EXPERIMENTAL STUDY ON OPTIMAL OPERATION CONTROL STRATEGY OF QUASI-TWO-STAGE COMPRESSION HEAT PUMP AIR CONDITIONING SYSTEM BASED ON FULL MICRO-CHANNEL HEAT EXCHANGER

by

Dengke ZHAO[a,b], Haijun LI[a*], Gang CHEN[a], Yibo ZHANG[a], and Jiayang GAO[a]

[a]School of Energy and Environment, Zhong Yuan University of Technology, Zhengzhou, China
[b]School of Mechanical and Electrical Engineering, Henan Agricultural University, Zhengzhou, China

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In environments with significant temperature variations, the quasi-two-stage compression heat pump air conditioning system demonstrates notable advantages in mitigating key challenges such as high compressor exhaust temperature, exhaust pressure, low system heat exchange, and COP. This study optimizes the optimal refrigerant charge amount and superheat set-point control strategy of the electronic expansion valve for the system. By adjusting the method of supplying the gas and the superheat set-point of the electronic expansion valve, the system is able to achieve stable and efficient operation. The experimental findings indicate that following the optimization of operational control strategies, the system heat exchange capacity increased by 18.8%, the COP improved by 12.3%, and the compressor exhaust temperature decreased by 11.3 °C.

Key words: *full micro-channel, quasi-two-stage compression, heat pump, control strategy*

Introduction

In recent years, the introduction of the *dual carbon* goal and the escalation of the energy crisis have prompted a re-evaluation of the current energy landscape. The government has demonstrated its support for the development of pure electric buses, which offer significant advantages in terms of energy efficiency and emissions reduction. However, the heat pump system, which serves as the primary power-consuming equipment of electric buses, severely constrains the vehicle's safety, reliability, range, and application scope [1-5]. The heat pump system, which is required to operate year-round in variable conditions, particularly in extreme environments, experiences a significant decline in efficiency. This is accompanied by safety concerns such as excessively high compressor exhaust temperature and pressure [6]. The quantity of refrigerant present in the heat pump system, a crucial component, has a profound impact on its functionality. Insufficient refrigerant charge results in ineffective utilization of the evaporator area, premature evaporation of the refrigerant into gas before completing the cycle, leading to excessively high compressor suction temperature, elevated exhaust

* Corresponding author, e-mail: haijun_li007@126.com

temperature, and hindering normal system operation. Conversely, an excessive refrigerant charge can result in a dangerously high compressor discharge pressure, which has the potential to damage the system [7], and it was concluded that micro-channel heat exchanger is much needed. The electronic expansion valve (EEV) serves as the primary regulating valve of the heat pump system, offering several advantages. These include high control precision, fast response speed, a wide adjustment range, and adaptability to large load changes.

The adjustment of the superheat set-point of the EEV represents a pivotal methodology for the optimization of the operation of HVAC systems, particularly in terms of heating performance and stability. The superheat set-point determines the extent to which the refrigerant is heated above its saturation temperature at the end of the evaporator. By setting this parameter with precision, the EEV is better able to regulate the degree to which the valve opens, thereby effectively controlling the flow of refrigerant through the system. [8] The quasi-two--stage compression cycle heat pump technology incorporates an intermediate gas supplementation loop based on the heat pump cycle. Depending on the different stages of compressor mixing, the gas supplementation can be categorized as low pressure or medium pressure. The introduction of low pressure gas supplementation has the effect of reducing the temperature of the exhaust gases produced by the compressor, while the incorporation of medium pressure supplementation allows for an increase in the load on the condenser under conditions of low evaporation temperature, thus enhancing the system's heating capacity. Consequently, the investigation of refrigerant charge quantity, EEV adjustment characteristics, and gas supplementation control strategy for the heat pump system is of paramount importance.

