efficiency and isoentropic efficiency and related literature fitting the available volumetric efficiency and isoentropic efficiency calculation formula is as follows[11-13], this can establish a more accurate compressor model.

$$\eta_v = \frac{q_m}{\rho_1 v_h n} \tag{1}$$

n_v—volume efficiency; q_m—mass flowrate, kg/s; p_1—inhalation density, kg/m^3; v_h—theoretical displacement, m^3; n—compressor speed, r/min.

$$\eta_i = \frac{w_s}{w} \tag{2}$$

n_i—isoentropic efficiency; w_s—adiabatic power consumption of compressor, w; w—compressor power consumption.

The designed system mainly has two types of heat exchangers: plate type and microchannel parallel flow. Plate heat exchangers are used in CO_2 circuits, including air cooler and evaporators; microchannel parallel flow heat exchangers are used in refrigerant circuits, including indoor heat exchangers, cooler and outdoor heat exchanger. According to the test data provided by the heat exchanger manufacturer, the corresponding heat exchanger heat transfer model is established, and the Gnielinski heat transfer correlation[14] is selected. Based on parameters such as the dimensions of the target model and the thermal conductivity of each material, the passenger compartment of two rows and five seats will be modeled. Because the modules such as batteries and motors are more complex, they are simplified, and the battery and motor are regarded as mass points with internal heat sources, and they are given a fixed heat value. According to the design of the pure electric car CO_2 heat pump module and the heat management system of the two operating modes of refrigeration and heating, the simulation model of vehicle refrigeration and heating is established respectively. The model of the refrigerant side is shown in **Figure 1**, and the model of the refrigerant side of the refrigeration mode is shown in **Figure 2**. Due to the use of secondary circuits, the model of the refrigerant side is the same in the cooling and heating modes.

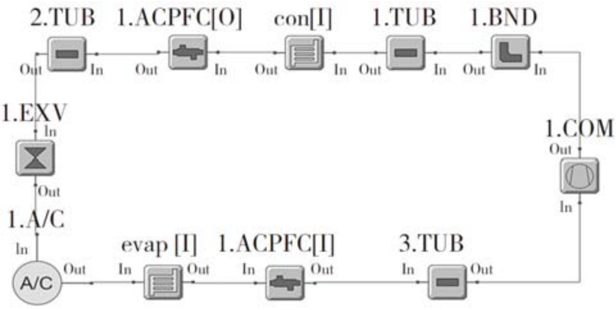

Figure 1. Refrigerant side model of CO_2 heat pump module and thermal management system.

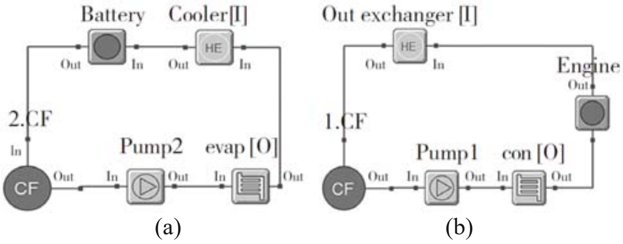

Figure 2. Model of refrigerant carrier side in refrigeration mode. Note: **(a)** Battery circuit; **(b)** Motor circuit.

3. Refrigerant charge amount is determined

The refrigerant is the handler of the entire system and is the most important key factor. Refrigerant charge can greatly affect the performance of the system, so there is an optimal charge for any system to optimize system performance. The current way of determining refrigerant charge can be divided into theoretical calculations and experimental verification, but both methods have their own limitations. The theoretical method requires a lot of calculation, and the calculation results are not accurate; the experimental law requires a lot of manpower and material resources, and the experimental process is cumbersome. Using the KULI simulation platform to determine refrigerant charge is fast and accurate, and is being adopted by more and more scholars. This paper first determines the optimal charge of refrigerant in refrigeration and heating mode, and then determines the optimal charge of refrigerant suitable for various working conditions throughout the year according to the crossover method.

3.1 Refrigeration mode

According to the national standard[15,16], determine the simulation working conditions of the refrigeration mode charge, as shown in **Table 1**.

