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Approaching optimal high pressure by charge management in transcritical CO_2 heat pump water heater

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Abstract

Transcritical CO_2 cycle has been widely applied in heat pump systems. Optimal high pressure is crucial to the performance of transcritical CO_2 heat pump water heater (HPWH). However, current techniques for high pressure control cannot meet the requirements of easy-to-implement and low COP loss simultaneously. From the perspective of charge management, this work aims to approach optimal high pressure in CO_2 HPWH without high pressure control.

By modeling a running CO_2 HPWH, we in-depth investigated the relationship between optimal high pressure and refrigerant charge. It was found that for practical systems, the essence of optimal high pressure control is to manage optimum refrigerant charge of the system. If the optimum charge of the system remains constant under any working conditions, the system can be guaranteed at optimal high pressure with fixed refrigerant charge. Then the influence of water inlet temperature, water outlet temperature and ambient temperature on optimum refrigerant charge was analyzed. Two system design methods with the aim of minimizing the change of optimum charge under different working conditions are presented, which are re-sizing heat exchangers and adding additional refrigerant reservoir. The minimum average COP loss of the redesigned system can be reduced to only 2.82% and 0.67%, respectively.

Keywords: Heat pump water heater; Transcritical CO_2 cycle; Optimal high pressure; Refrigerant charge management

Nomenclature

A	cross-sectional area in pipe (m^3)	V	volume (m^3)
COP	coefficient of performance (-)	v	specific volume (m^3·kg^{-1})
DB	dry bulb	WB	wet bulb
EXV	electronic expansion valve	x	flowing vapor mass quality (-)
HPWH	heat pump water heater	$\dot{m}$	mass flow rate (kg s^{-1})
L	length (m)	p	pressure (kPa)
M	refrigerant charge (kg)	T	temperature (°C)
S	slip ratio		
Greek symbol			
α	void fraction	ρ	density (kg·m^{-3})
Subscripts			
c	condensing	GC	gas cooler
COMP	compressor	L	liquid
e	evaporator, evaporating	RES	reservoir
EVAP	evaporator	SH	superheat
G	gas	TP	two phase

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1. Introduction

With the increasing awareness of environmental protection around the world, natural refrigerants, as carbon dioxide, have received extensive attention. Since transcritical CO_2 cycle was first proposed by Lorentzen [1, 2] in the 1990s, it has been successfully applied in heat pump systems [3-5]. The heat transfer process (gas cooling) of transcritical CO_2 cycle is in the supercritical region, which leads to high discharge temperature and large temperature glide. Therefore, CO_2 heat pump water heater (HPWH) is capable of matching water temperature profile, and supply high temperature water with rather high efficiency [6].

In supercritical region, temperature of CO_2 is independent of pressure. The change of discharge pressure will affect cooling capacity and power consumption simultaneously. Consequently, optimization of discharge pressure is crucial to system efficiency. In open literature, two main methods can be concluded for high pressure control: offline method and online method.

The offline method is primarily proposed and then thoroughly investigated. It is to fit the correlations of optimal high pressure and other parameters of the system based on thermodynamic model or experimental data. Kauf [7] first proposed a linear equation of optimal high pressure with a single variable. On this basis, Liao, Zhao [8] analyzed the influence of gas cooler outlet temperature, the evaporating temperature and compressor performance on optimal high pressure. Regressions were made to various forms of correlations. Sarkar, Bhattacharyya [9] conducted energetic and exergetic analyses for optimization, and developed correlations of optimal high pressure with gas cooler outlet temperature and the evaporating temperature. Later, many researchers have in-depth investigated different forms of optimal high pressure correlations in transcritical CO_2 cycle [10-14].

Although the offline method is simple and easy to implement, the coefficients of correlations are closely related to the structure of the system. Meanwhile, it was found by Cecchinato, Corradi [15] that the direct use of pressure control correlations might cause huge performance deterioration up to -30% in HPWHs. Furthermore, the research of Liang, He [16] indicates that for CO_2 HPWH, multiple factors will influence the optimal high pressure, resulting in complexity of control correlations.

