EXPERIMENTAL STUDY ON QUASI-TWO-STAGE COMPRESSION HEAT PUMP AIR CONDITIONING SYSTEM FOR PURE ELECTRIC BUS

by

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In response to the challenge of achieving efficient low temperature heating performance in heat pump air conditioning systems for pure electric buses, a quasi-two-stage compression heat pump air conditioning system incorporating fully micro-channel technology is developed. A series of experimental studies were conducted to investigate the influence of different compressor speeds and fan frequencies on system performance, resulting in corresponding performance curves. In a low temperature environment of –10 °C, the compressor exhaust temperature is maintained at 44.34 °C. Concurrently, the system heat exchange capacity is enhanced by 11.3% to 47% compared to the non-supplemented air system. Moreover, at a compressor speed of 4000 rpm, the system COP was found to be 2.72. The implementation of variable frequency compressor speed and fan frequency in the pure electric bus heat pump system has resulted in enhanced system performance, leading to energy savings.

Key words: *pure electric bus, quasi-two-stage compression, heat pump, R407c, performance experiment*

Introduction

The advent of *dual carbon* objectives and the intensification of the energy crisis, coupled with the transformation of the energy landscape, has led to the emergence of new energy vehicles as focal points, given their environmental benefits and contributions to sustainable development. As living standards rise, the prevalence of air-conditioned buses is rapidly expanding. In 2013, the production of pure electric buses was minimal, with only 1607 units manufactured. However, by 2023, according to Energy Conservation and New Energy Vehicles Yearbook, released by Ministry of Industry and Information Technology, Equipment Industry Development Center, this figure had increased significantly, reaching 30886 units, demonstrating a remarkable transformation in China's new energy vehicle industry over the course of a decade. As with conventional fuel buses, electric bus air conditioning systems play a crucial role in ensuring comfort and safety, necessitating continual advancements in their development and exploration [1, 2].

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Conventional bus air conditioning systems typically employ techniques such as PTC heaters or residual heat from fuel engines to provide auxiliary heating during winter. However, these approaches exhibit low energy efficiency and are incongruent with the developmental ethos of electric bus air conditioning systems. Heat pump air conditioning systems represent an optimal solution for electric buses, as they fulfill the design criteria by integrating heating and cooling functionalities, thus serving as a pivotal development pathway for electric bus air conditioning systems.

While heat pump air conditioning systems effectively address the energy efficiency and environmental concerns associated with traditional bus air conditioning, operating in heating mode within low temperature environments presents challenges. This phenomenon is accompanied by an increase in compressor exhaust temperature and the formation of severe frost on the external heat exchanger, which results in a gradual reduction in heating capacity. Consequently, the performance of the heat pump air conditioning system is compromised in low temperature environments, which in turn affects the thermal comfort of passengers within the vehicle.

This study focuses on a 7 m pure electric bus equipped with a fully micro-channel low pressure supplemental refrigerant heat pump air conditioning system, which serves as an experimental platform to investigate the performance of the system in low temperature environments. The incorporation of a scroll variable frequency compressor, in conjunction with variable frequency condensing and evaporating fans, enables the system to adapt to varying air flow rates during both the condensing and evaporating processes. In the past, air conditioning heat exchangers for buses of 7 m and 12 m lengths have predominantly utilized fin-tube and tube-and-fin configurations [3, 4]. In order to enhance heat transfer efficiency, minimize the refrigerant charge, and augment the cooling capacity ratio of the air conditioning system, parallel flow heat exchangers are employed for both the condenser and evaporator in this configuration. It is noteworthy that ongoing research by domestic and international experts has continually refined the design of micro-channel heat exchanger structures [5]. In the context of low temperature environments, the reduction in compressor suction mass-flow during heating mode results in elevated compressor exhaust temperatures, which can exceed 120 °C. This has a detrimental impact on the performance and lifespan of the compressor, and also results in the entire air conditioning system being unable to operate stably and efficiently. The low pressure supplemental refrigerant heat pump air conditioning system effectively addresses this issue by significantly reducing the exhaust temperature and increasing the refrigerant subcooling, thereby enhancing the system's heating capacity. In a low temperature environment with a temperature of –10 °C, the compressor exhaust temperature decreases from 120 °C to 42 °C, while the heat exchange capacity increases by 11.5%.