Table 1. Refrigerant charge simulation conditions of cooling mode

Ambient temperature /°C	**Indoor air inlet temperature /°C**	**Compressor speed /(r/min)**	**Outdoor air volume /(m^3/h)**	**Indoor air volume /(m^3/h)**
35	27	4,740	1,400	450

As shown in **Figure 3**, as the refrigerant charge increases, the cooling capacity growth rate increases first and then decreases, and the maximum value occurs at the charge of 800 g. This is because as the charge increases, the refrigerant mass flow will also increase, and the evaporation temperature will increase, so that the enthalpy value of the evaporator inlet and outlet increases plus, so the cooling capacity growth rate increases. When the charge amount is further increased, the evaporation temperature continues to increase so that the heat exchange temperature difference between the refrigerant and the carrier refrigerant decreases, and the heat exchange temperature difference dominates, so the heat exchange effect is reduced and the cooling capacity growth rate is reduced. Similarly, the growth rate of compressor power consumption has also increased first and then decreased. Because the refrigerant increases at the beginning, the exhaust pressure rises rapidly, while the inspiratory pressure changes relatively little, and the mass flow of the refrigerant also increases, so the compressor power consumption growth rate is faster. Subsequently, the exhaust pressure tends to be stable, and the exhaust temperature decreases, so that the unit power consumption of the compressor is reduced, while the mass flow of refrigerant continues to increase, and the combination of the two makes the compressor power consumption growth rate slower. Therefore, as the refrigerant charge increases, the COP_c shows a tendency to increase first and decrease later, with a maximum value of 2.21 at 800 g. It can be seen that the system's refrigerant is optimal in summer cooling mode, its charge volume is about 800 g.

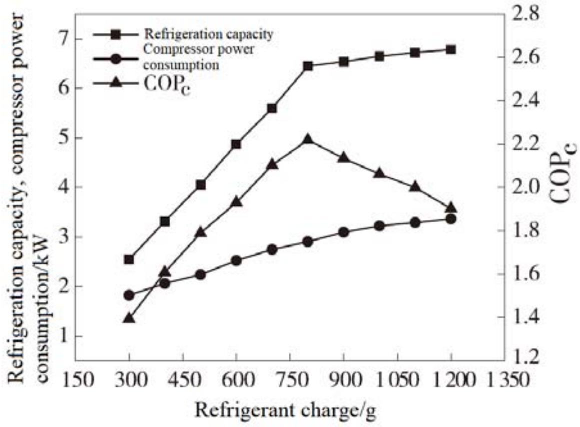

Figure 3. Effect of refrigerant charge on cooling capacity, compressor power and COP_c.

3.2 Heating mode

The charge simulation scenario in heating mode is shown in **Table 2**.

Table 2. Refrigerant charge simulation conditions of heating mode

Ambient temperature /°C	**Indoor air inlet temperature /°C**	**Compressor speed /(r/min)**	**Outdoor air volume /(m^3/h)**	**Indoor air volume /(m^3/h)**
7	20	3,780	1,400	450

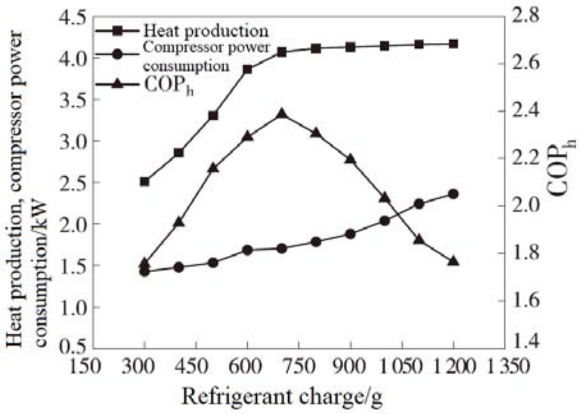

Figure 4. Effect of refrigerant charge on heating capacity, compressor power and COP_h.