In recent times, a considerable number of experts and scholars have conducted extensive research in this field. In a study by Shi *et al.* [9], the impact of refrigerant charge quantity on heat pump performance in low temperature environments was investigated. The researchers discovered that exceeding the optimal refrigerant charge amount resulted in a sharp increase in compressor power and a significant decline in system performance. Consequently, it is of paramount importance to consider the unit energy efficiency and compressor operating state when determining the optimal refrigerant charge amount. This approach ensures both high-efficiency operation and safe and stable unit operation. The research conducted by Choi and Kim [10] provides evidence that the use of EEV is advantageous in the management of heat pump system efficiency and performance under heating conditions. This advancement in HVAC technology presents a compelling case for modernizing existing systems or designing new installations with EEV for enhanced energy efficiency and performance. Wang [11] proposed a coupling throttling device controlled by an electromagnetic valve and capillary tube for gas supplementation, from the perspective of technical feasibility and economic affordability. The results of the experimental comparison between throttle valve-controlled gas supplementation and capillary tube and electromagnetic valve coupling throttling gas supplementation indicate that the heat pump unit performance under stable low temperature conditions was comparable between the two methods. Yu *et al.* [12] optimized the control strategy of superheat for EEV. The introduction of compensation signals, including gas supplementation superheat and condensing pressure, enabled the adjustment of the opening degree of the expansion valve, thereby regulating the refrigerant flow on the suction and gas supplementation sides. This improved the stability of the unit's operation. Wang [13] conducted research on the matching characteristics of EEV opening and flow rate for heat pump systems, as well as gas supplementation characteristics. This research laid the foundation for the selection and optimization of EEV and control systems. They conducted a study on the heating performance of air-source heat pump units with economizers under different main valve openings and gas supplementation methods. The researchers discovered that the main EEV had an optimal opening position, indicating a specific setting that maximized system performance. Furthermore, the study revealed that adjusting the opening of the gas supplementation EEV could effectively enhance the heat pump system heating capacity.

Principle of system circulation

The quasi-two-stage compression cycle is primarily composed of equipment such as compressors, condensers, evaporators, intermediate heat exchangers, dryers, liquid storage tanks, gas-liquid separators, main routes, EEV, valves and the full micro-channel heat exchanger. The system process is depicted in fig. 1. The system is classified into two categories: low pressure and medium pressure replenishment. These categories are depicted in the theoretical cycle diagram in fig. 2. The theoretical cycle of the quasi-two-stage compression heat pump system, which illustrates the principle of low pressure gas supplementation, is depicted in fig. 2(a). The theoretical cycle principle of medium pressure gas supplementation in a quasi-two-stage compression heat pump system is illustrated in fig. 2(b).