Experimental device

A significant body of research has been conducted by experts and scholars with the objective of identifying the stable operating characteristics of low pressure gas supply technology in low temperature environments. Tang *et al.* [6] observed that the economizer system of a scroll compressor can effectively reduce the compressor exhaust temperature. At a low evaporating temperature of –15 °C, the heating capacity is 25% higher than that of a single-stage cycle system, and the coefficient of performance for heating is increased by 20.3%. Wang [7] conducted performance analysis and parameter optimization studies on the economizer system of a scroll compressor, and the results demonstrated that jet enthalpy increase technology can effectively address the issue of excessively high exhaust temperature during operation of the heat pump system under high pressure ratios and can enhance system performance. In a related study, Li and Qin [8] examined the refrigeration performance of the electric vehicle heat pump system. The system was found to exhibit significant advantages in reducing the compressor exhaust temperature during refrigeration. In a standard refrigeration external environment of 30 °C, the refrigeration capacity and compressor power of the system exhibited a notable increase of 12.97% and 4.92%, respectively, accompanied by a 7.94% enhancement in the system COP. When the compressor speed was elevated from 2000 rpm to 6000 rpm, the refrigeration capacity and compressor power of the system demonstrated a 6.01% to 12.31% and 2.06% to 6.24% growth, respectively, accompanied by a 0.63% to 6.51% expansion in the system COP.

The thermal cycle principle diagram of the low pressure supplemental refrigerant heat pump system with an intermediate heat exchanger utilized in this system is depicted in fig. 1. A portion of the refrigerant exiting the condenser enters the intermediate heat exchanger after throttling through an auxiliary electronic expansion valve (EEV). Another portion of the main refrigerant exchanges heat with auxiliary refrigerant in the intermediate heat exchanger, thereby making the main refrigerant colder and increasing the degree of subcooling, thus improving the heat exchange capacity. The process of supplemental refrigerant injection is completed instantaneously and can be considered an adiabatic isobaric enthalpy increase process.