Figure 4 shows that as the refrigerant charge increases, the heat growth rate increases first and then decreases. As the refrigerant charge increases, the exhaust pressure rises at the beginning, the enthalpy value of the compressor outlet increases, and the unit heat and mass flow increase so that the heat system increases. When the charge volume is further increased, the exhaust pressure tends to be stable and the exhaust temperature decreases, while the refrigerant charge is still increasing, and the combination of the two makes the heat slowly increase. Same as the refrigeration mode, with the increase of refrigerant charge, the compressor power consumption continues to increase and the growth rate increases first and then decreases, and the COP_h increases first and then decreases slowly, and when the recharge amount is 700 g, the maximum value of COP_h is 2.38. It can be seen that the optimal refrigerant charge of the system in winter heating mode is about 700 g.

In summary, the optimal recharge of refrigerant in refrigeration and heating mode is about 800 g and 700 g, respectively. In order to make the system run under full operating conditions and perform well, 750 g is selected as the system charge.

4. Refrigeration simulation

The common refrigeration operating conditions of automotive heat pump air conditioning systems[17,18] are selected, as shown in **Table 3**, and the refrigeration mode under this operating condition is simulated.

4.1 Relationship between system performance and runtime

Select the maximum load refrigeration working condition of the automobile air conditioner (ambient temperature 50 °C, outdoor heat exchanger oncoming wind speed of 4.5 m/s, indoor heat exchanger inlet air temperature of 50 °C, relative humidity of 26%, supply air volume of 450 m^3/h) for transient simulation in the refrigeration mode, and the relationship between system performance and operating time is shown in **Figure 5** and **Figure 6**.

Table 3. Simulation condition of refrigeration

Ambient temperature /°C	Indoor heat exchanger inlet air temperature/°C	Compressor speed /(r/min)	Outdoor air supply wind speed/(m/s)	Indoor air volume /(m^3/h)
50	50	2,500, 3,000, 3,500, 4,000	4.5	450

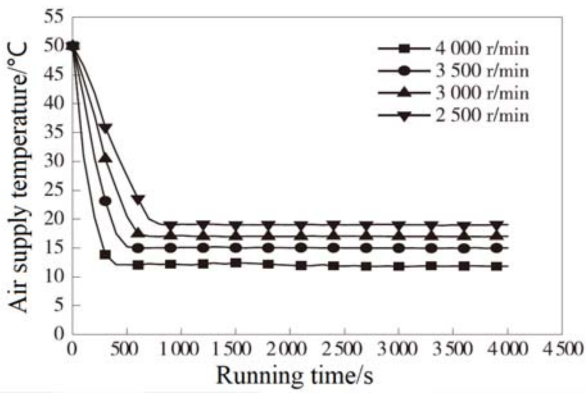

Figure 5. Effect of running time on air supply temperature.

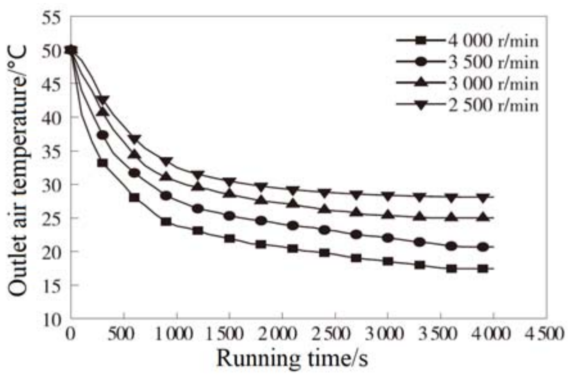

Figure 6. Effect of running time on air outlet temperature.

As shown in **Figure 5**, as the running time increases, the supply air temperature at compressor speeds of 2,500, 3,000, 3,500, 4,000 r/min rapidly decreases from the initial temperature of 50 °C and remains at a stable supply air temperature of 19.0, 17.0, 14.9, 11.6 °C, respectively, the time required to drop to a stable temperature is 800, 700, 600, 400 s, and the cooling rate is 0.038, 0.047, 0.058, 0.096 °C/(r/min), indicating that the faster the compressor speed, the lower the supply air temperature, the faster the cooling rate. This is because the faster the speed under the same conditions, the more cooling capacity, making the surrounding air easier to be cooled and cooled down, it is easier to get the required supply air temperature in a short period of time, and the cooling amplitude of the compressor speed from 3,500 to 4,000 is 3.3 °C, which is greater than the cooling amplitude of 2.0 °C when increasing from 2,500 to 3,000.