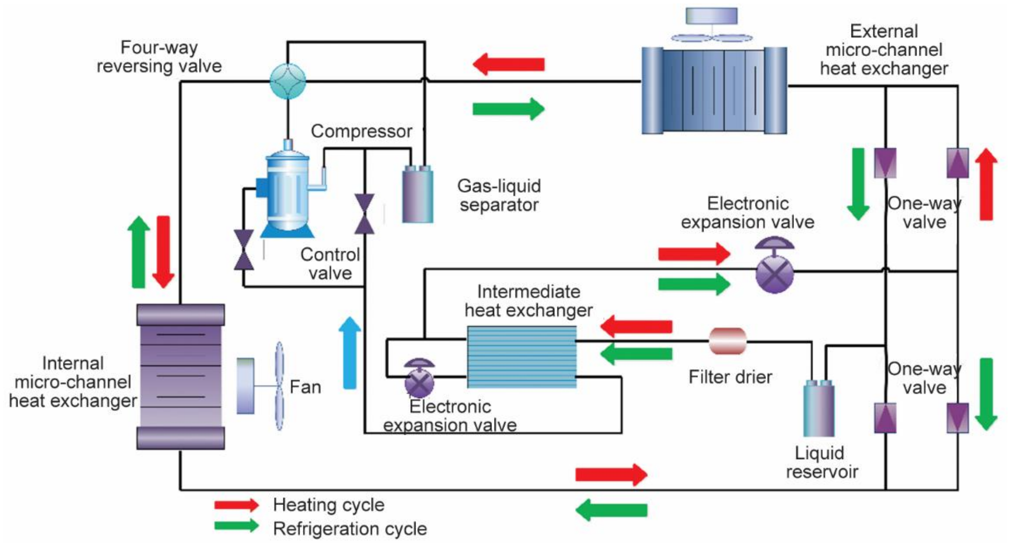

Figure 1. System cycle principle

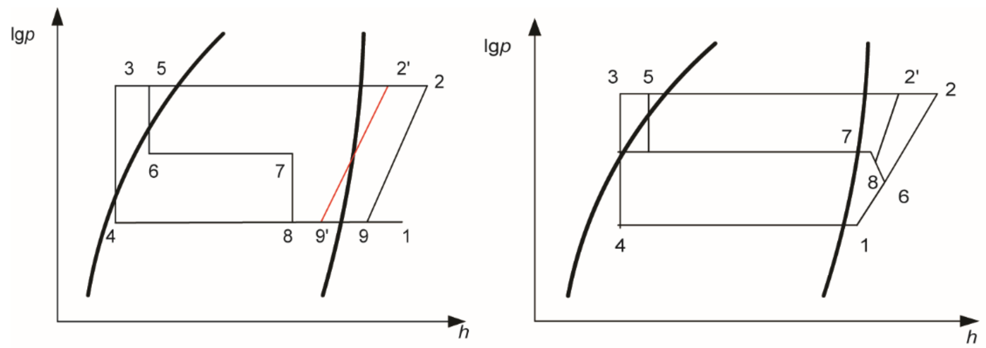

Figure 2. Theoretical cycle of the system; (a) low pressure gas supplement theory cycle and (b) medium pressure gas supplement theory cycle

Experimental set-up

The experimental bench employs a Shanghai Hitachi digital scroll frequency conversion compressor with a displacement of 34 cc and a speed range of 2000-6000 rpm. The system employs parallel flow heat exchangers throughout. The external heat exchanger has a surface area of 3.77 m^2, while the internal heat exchanger has a surface area of 3.56 m^2. The axial flow fan situated outside the vehicle has a rated air volume of 6000 m^3/h, while the centrifugal fan located within the vehicle has a rated air volume of 2000 m^3/h. System valves are presented in tab. 1.

Table 1. System valves and performance parameters

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.

Experimental plan

The experimental conditions for the optimal refrigerant charge amount of the system are as follows in tab 2.

Table 2. Experimental plan for optimal amount of refrigerant charge

Test condition	Compressor speed	Outside temperature DB/WB	In-car air-flow percentage	In-car temperature DB/WB	Gas supplementation method	Main EEV superheat
	4000 rpm	35 ℃/28 ℃	99%	27 ℃/19.5 ℃	Without gas supplementation	5 K

Meanwhile experimental conditions for assessing the impact of the gas supplementation method on system performance were: ambient temperature outside the car was -10 ℃, while the temperature inside the car was 15 ℃, the compressor speed was 3000 rpm, the main EEV superheat was 5 K, while the auxiliary EEV was 10 K, and the fan inside and outside the car was a large air volume.

The experimental plan for the impact of EEV superheat on system performance is presented in tab. 3.

Table 3. Experimental plan for the impact of EEV superheat on system performance

	Main EEV superheat	Auxiliary EEV superheat
Outside ambient temperature [℃]	–10	
Compressor speed [rpm]	2500	
Superheat set-point [K]	5 K, 3 K	20 K, 10 K
Gas supplementation method	Without gas supplementation	Low pressure gas supplementation
Fan air-flow	Maximum air-flow	

Results analysis

Study on optimal amount of refrigerant charge

As the quantity of refrigerant in the cycle fluid increases, the cooling capacity initially improves gradually, reaching a stable state. Specifically, an increase in the refrigerant charge quantity from 2.2-2.9 kg resulted in a 14.03% increase in cooling capacity. Upon further increase in the charge amount to 4.1 kg, the cooling capacity reaches a peak, exhibiting an 8.1% increase. Nevertheless, the system cooling capacity exhibits a slight decline of 1.9% when the refrigerant charge quantity increases from 4.1-5.1 kg. Figure 3 illustrates that an increase in the refrigerant charge in the cycle fluid does not necessarily result in enhanced performance.

As illustrated in fig. 4, an increase in the refrigerant charge amount in the cycle fluid is associated with a gradual improvement in the COP of the system, reaching a maximum value of 3.18 when the charge amount reaches 4.0 kg. Subsequently, the COP exhibits a slight decline, with a reduction of 5.3%.