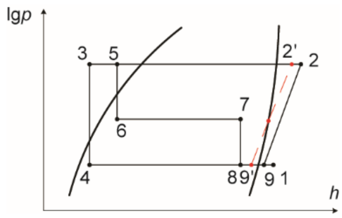

Figure 1. Low pressure gas supplement system

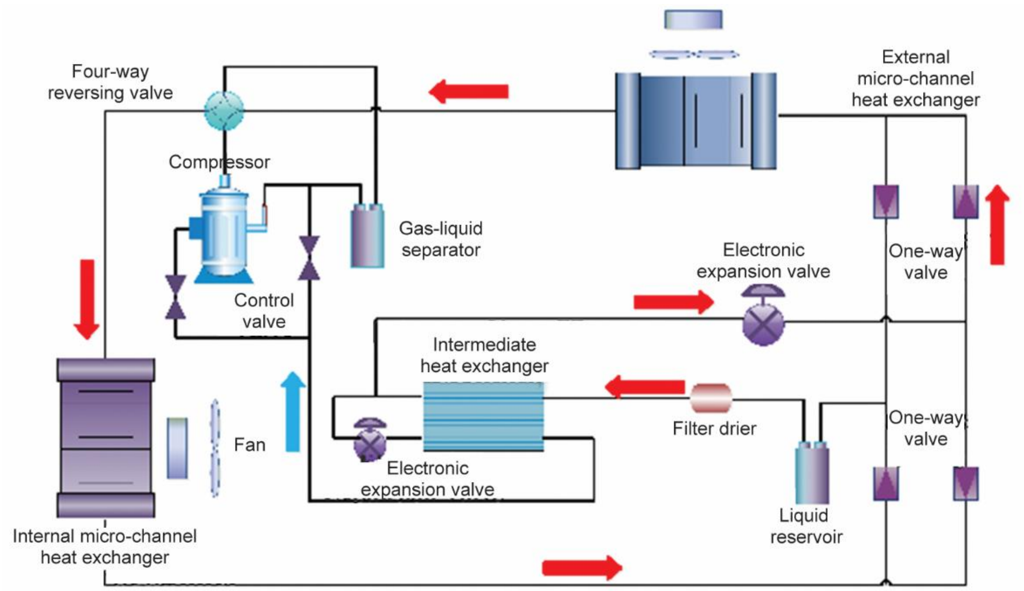

Figure 2. System cycle principle

The process of the fully micro-channel low pressure supplemental refrigerant heat pump air conditioning system for electric buses is illustrated in fig. 2. The micro-channel is widely used to control the heat transfer efficiency in micro devices as discussed in [9-13]. The process of the system for electric buses operating in heating mode is as: the primary refrigerant exits the compressor outlet (point 1 via 9 to 2 in fig. 1), passes through the four-way reversing valve, enters the interior heat exchanger, undergoes condensation (point 2 to 3 in fig. 1), and releases heat to provide warmth for the interior of the bus. Subsequently, the primary refrigerant is throttled and subjected to a pressure reduction via the main EEV (point 3 to 4 in fig. 1) prior to entering the external heat exchanger, undergoes evaporation (point 4 to 1 in fig. 1). Concurrently, the supplementary refrigerant is throttled and subjected to a pressure reduction via the auxiliary EEV (point 5 to 6 in fig. 1), engages in a heat exchange with the principal refrigerant within the intermediate heat exchanger (point 6 to 7 in fig. 1). Subsequently, the low pressure supplemental refrigerant is injected into the compressor working chamber (point 7 to 8 in fig. 1), mixing with the primary refrigerant exiting from the evaporator (point 8 to 9 in fig. 1).

Experimental equipment

The experiment was conducted in a laboratory that maintained a constant temperature and humidity. The laboratory is comprised of two independent rooms, which are designed to simulate the temperature and humidity conditions of the vehicle's interior and exterior. The temperature control range in the simulated interior environment is –30 °C to 45 °C, while the temperature control range in the simulated exterior environment is –30 °C to 55 °C, with a control precision of ±0.01 °C. In this experiment, Carel electronic expansion valves and Emerson pressure sensors were employed. To ensure the accuracy and precision of the data, pressure and temperature parameters were collected at multiple points. The air-flow rate within the vehicle was quantified using an air-flow measurement apparatus, and the compressor speed and the air-flow rates of the interior and exterior fans could be adjusted via the control panel. System valves are presented in tab. 1.

Table 1. Main parameters of the experimental prototype and test instrument

Equipment name	Specifications
Main EEV	CAREL E^2V-18, capacity: 12.6 kW, adjustment range: 10%~100%
Auxiliary EEV	CAREL E^2V-11, capacity: 5 kW, adjustment range: 10%~100%
Intermediate heat exchanger	Weal Yield (Jiangsu) plate heat exchanger model B3-014-10D-3.0; design capacity: 5 kW; design temperature: –160~200℃
Four-way valve	Dunan DSF-20, applicable capacity: 7.1~25 kW

As shown in tab. 1, the system uses electronic expansion valves that respond faster.

A prototype system was developed based on the circulating principle of a low pressure gas supplement heat pump system with an intermediate heat exchanger. The system performance in a low temperature environment was experimentally studied.

The compressor of this system employs a variable-frequency electric scroll compressor with three speeds. The compressor operates at three distinct speeds: 4000 rpm, 3000 rpm, and 2000 rpm. The displacement of the compressor is 34 cm^3. The system is lightweight, exhibits minimal vibration and noise, and is highly reliable. The refrigerant utilized is a blend of R407c, with a system refrigerant charge of 4 kg. The R407c is a medium – and low temperature refrigerant that is renowned for its energy-saving and environmentally friendly properties, with a high cooling capacity per unit volume. The use of R407c in automotive air conditioning systems has the effect of reducing the volume and mass of compressors and components to a significant degree [14]. Variable frequency fans are employed both within and without the vehicle. The internal fan is equipped with three-speed settings (99%, 80%, and 50%), while the external fan has two speeds (99%, 85%).

Experimental results and analysis

The observations presented in fig. 3 demonstrate a clear correlation between the decline in ambient temperature and a reduction in evaporating temperature. This, in turn, results in a decline in compressor suction gas mass-flow rate and a continuous drop in system heating capacity. In particular, at a compressor speed of 4000 rpm, as the outside ambient temperature decreases, resulting in a lower evaporating temperature, the system heating capacity experiences a consistent decrease. In comparison to the standard heating condition of 7 °C, the system heat exchange capacity is reduced by 41.4% at –10 °C with low pressure (LP) gas supplement, by 32.1% at –5 °C, and by 18.9% at 0 °C. This demonstrates a progressive decline in heating capacity due to ambient temperature variations.