As shown in **Figure 6**, when the compressor speed is increased from 2,500 r/min to 4,000 r/min, the passenger compartment air outlet temperature drops rapidly from the initial temperature of 50 °C and eventually reaches stability, and when the running time is 500 s, the compressor speed increases from 2,500 r/min to 4,000 r/min, and the outlet temperature is 38.6, 36.3, 31.7, 28.0 °C respectively, and the cooling amplitudes are respectively 11.4, 13.7, 18.3, 22.0 °C, the cooling rates were 0.0228, 0.0274, 0.0366, 0.044 °C/s, the outlet air temperature at stable time was 28.1, 25.0, 20.6, 17.4 °C, and the cooling amplitude was 21.9, 25.0, 29.4, 32.6 °C respectively. This shows that the compressor speed has a greater impact on the temperature, that is, the larger the compressor speed, the faster the outlet temperature drop rate, and the lower the final stable outlet temperature. From the human body thermal comfort requirements[19] can be known that the summer temperature range is 20 ~ 26 °C, so the compressor speed is more than 3,000 r/min, the passenger compartment to meet the cooling requirements.

As shown in **Figure 7**, when the compressor speed is increased from 2,500 r/min to 4,000 r/min, the battery inlet refrigerant temperature (hereinafter referred to as the inlet temperature) drops rapidly and eventually reaches stability, and the stable temperature is 19.0, 16.0, 13.7, 11.4 °C respectively. This means that the higher the compressor speed, the lower the inlet temperature, and the easier it is for the inlet temperature to reach equilibrium.

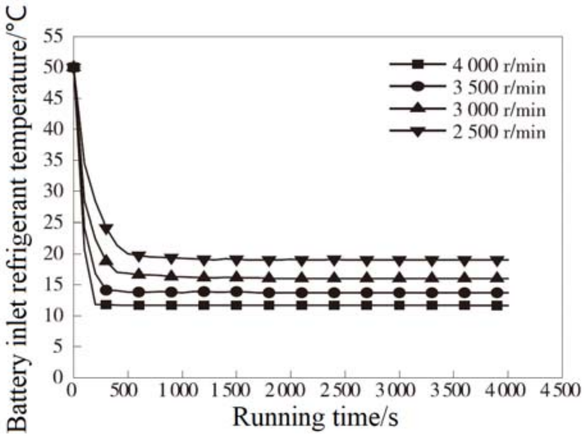

Figure 7. Effect of running time on temperature of refrigerant at battery inlet.

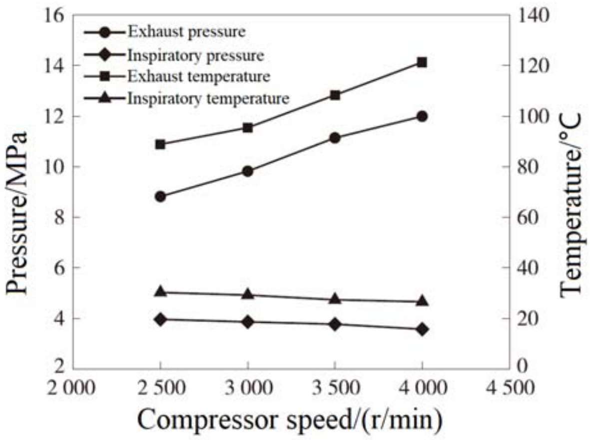

Figure 8. Effect of compressor speed on inlet and outlet pressure and temperature.

4.2 The relationship between system performance and compressor speed change

Under different compressor speeds, when the running time is 4,000 s, the status points of the system have reached a stable state, and the parameters under the running time point are recorded to obtain the relationship between the suction and exhaust pressure, suction and exhaust temperature, compressor power consumption, cooling capacity, COP_c, etc. with the compressor speed.