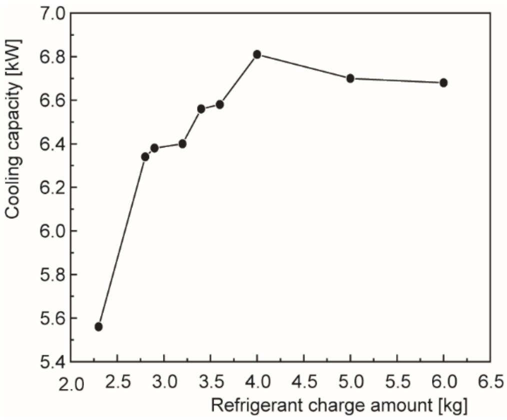

Figure 3. Variation of cooling capacity with amount of refrigerant

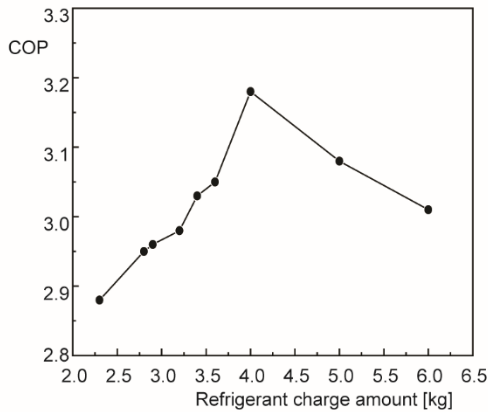

Figure 4. Variation of COP with amount of refrigerant

Figure 5 depicts the trend of exhaust temperature with increasing refrigerant charge amount. At the initial charge amount of 2.3 kg, the exhaust temperature reaches 92 °C. Subsequently, as the charge amount increases to 4.0 kg, the exhaust temperature decreases to 81.75 °C, representing an 11.1% reduction. Upon further increase of the charge amount to 6.0 kg, the exhaust temperature exhibits a slight decrease to 80.52 °C, with a mere 1.5% reduction. This represents a minimal change in the temperature profile.

The data presented in fig. 6 indicates that as the quantity of refrigerant in the system changes, the sub-cooling also varies in a corresponding manner. In the initial stage, when the charge amount is only 2.3 kg, the sub-cooling is relatively low, at only 0.53 K. Subsequently, as the charge amount increases to 2.8 kg, the sub-cooling at the condenser outlet increases to 1.05 K. As the refrigerant charge amount continues to increase, the sub-cooling continues to rise and eventually stabilizes at approximately 1.8 K, representing an increase of approximately 71% compared to the previous value. In conclusion, the analysis reveals that when the refrigerant charge reaches 4.0 kg, the cooling capacity, COP, sub-cooling, and exhaust temperature of the set-up heat pump system are all optimal and tend to stabilize. This indicates that this refrigerant charge amount represents a performance peak during system operation.

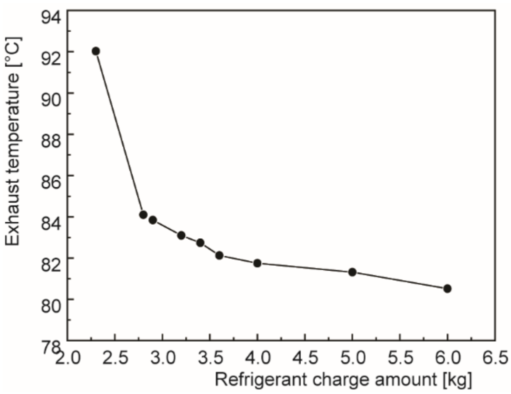

Figure 5. Variation of exhaust temperature with amount of refrigerant

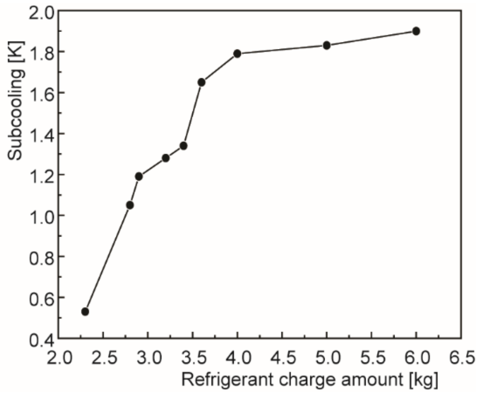

Figure 6. Variation of sub-cooling with amount of refrigerant

Gas supplementation method impact on system performance

As illustrated in fig. 7, under identical operational conditions and with a compressor speed of 3000 rpm, without gas supplementation (NG) results in significantly elevated exhaust temperatures in comparison to both low pressure (LP) and mid-pressure (MP) gas supplementation methods. Furthermore, the exhaust temperature without gas supplementation increases by 28.3% from the system operation at 15 minutes to the end of 120 minutes. In contrast, when the system employs MP and LP gas supplementation techniques, the compressor exhaust temperature remains relatively stable, oscillating between 34 °C and 41 °C. The influence of different gas supplementation methods on system heating capacity is depicted in fig. 8. After 120 minutes of system operation, the heat exchange capacity for MP gas supplementation is 3.51 kW, for without gas supplementation is 3.47 kW, and for LP gas supplementation is 3.08 kW. The incorporation of MP gas supplementation into the system resulted in a 12.3% increase in the heat exchange capacity relative to the absence of supplementation. Conversely, the absence of supplementation led to 11.2% enhancement in the heat exchange capacity relative to LP gas supplementation. The heat exchange capacity is elevated by 1.1% in the presence of MP gas supplementation in comparison to without gas supplementation.