This decline in heating capacity subsequently affects the system COP, as illustrated in fig. 4. At –10 °C with low pressure gas supplement, the system COP is 2.72, representing a 32.5% reduction compared to the COP at the standard heating condition of 7 °C. Furthermore, the implementation of low pressure gas supplement technology results in an increase in compressor power, which further contributes to the diminished COP of the system when compared to the system operating without gas supplement (NG). To enhance the system performance, it is necessary to make adjustments to the compressor speed, condensing air-flow rate, and the control of the superheat and opening of the electronic expansion valve. Figures 6-12 illustrate the impact of varying compressor speeds and condensing air-flow rates on the performance of the heat pump system.

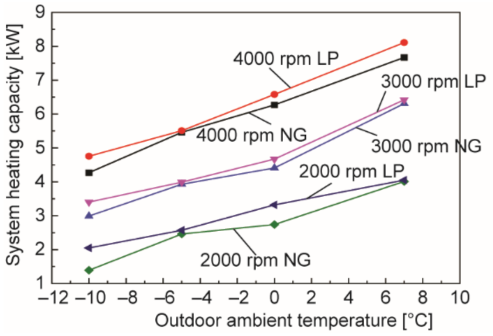

Figure 3. Outside ambient temperature impact on the system heating capacity

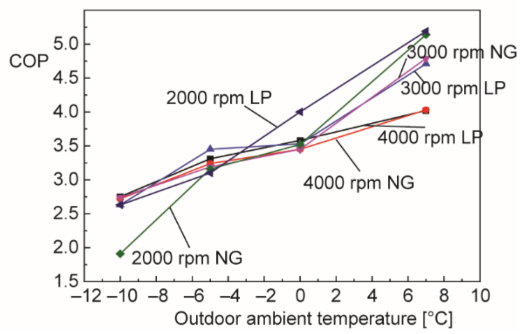

Figure 4. Outside ambient temperature impact on the system COP

The supply air temperature is a pivotal parameter in evaluating thermal comfort within the vehicle. As such, it is a significant criterion for assessing the performance of low temperature heating systems. The influence of the external ambient temperature on the supply air temperature is depicted in fig. 5. As the outside temperature declines, the evaporation temperature also decreases, resulting in a corresponding decline in the condensing temperature and, consequently, the supply air temperature. Specifically, at a compressor speed of 4000 rpm, when the outside temperature decreases from 7 °C to –10 °C, the supply air temperature post low pressure gas supplement usage decreases by 6.37 °C, reaching 28.33 °C at –10 °C.

The compressor speed ranges from 2000 rpm to 4000 rpm, with outside ambient temperatures of 0 °C, –5 °C, and –10 °C. The primary electronic expansion valve exhibits a superheat of 5 K, while the auxiliary electronic expansion valve exhibits a superheat of 10 K to 20 K. Both internal and external fans are set to a high air-flow.

The effects of compressor rotational speed variation on compressor exhaust pressure and exhaust temperature are illustrated in figs. 6 and 7, respectively.

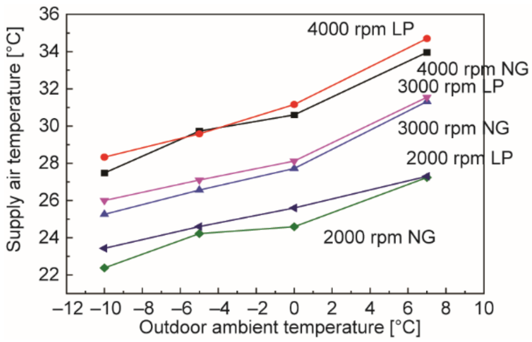

Figure 5. Outside ambient temperature impact on the supply air temperature

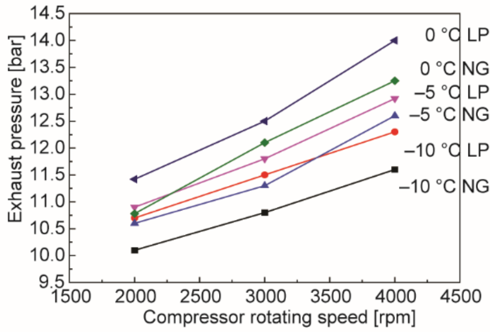

Figure 6. Effect of compressor speed impact on the exhaust pressure

Figures 6 and 7 demonstrate that both the compressor exhaust pressure and exhaust temperature increase as the compressor speed rises, with the exhaust pressure exhibiting a particularly linear relationship. At an outside ambient temperature of –10 °C, with a compressor speed of 2000 rpm, the exhaust temperature of the compressor without gas supplement peaks at 121.4 °C, which is significantly higher than when low pressure gas supplement technology is applied. This technology effectively addresses the issue of excessively high compressor exhaust temperatures in low temperature environments, which could otherwise lead to unstable system operation. While the exhaust pressure with low pressure gas supplement is slightly higher than without, it remains at approximately 12 bar when the compressor speed is 4000 rpm. At lower compressor speeds, the exhaust pressure remains below 12 bar, ensuring the system's safe and reliable operation.