As shown in **Figure 8**, as the compressor speed increases, the compressor exhaust pressure and exhaust temperature continue to increase, the exhaust pressure increases from 8.8 MPa to 11.9 MPa, the increase rate is 2.06 kPa/(r/min), the exhaust temperature increases from 88.8 °C to 121.3 °C, the increase rate is 0.02 °C/(r/min); the suction pressure and inspiratory temperature are slowly reduced, the suction pressure is reduced from 3.9 MPa to 3.5 MPa, and the reduction rate is only 0.26 kPa/(r/min), the suction temperature drops from 30.2 °C to 26.5 °C at a rate of 0.002 °C/(r/min), much lower than the rate of increase in exhaust temperature. This is because the compressor speed increases, the system exhaust volume increases, resulting in an increase in mass flow, increasing the phase transformation heat in the evaporator, so that the evaporation temperature and superheat are reduced, so the suction temperature is reduced, and the temperature and pressure in the two-phase zone correspond one-to-one, so the evaporation pressure is reduced. At the same time, the compressor pressure ratio increases rapidly so that the exhaust pressure and exhaust temperature increase rapidly.

As shown in **Figure 9**, as the compressor speed increases, both the cooling capacity and compressor power consumption increase, while the COP_c decreases. At a speed of 2,500 r/min, the cooling capacity, compressor power consumption, and COP_c are 4.8 kW, 1.8 kW and 2.6, respectively. When the speed is increased to 4,000 r/min, the cooling capacity, compressor power consumption, and COP_c are 6.4 kW, 3.5 kW and 1.8 respectively. This is because the compressor speed increases the exhaust volume and refrigerant mass flow increase, and the unit refrigeration capacity does not change much, and the comprehensive refrigeration capacity increases; at the same time, the exhaust pressure and exhaust temperature are increasing rapidly, so that the compressor outlet enthalpy value increases, and the unit power consumption increases, so the compressor power consumption increases; but the increase rate of the refrigeration capacity is 0.73 W/(r/min), which is less than the increase rate of compressor power consumption 0.86 W/(r/min). Therefore, there will be a downward trend in COP_c.

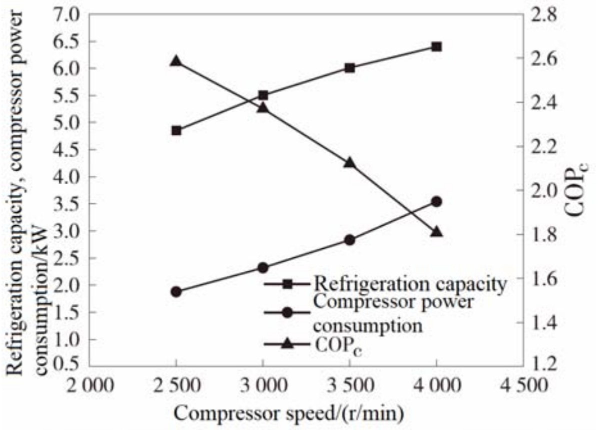

Figure 9. Effect of compressor speed on refrigerating capacity, compressor power and COPc**.**

5. Thermal simulation

5.1 Comparison with PTC heating systems

The system is evaluated from the perspective of energy consumption, and based on the research results of other existing scholars on PTC heating[20], compared with the traditional PTC heating system, the energy consumption at different ambient temperatures is obtained.

As can be seen from **Figure 10**, the energy consumption of PTC electric heating (assuming a heating efficiency of 100%) at four ambient temperatures (-20, -10, 0, 10 °C) is 3.55, 4.00, 4.25, 4.30 kW, respectively, and the energy consumption of heating with CO_2 heat pumps is 1.40, 1.69, 1.85, 1.93 kW, respectively. It is explained that PTC electric heating heating consumes more electricity than CO_2 heat pump heating. The use of CO_2 heat pump heating at four ambient temperatures (-20, -10, 0, 10 °C) saves 60.0%, 57.7%, 56.4% and 55.1% of electrical energy compared with PTC electric heating, respectively. The average electricity consumption of PTC electric heaters is 4.03 kW, and the average electricity consumption of CO_2 heat pumps is 1.71 kW. This means that the use of CO_2 heat pumps instead of PTC can save 2.30 kW of electrical energy and reduce heating energy consumption in the passenger compartment by 57.5%. It follows that heating with a CO_2 heat pump saves energy and improves system economy.