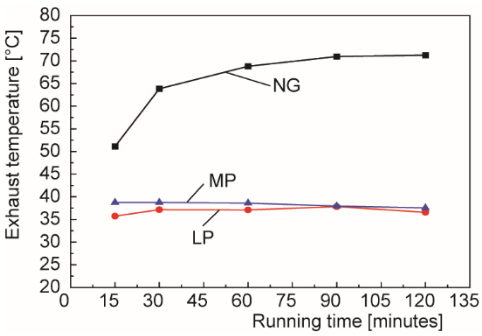

Figure 7. Impact of gas supplementation methods on exhaust temperature

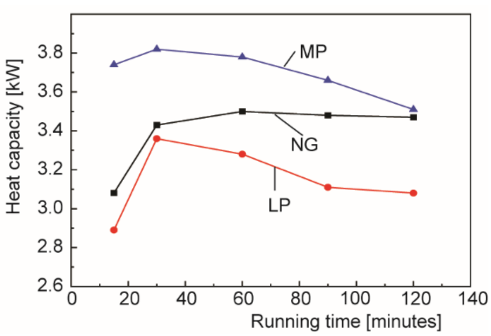

Figure 8. Impact of gas supplementation methods on system heating capacity

The EEV superheat degree impact on system performance

The influence of variable main valve superheat degree on system performance is illustrated in fig. 9.

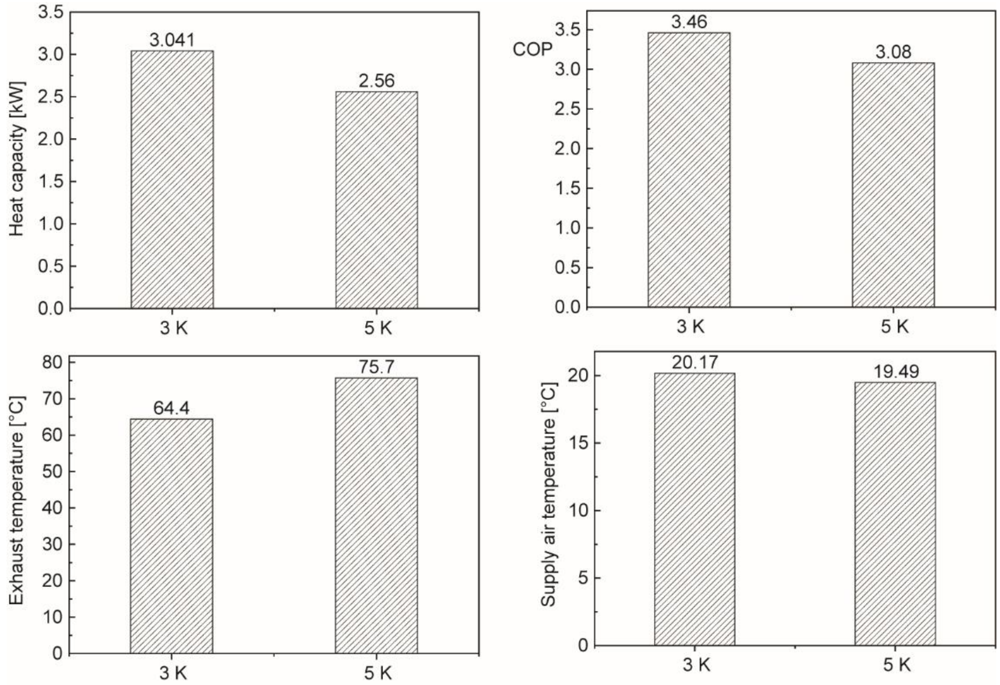

Figure 9. Impact of main valve superheat on system heating performance

Figure 9 illustrates that the system heat exchange capacity at a main valve superheat degree of 3 K is 18.8% higher than that at 5 K. Moreover, as the compressor power variation is almost constant, the system COP at a main valve superheat degree of 3 K is 12.3% higher than that at 5 K. Concurrently, the exhaust temperature decreased by 11.3 °C relative to that at the 5 K superheat degree. Moreover, as the opening degree of the main valve increases, the compressor's discharge mass-flow rate also rises, resulting in an increase in the supply air temperature.