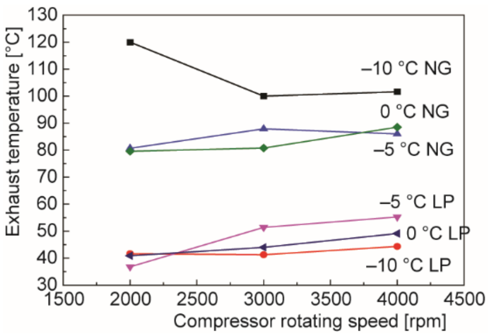

Figure 7. Effect of compressor speed impact on exhaust temperature

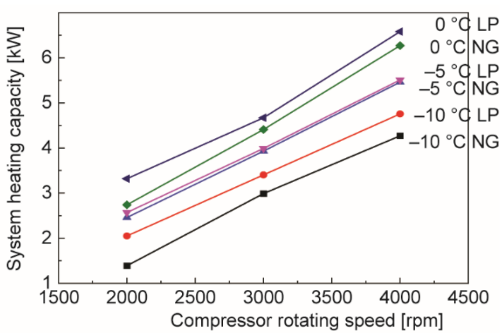

Figure 8. Effect of compressor speed impact on system heating capacity

Figure 8 illustrates the impact of varying compressor rotational speeds on the heating capacity of the system.

Figure 8 illustrates that the system heating capacity increases with an increase in compressor speed. In a low temperature environment with a temperature of –10 °C, a compressor speed of 4000 rpm, and the implementation of low pressure gas supplement technology, the single-sided system heating capacity reaches 4.8 kW. The introduction of low pressure gas supplement results in a reduction in the temperature of the refrigerant, which in turn leads to an increase in the mass-flow rate of the compressor suction gas. Consequently, in a –10 °C low temperature environment, the system heating capacity with low pressure gas supplement is 11% to 47% higher than that without gas supplement.

The influence of varying compressor speeds on the system COP is illustrated in fig. 9.

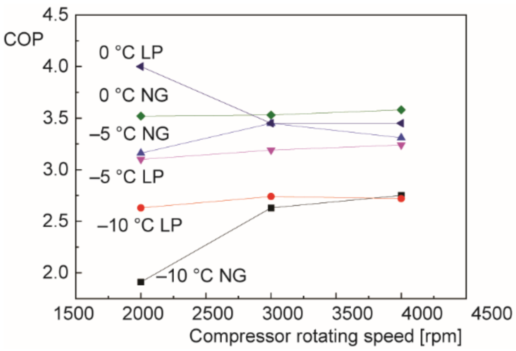

Figure 9. Effect of compressor speed impact on the system COP

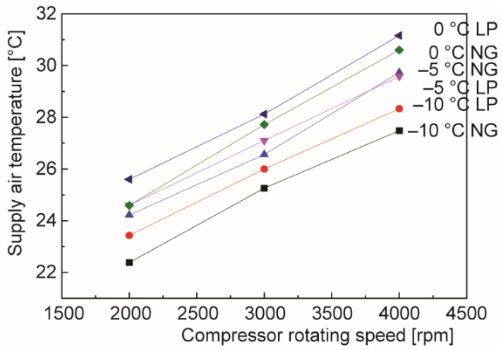

Figure 10. Effect of compressor speed impact on the supply air temperature

As the compressor speed increases, the mass-flow rate of the exhaust gas also increases, resulting in an increase in the condensing temperature. Consequently, the supply air temperature rises in tandem with the increase in compressor speed. Conversely, the exhaust temperature following the implementation of low pressure gas supplement technology is observed to be 2.9% to 4.7% higher than that observed in the absence of gas supplement, fig. 10.