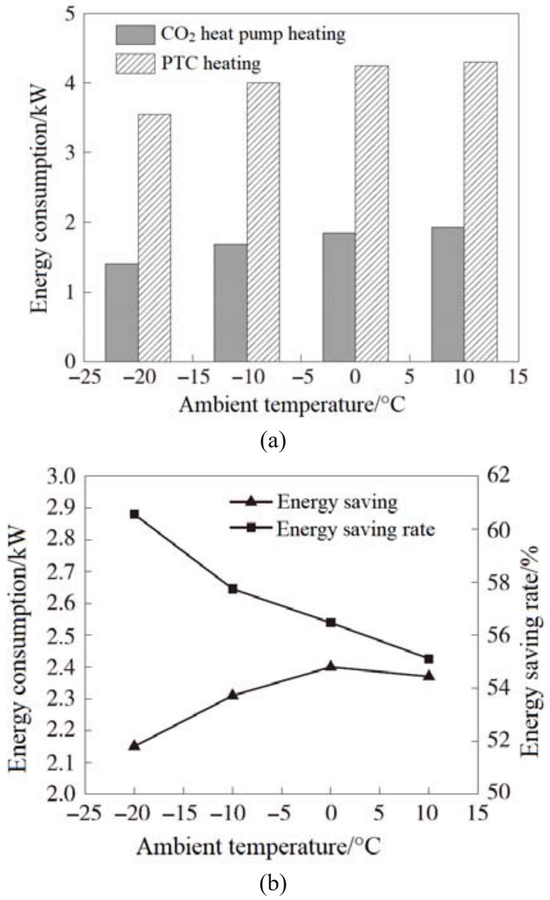

Figure 10. Comparison of energy consumption between PTC heating and CO_2 heat pump heating.5.5.2 Comparison with CO_2 heat pump systems without waste heat recovery**.**

Based on the test conditions in the literature[10] (as shown in **Table 4**), the CO_2 heat pump system that simulates the system performance at different ambient temperatures without waste heat recovery obtains a comparison of the supply air temperature of the two systems, as shown in **Figure 11**.

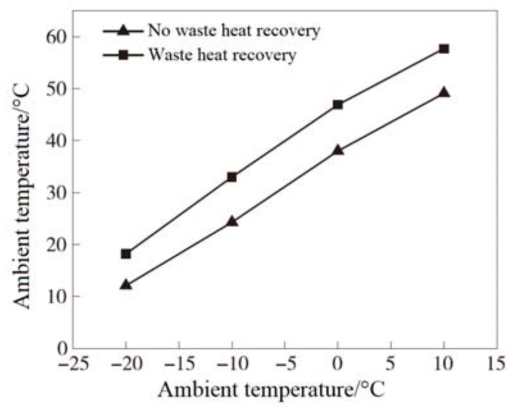

Figure 11. Comparison of air supply temperatures with and without waste heat recovery**.**

Table 4. Simulation condition of heating

Ambient temperature/°C	Indoor heat exchanger inlet air temperature/°C	Compressor speed /(r/min)	Outdoor air supply wind speed/(m/s)	Indoor air volume/(m^3/h)
-20, -10, 0, 10	-20, -10, 0, 10	3,900	4.5	450

As shown in **Figure 11**, after increasing the waste heat recovery function, the supply air temperature of the system under 4 ambient temperatures (-20, -10, 0, 10 °C) has been significantly improved, and the increase is 6.1, 8.7, 8.9, 8.6 °C, which is an average increase of 8.07 °C, and the lift rate is 26%, indicating that the increase in waste heat recovery can meet the supply air requirements faster, and also indicates that the waste heat recovery can be increased under certain harsh conditions to obtain the appropriate supply air temperature.

Compare the energy consumption of two systems when they reach the same supply air temperature from the same inlet temperature at different ambient temperatures, as shown in **Figure 12**.