The impact of variable auxiliary valve superheat degree on system performance is illustrated in fig. 10.

Figure 10 illustrates that as the set-point value of the auxiliary valve superheat degree decreases, the opening degree of the auxiliary valve increases. This results in an increase in the refrigerant flow directly entering the compressor through the auxiliary loop. Consequently, the exhaust temperature corresponding to an auxiliary valve superheat degree of 10 K is 10.2 °C lower than that of 20 K. As the opening degree of the auxiliary valve increases, the compressor power also increases, which in turn leads to a decrease in the system COP.

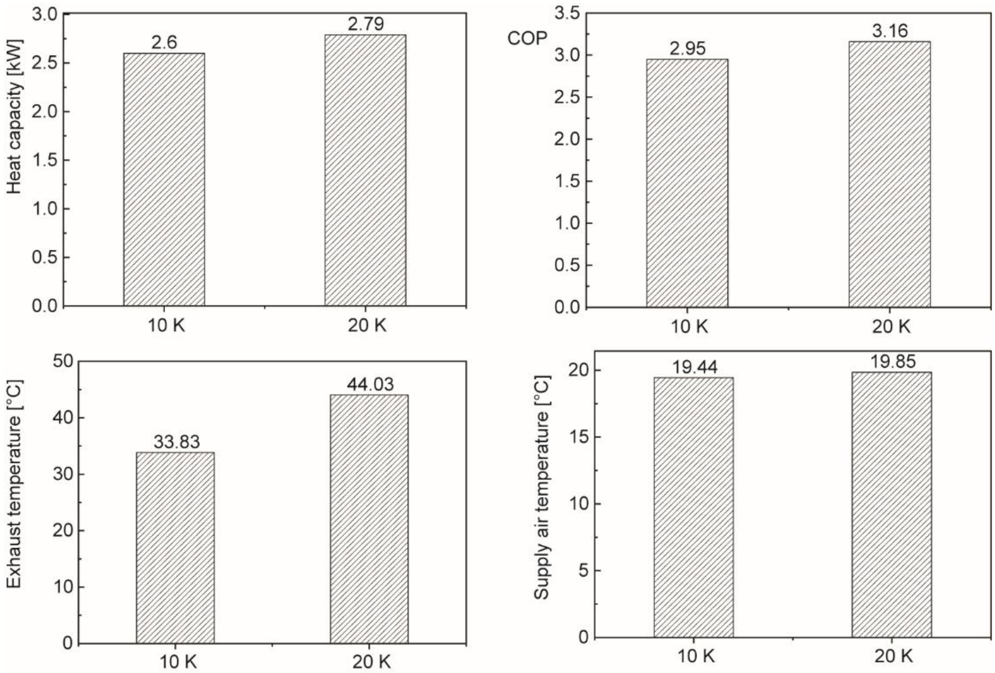

Figure 10. Impact of auxiliary valve superheat on system heating performance

Conclusions

- The optimal refrigerant charge for cooling operation is determined to be 4.0 kg based on dynamic refrigerant charging data.
- The exhaust temperature is controlled at 44.34 °C by implementing LP gas supplementation. The heat exchange capacity of the single-side system is observed to reach 4.8 kW, representing an improvement of 11% to 47% compared to the system without gas supplementation.
- Adjusting the superheat of the main valve to 3 K increases the system heat exchange capacity by 18.8% compared to when it is set to 5 K, and the COP increases by 12.3%. Similarly, setting the superheat of the supplementary valve to 10 K results in a decrease in exhaust temperature by 10.2 °C compared to when it is set to 20 K. These findings offer valuable insights for the development of optimal control strategies for heat pump systems.

The efficiency of the micro-channel heat exchanger in fig. 1 can be greatly increased using nanofluids [14], the metal nanoparticles in the nanofluid can greatly enhance the heat exchange through the boundary, as revealed in [15]. The micro-channel heat exchanger using nanofluids can be also used to control temperature of micro devices, especially the micro-electromechanical systems [16, 17].

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