Given the drawbacks of offline correlations, researchers recently explored a more effective way, called online method. Cecchinato, Corradi [15] proposed that a real-time algorithm is a more effective and robust solution for determining the optimal high pressure. Minetto [17] took the lead in using real-time control algorithm in a CO_2 HPWH. Zhang and Zhang [18] used the steepest descent method to update the set value of the high pressure, and realized the real-time control of the single-stage transcritical CO_2 system by using the PI controller. Later, more comprehensive work has been conducted by researchers on real-time control problems [19-23].

However, the real-time optimization algorithm of on-line method is rather complex, with long time of optimization and high cost of hardware implementation. In a word, there are still many problems unsolved in control of optimal high pressure in transcritical CO_2 system. How to achieve the optimal performance rapidly while ensuring the stability of the system still has not been fundamentally solved.

In this work, a new approach for system optimization of CO_2 HPWH is proposed. From the perspective of charge management, this work aims to approach optimal high pressure in CO_2 HPWH and spare the traditional high pressure control methods. In section 2, a simulation model based on a running CO_2 HPWH was established. In section 3, we quantitively analyzed the relationship between optimal high pressure and refrigerant charge, then proposed charge management approach. In section 4, main factors affecting optimum charge are analyzed. In section 5, two system design methods with the aim of minimizing the change of optimal charge are presented.

2. Modeling and validation

In this section, we tested a running CO_2 HPWH and established its simulation model using GREATLAB [24].

2.1. Test unit

The test unit was officially put into operation in May 2018, and the external influences such as fouling of heat exchangers are negligible. Therefore, the practical characteristics of the CO_2 HPWH can be truly reflected. The nominal heating capacity of this unit is 140kW, with water entering temperature 15°C and water supply temperature 55°C. Fig. 1 shows the schematic of the unit. Tap water is directly heated in the CO_2 HPWH, and then stored in high temperature water tank for further use.

Field test on the unit was conducted in April 2019. Refrigerant parameters and water temperatures were manually collected from the machine's control panel. Besides, in order to gain more information on the evaporator, measurement of evaporator tube wall temperatures was conducted, as shown in Fig. 2. is a summarization of the data collection.

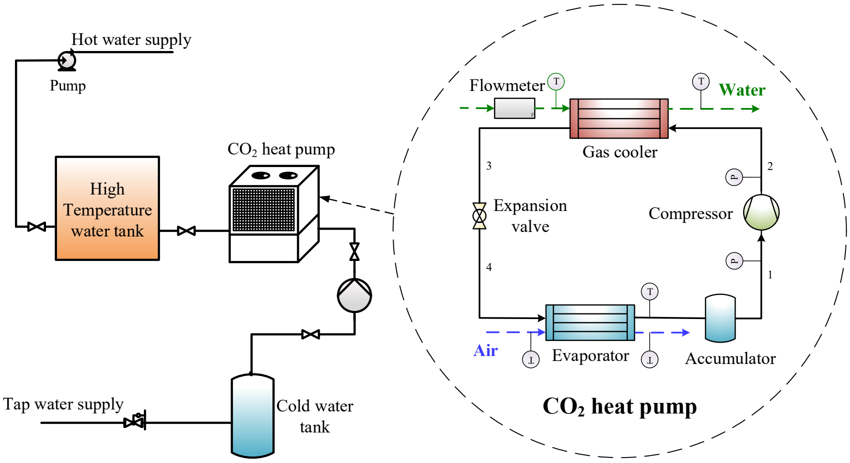


Fig. 1 Schematic of the tested CO_2 HPWH

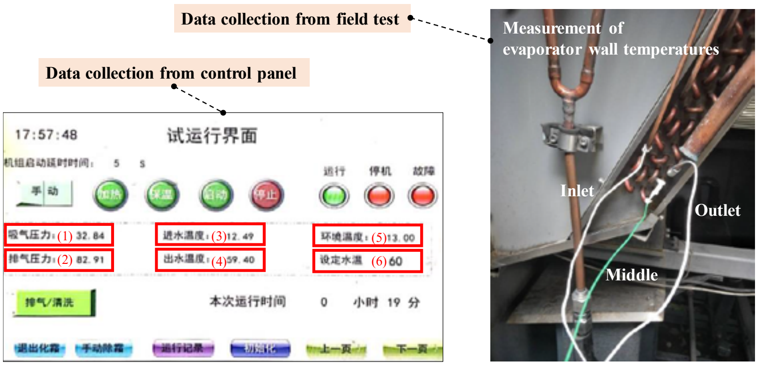


Fig. 2 Field data collection

(1) Suction gauge pressure, 32.84bar. (2) Discharge gauge pressure, 82.91bar. (3) Water entering temperature, 12.49°C. (4) Water leaving temperature, 59.40°C. (5) Ambient temperature, 13.00°C. (6) Set supply water temperature, 60°C.