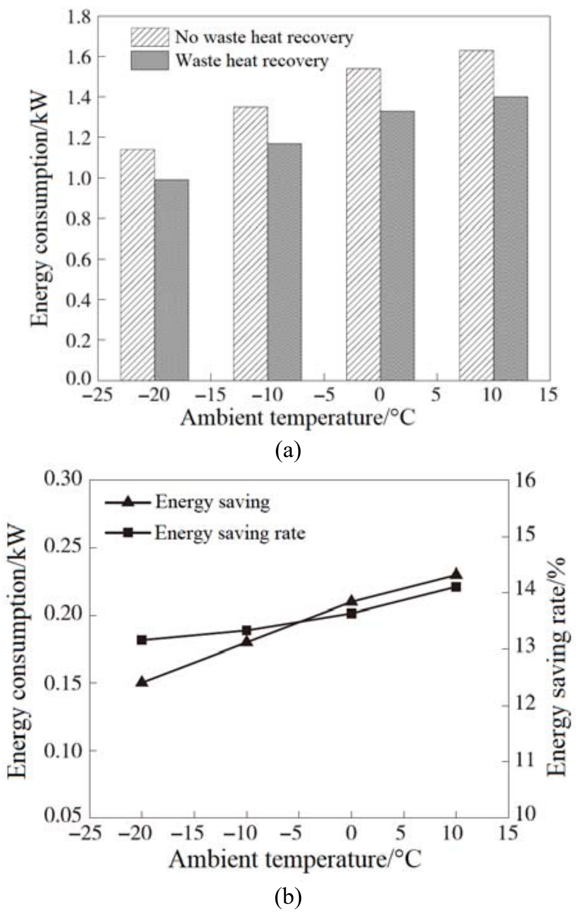

Figure 12. Comparison of heat production and compressor energy consumption with and without waste heat recovery.

As shown in **Figure 12**, four ambient temperatures (-20, -10.0, 10 °C), from the same inlet temperature (-20, -10.0, 10 °C) to the same supply air temperature (12.1, 17.2, 30.4, 41.2 °C), system compressor consumption with heat recovery The energies are 0.99, 1.17, 1.33, 1.40 kW respectively, lower than no CO_2 heat pump systems with heat recovery (1.14, 1.35, 1.54, 1.63 kW), 4 ambient temperatures (-20, -10, 10 °C). The heat recovery underneath is 13.1% less than that without heat recovery. 13.3%, 13.6%, 14.1% of electric energy with heat recovery, the average energy consumption of CO_2 heat pump system is 1.22 kW without heat transfer, the average energy consumption of the recovered CO_2 system is 1.41 kW, and it can be found that increasing heat recovery can save 0.19 kW power and reduce multiplier cabin 13.4% heating energy consumption. As a result, the heat is increased. Recycling can reduce the energy consumption of the system to a certain extent and save energy.

6. Conclusion

This paper uses the KULI simulation platform to establish the model of each module and vehicle of CO_2 heat pump module and thermal management system of pure electric car. The optimal refrigerant charge of the system is determined by simulation, and the performance of the CO_2 heat pump module and the thermal management system in the two modes of refrigeration and heating is simulated, and the system is compared and analyzed with the relevant literature. The conclusion is as follows.

Using KULI simulation software, the optimal refrigerant charge for summer and winter operating conditions was determined and the following results were obtained: the optimal charge was around 800 g in summer and around 700 g in winter. Considering that pure electric vehicles need to operate under various working conditions throughout the year, the selected refrigerant charge should meet all the workers as much as possible and make the system operating performance optimal, so the middle value of the two modes is selected, that is, 750 g, as the refrigerant charge of the system.

The refrigeration mode is simulated to obtain the impact of compressor speed changes on system performance. The results show that with the increase of running time, the temperature of the refrigerant at each status point of the system and the battery inlet tends to be stable, and the summer supply air temperature reaches equilibrium within

800s, and the supply air temperature decreases with the increase of the compressor speed; in the same state, although increasing the compressor speed can increase the cooling capacity, the exhaust temperature and pressure of the system will also increase, so that the compressor power consumption will increase, and finally lead to a decrease in COP_c. The minimum COP_c value is 1.8, and the compressor speed has less effect on the suction temperature and pressure.

By comparing with PTC electric heating and CO_2 system without waste heat recovery, it is found that the energy consumption of this system is only 42.5% of PTC electric heating, which is 86.6% of the CO_2 system without waste heat recovery, and after increasing waste heat recovery, the supply air temperature of the system can be increased by 26%, which proves the energy saving of the system.

Conflict of interest

The authors declared that they have no conflict of interest.

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