This paper examines the influence of condenser air-flow on system performance under conditions of an outside temperature of –5 °C and –10 °C, a compressor speed of 4000 rpm, a main electronic expansion valve superheat setting of 5 K, and an auxiliary circuit electronic expansion valve superheat setting ranging from 10 K to 20 K. The impact of varying condenser air-flow volumes on system performance was evaluated. In particular, the influence of condenser air-flow volume on the system heat exchange is depicted in fig. 11.

Figure 11 illustrates the impact of condenser air-flow on system heating capacity. The heating capacity of the system in question is observed to increase as a consequence of an increase in the condenser air-flow. At an outside temperature of –10 °C, the adoption of low pressure gas supplement technology resulted in an increase in condenser air-flow from 50% to the maximum condenser air-flow, which in turn led to an increase in system heat exchange from 3.74 kW to 4.76 kW. At an outside temperature of –5 °C, the system heat exchange increased from 4.85 kW to 5.51 kW. Specifically, at an outside temperature of –10 °C and a compressor speed of 4000 rpm, the system heating capacity with low pressure gas supplement increases by 6.8% to 73.3% compared to the system without gas supplement. The single-side system heating capacity reaches 4.8 kW at the maximum condenser air-flow with low pressure gas supplement, and the heating capacity after activating the dual-side system can meet the requirements of the entire vehicle load.

The influence of fluctuating condenser air-flow on the COP is depicted in fig. 12.

The preceding findings demonstrate that as the condenser air-flow increases, the system heating capacity gradually improves, resulting in an enhancement of the system COP. Specifically, at an outside temperature of –10 °C, under different compressor speeds and gas supplement modes, the system COP at maximum condenser air-flow is 34.1% to 64.7% higher than the COP at 50% condenser air-flow. The increase in system COP is more pronounced at a compressor speed of 2000 rpm, with the COP of low pressure gas supplement approaching that of the 3000 rpm compressor speed. Moreover, at various levels of condenser air-flow, the COP of low pressure gas supplement is 39.2% to 62.9% higher than the COP without gas supplement.

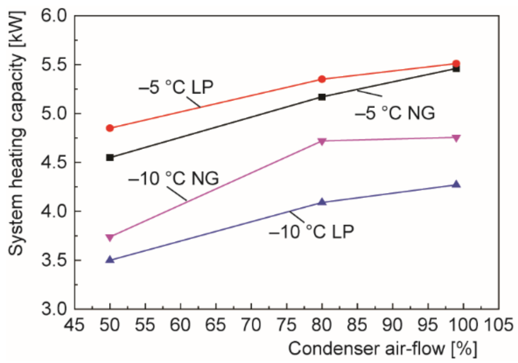

Figure 11. The amount of condenser air-flow impact on the heating capacity

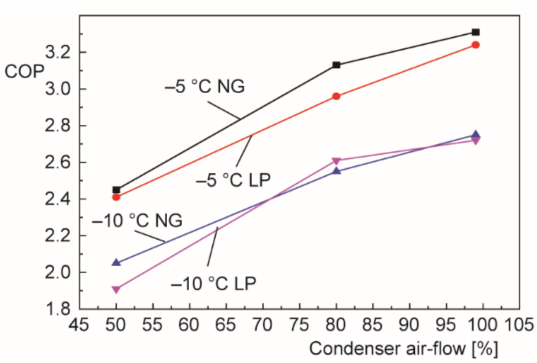

Figure 12. The amount of condenser air-flow impact on the system COP

Conclusions

The analysis of experimental data indicates that the performance of the system can be enhanced by adjusting the system control strategy according to user requirements. This includes adjusting compressor speed, fan frequency, and supplementary gas volume.

A heating cycle with gas supplement technology has been designed to experimentally investigate the dynamic characteristics of the system in low temperature environments. When the system is in the heating mode at a temperature of –10 °C, the exhaust temperature of the compressor without the addition of gas reaches 121.4 °C. With the application of low pressure gas supplement, the exhaust temperature is maintained at 44.34 °C, with an exhaust pressure of 12.3 bar. The single-side system heat exchange capacity was found to reach 4.8 kW, representing an increase of 11% to 47% compared to the system without gas supplement. Furthermore, the exhaust temperature with low pressure gas supplement is 2.9% to 4.7% higher than without gas supplement. Consequently, in a low temperature environment of –10 °C, the activation of low pressure gas supplement effectively increases the system heat exchange capacity and exhaust temperature, thus satisfying the requirements for a 7m bus thermal comfort.

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