Table 1 Information of data collection

Parameters	Measurement	Uncertainty
Refrigerant pressures	Panel readings	± 0.5%
Refrigerant /water temperatures		± 0.5°C
Air velocity	Air flow anemometer	± 3%
Dry bulb temperature of air	Humidity & temperature meter	± 1°C
Relative humidity of air		± 3%
Evaporator wall temperatures	Temperature meters	± 0.5°C
Hot water supply	Flow meter	± 3%

2.2. System modeling

2.2.1. Components models

According to the known information, we established a simulation model of the test unit using GREATLAB. GREATLAB is a series of in-house developed software, which focuses on refrigeration, heat pump, and air-conditioning systems modeling and analysis. Detailed information on modeling can be found in the GREATLAB references [24, 25]. Specification of the unit and component models used for system simulation are listed in 오류! 참조 원본을 찾을 수 없습니다..

Tube-by-tube incremental model is adopted for air-cooled finned-tube evaporator modeling. Incremental model can take detailed geometric parameters of evaporator into account, as well as air maldistribution and refrigeration distribution of the coil. Fig. 3 shows the screenshot of finned-tube heat exchangers modeling tool in GREATLAB. The geometric and other operating parameters of the actual heat exchanger are input into the modeling tool to predict the performance of the heat exchangers, including refrigeration or heating capacity, refrigerant and air temperature profiles, refrigerant and airside pressure drops, etc.

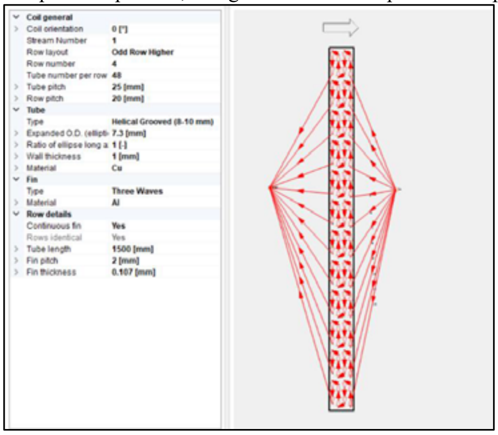


Fig. 3 Screenshot of finned-tube heat exchangers modeling tool

Table 2 Specification of the test unit

Name	Specification	Simulation model type
Refrigerant	CO_2	REFPROP 9.0[26]
Compressor	Reciprocating, single-speed	Curve-fitting model
Evaporator	Finned-tube; 4 rows; 1700mm × 914.4mm × 56mm	Incremental tube-by-tube model
Gas cooler	Brazed plate	1D incremental model
Expansion device	EXV	Curve-fitting model
Fan	4 axial fans for evaporator, single speed	Curve-fitting model

A curve-fitting model was used to describe the performance of single-speed reciprocating compressor. Namely [27],

$$y = c_1 + c_2 p_e + c_3 p_c + c_4 p_e^2 + c_5 p_e p_c + c_6 p_c^2 + c_7 p_e^3 + c_8 p_e^2 T_c + c_9 p_e p_c^2 + c_{10} p_c^3 \quad (1)$$

where y represents power consumption or refrigerant mass flow rate of the compressor. p_e and p_c stand for the suction pressure and discharge pressure for transcritical compression cycles, respectively. Coefficients c_1 - c_{10} are curve-fitted with relevant data of the compressor from manufacturer.

The Electronic expansion valve and fan were modeled using performance curves provided by the manufacturer.

Finally, with all component models in place, we developed the system model in GREATLAB [24], as shown in Fig. 4. The system solver is based on the Newton-Raphson algorithm.

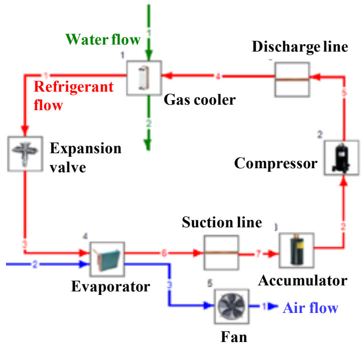


Fig. 4 Screenshot of CO_2 HPWH model in GREATLAB

2.2.2. System charge model

The total refrigerant charge of system is calculated by adding up the charge present in each component and connecting tubes, as described by

$$M = M_{\mathrm{EVAP}} + M_{\mathrm{GC}} + M_{\mathrm{COMP}} + M_{\mathrm{other}} \tag{2}$$

The evaporator is divided into two-phase section and superheat section based on the state of refrigerant. Total charge of the evaporator is calculated by summing up the charge in each discrete element:

$$M_{\mathrm{EVAP}} = M_{\mathrm{TP,e}} + \int_0^{V_{\mathrm{SH,e}}} \rho \mathrm{d}V \tag{3}$$

The total mass in two-phase section can be obtained by

$$M_{\mathrm{TP,e}} = \int_0^{L_{\mathrm{TP}}} \rho_{\mathrm{TP}} A \mathrm{d}L = \int_0^{L_{\mathrm{TP}}} [\alpha \rho_{\mathrm{G}} + (1-\alpha)\rho_{\mathrm{L}}] A \mathrm{d}L \tag{4}$$

where α represents void fraction. ρ_G and ρ_L are the density of gas and liquid, respectively. A stands for the cross-sectional area in pipe.

Therefore, the accuracy of calculated refrigerant charge is closely related to average density in two-phase section, which is determined by void fraction. Void fraction represents the cross-sectional area occupied by vapor,

$$\alpha = A_{\mathrm{G}}/A \tag{5}$$

It is assumed that the liquid and vapor phases are separated into two streams that flow through the tubes with different velocities, u_G, u_L [28]. The void fraction is generally represented as a function of various types of properties. As for refrigerant in-tube flow, it can be described as

$$\alpha = \frac{1}{1+s\left(\frac{1}{x}-1\right)\frac{\rho_{\mathrm{G}}}{\rho_{\mathrm{L}}}} \tag{6}$$

where x represents flowing vapor mass quality, namely,

$$x = \frac{\dot{m}_{\mathrm{G}}}{\dot{m}_{\mathrm{L}}+\dot{m}_{\mathrm{G}}} = \frac{\dot{m}_{\mathrm{G}}}{\dot{m}} \tag{7}$$

S is the slip ratio, estimated differently by various investigators. The zivi method [29] is mostly recommended on the basis of simplicity, where S is given by

$$S = \left(\frac{\rho_G}{\rho_L}\right)^{-\frac{1}{3}} \tag{8}$$

Due to the fluid properties of CO_2, transcritical CO_2 cycle is under high working pressures. The density ratio of gas and liquid is much smaller than traditional refrigerants. Given this, the slip ratio S is considered as a constant value of 1 during the calculations and the model is reduced to a homogeneous one.

In gas cooler, the refrigerant is supercritical single-phase fluid. Thus, the charge is calculated by the 1D incremental model:

$$M_{GC} = \int_0^{V_{GC}} \rho \mathrm{d}V \tag{9}$$

As for the compressor, refrigerant in compressor contains single-phase vapor inside the cylinder and refrigerant dissolved in lubricant oil.

$$M_{COMP} = \rho_{COMP} V_{COMP} + M_{oil} \tag{10}$$

Finally, the refrigerant charges in the connecting tubes are calculated assuming the tubes to be adiabatic.

2.3. Model validation

In order to validate the accuracy and reliability of the simulation model, we conducted a comparison of simulation results and test data under same operating condition. The tap water entering temperature is 12.5°C, water supply temperature is 59.4°C and ambient temperature is DB 11.6°C / WB 9.0°C.

The comparison results are shown in 오류! 참조 원본을 찾을 수 없습니다.. The error between simulation results and test data is less than 5%, which indicates desirable accuracy and reliability of the simulation model.

Table 3 Comparison of simulation results and test data

Parameter / unit	Test	Simulation	Error
Suction pressure / bar	34.4	33.8	-1.8%
Discharge pressure / bar	84.9	83.9	-1.2%
Evaporator entering temperature / °C	4.5	4.7	0.2
Evaporator leaving temperature / °C	10.3	9.6	-0.7
Dry bulb temperature of outlet air / °C	7.2	6.9	-0.3
Hot water supply / L s^{-1}	0.67	0.70	4.3%

On the other hand, the simulation model of CO_2 HPWH built in GREATLAB has also been well validated in our previous work [30]. The model was established to predict monthly power consumption of CO_2 HPWHs, as illustrated in Fig. 5. The predicted results were in good agreement with actual data, with only 0.6% deviation in sum.

Therefore, the simulation model can well predict the performance of CO_2 HPWH and be used for in-depth analysis of the system.

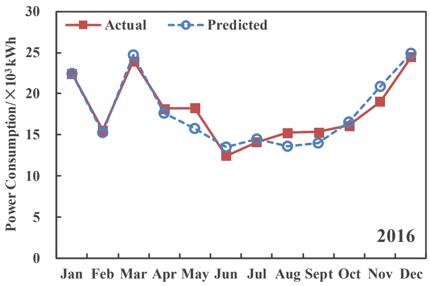


Fig. 5 Comparison between predicted and actual monthly power consumption in 2016 [30]

3. Optimal high pressure and refrigerant charge

In this section, we quantitively investigated the relation of compressor discharge pressure and overall refrigerant charge of system. Then from the perspective of charge management, a new approach for system optimization is proposed.

Based on previous research, the pressure in the high side of transcritical CO_2 cycle is determined by refrigerant charge, inside volume and temperature, as described by [31]

$$p = p(v, T) = p\left(\frac{V}{m}, T\right) \tag{11}$$

As a result, three basic approaches to controlling high pressure can be found in practical transcritical CO_2 systems. However, in most cases, inside volume of high side is fixed and temperature/pressure relation will be influenced by leakage. Thus, high-side charge control is mostly adopted pressure control method.

In systems with high-side charge control, a refrigerant buffer must be provided to ensure the evaporator won't be flooded or dried out. High-side pressure is controlled by adjusting the expansion valve, which will change the balance between mass flow rate of compressor and valve flow rate, thereby changing refrigerant mass in high side. Therefore, controlling high pressure is essentially regulating the amount of refrigerant charge actually running in the system.

Based on this deduction, we made some calculations to investigate the relation of refrigerant charge (charge in accumulator is not counted) and discharge pressure. The results are shown in Fig. 6. As the discharge pressure changes, the COP will reach the maximum value, and the corresponding pressure is the optimal high pressure. Meanwhile, with the increase of discharge pressure, the refrigerant charge of the system increases monotonously. It is indicated that the discharge pressure is in a one-to-one correspondence with refrigerant charge. Therefore, the change in discharge pressure can be truly reflected by the change in the charge amount. Furthermore, the optimal high pressure also corresponds to the optimum charge of the system. That is to say, when the refrigerant charge in the system is optimum, the system is under optimal high pressure.

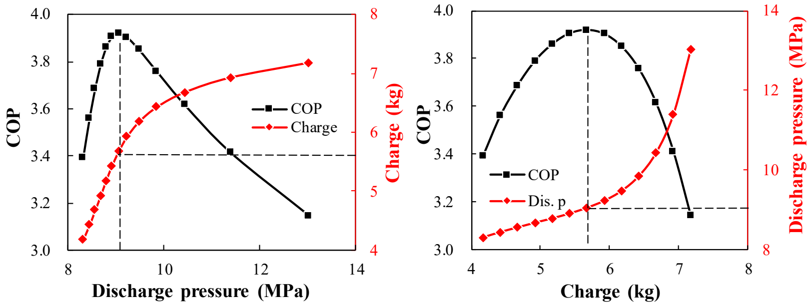


Fig. 6 The relation of optimal high pressure and optimum refrigerant charge

In a word, it was found that for practical systems, optimal high pressure control is fundamentally to manage optimum refrigerant charge of the system. Based on this conclusion, we propose that: if the optimum charge of the system remains constant under any working conditions, then the system can be guaranteed at optimal high pressure with fixed refrigerant charge. In this situation, the control objective of expansion valve switches to superheat rather than discharge pressure, which is the same as subcritical systems. Therefore, if reasonable system design methods are carried out with the aim of keeping optimum charge fixed, then the traditional high pressure control methods can be replaced by superheat control with fixed system charge.

4. Parametric analysis of optimum charge

In this section, parametric study of main factors affecting optimum charge of CO_2 HPWH and their effects is carried out. For a running CO_2 HPWH, ambient temperature, water entering and leaving temperatures are main external factors affecting optimal high pressure. Since frosting has a significant influence on system performance, ambient wet bulb temperature is selected for further analysis. Relevant calculation conditions are shown in **오류! 참조 원본을 찾을 수 없습니다.**. At each working condition, optimum charge of system under which COP reaches maximum value is calculated. To further demonstrate the root cause of the change in optimum charge, schematic diagrams of the change of charge in each component are drawn.

Table 4 Calculation conditions

Group	Ambient temperature (wet bulb) /°C	Relative humidity	Water entering temperature /°C	Water leaving temperature /°C
A	[-25, 25]	0.6	15	55
B	15	0.6	[1, 30]	55
C	15	0.6	15	[40, 90]

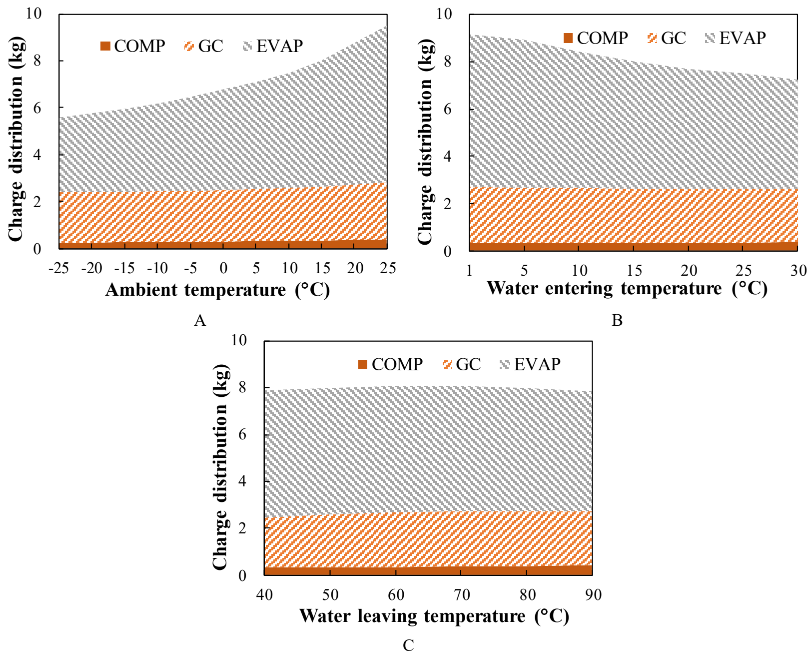


Fig. 7 The change of refrigerant charge in each component

As shown in Fig. 7A, with the increase of ambient temperature, the optimum charge of system increases by 69.6%. The change in evaporator is the main influencing factor, which accounts for 90% of the total increase. The contribution from gas cooler and compressor is 7.8% and 1.8%, respectively. The main cause is that rise of ambient temperature will lead to an increase in the evaporating temperature, resulting in significant density change in evaporator. However, the density change at high pressure side is relatively small, so that charge in gas cooler is almost unchanged.

Similarly, Fig. 7B illustrates the influence of water entering temperature on optimum charge. The optimum charge decreases by 27.9% with the increase of water entering temperature. The contribution of the change in evaporator is 97%, while of gas cooler and compressor is 4.0% and 0.5%, respectively. Since the gas cooler in HPWH is counterflow heat exchanger, an increase in water entering temperature will cause the rise of refrigerant temperature leaving gas cooler. Consequently, vapor quality of CO_2 entering evaporator will increase, resulting in mass decrease in evaporator.

The water leaving temperature has little influence on optimum charge, as shown in Fig. 7C.

In a word, ambient temperature and water entering temperature will both lead to significant change in optimum charge of CO_2 HPWH. Among all components, change in evaporator has the leading effect, while change in gas cooler and compressor have limited influence.

5. System charge management methods

According to the parametric analysis in section 4, the optimum charge of CO_2 HPWH will be influenced by working conditions, of which change in evaporator has major influence. As proposed in section 3, if optimum charge remains constant under any working condition, high pressure control in CO_2 HPWH can be replaced by traditional superheat control with fixed system charge. Therefore, in this section, two system charge management methods with the aim of restraining the change of optimum charge are proposed. Comparison with optimal high pressure system is also conducted.

5.1. Heat exchangers resizing

In the test unit, the evaporator is finned-tube heat exchanger, while gas cooler is compact brazed-plate heat exchanger. Refrigerant stored in evaporator is much more than in gas cooler. Therefore, re-sizing heat exchangers is proposed to improve the distribution of refrigerant in each component, so that the influence of change in evaporator can be restrained. Re-sized heat exchangers should also meet the nominal heating capacity requirement of the CO_2 HPWH.

Two approaches are adopted for re-sizing heat exchangers. On the one hand, reduce the inside volume of evaporator. The evaporator of the test unit is equipped with conventional 7mm heat exchange tube, which is now changed into 5mm tubes. The use of small diameter tubes can reduce charge by 30~40% in heat exchangers. Meanwhile, the capacity is ensured by increasing the number of tubes. Geometric parameters of the original and resized evaporators are listed in **오류! 참조 원본을 찾을 수 없습니다.**.

On the other hand, increase the inside volume of gas cooler. Considering the cost constraints and heat transfer capacity requirements, the number of gas cooler plates is increased by 20%.

Table 5 Geometric parameters of evaporator

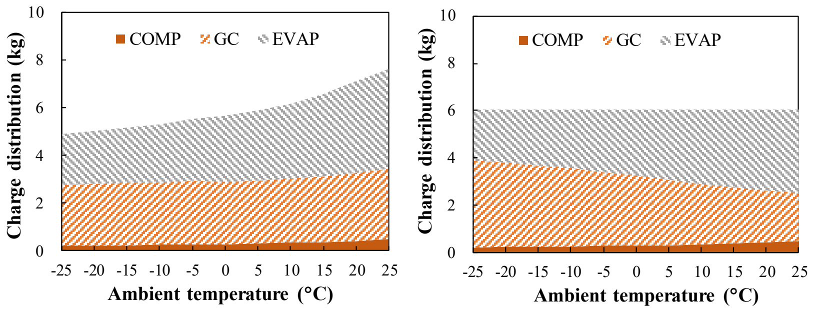
Parameters	Original	Resized
Row number	4	5
Tube number	48	60
Outer diameter	7.45 mm	5.25mm
Tube pinch	25mm	19.05mm
Row pinch	20mm	11.4mm
Length between tube sheets	1.5 m	1.5m
Circuit number	12	30
Flow configuration	Cross-counter flow	Cross-counter flow
Fin pitch	1.4 mm	1.4 mm
Fin thickness	0.115 mm	0.115 mm
Fin type	Wavy	Wavy

Based on the simulation model built in section 2, we established a new system model with superheat control instead of high pressure control. In new model, the overall system charge is fixed and the accumulator is eliminated, thus referred as fixed-charge system. In the following analysis, we quantitively compared the system performance of optimum system (with optimal high pressure control) and fixed-charge system.

Considering that for air-source CO_2 HPWH, ambient temperature varies in a larger range than water entering temperature, it is chosen as independent variable for further analysis. The charge of fixed-charge system is the optimum charge when ambient temperature is 0°C.

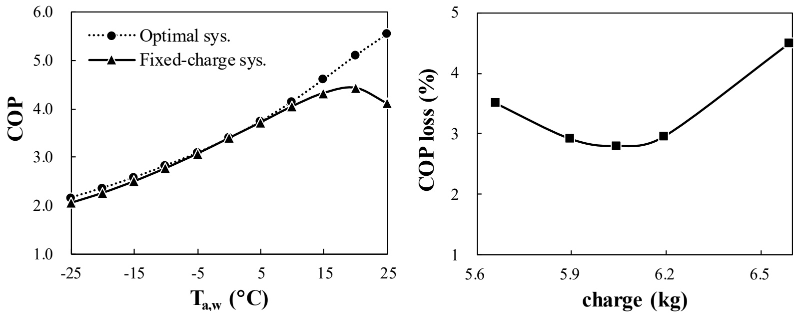
Fig. 8 illustrates the impact of ambient temperature on charge distribution after re-sizing heat exchangers. The relative change of optimum charge is 55.9%, 13.7% less than original system. The comparison of COP is shown in Fig. 9A. When ambient temperature varies from -25-10°C, COP of optimum system and fixed-charge system is very close. However, when ambient temperature is above 10°C, the COP of fixed-charge system drops significantly. Because in fixed-charge system, the evaporator occupies more refrigerant when ambient temperature increasing. It will cause charge in gas cooler far below the optimal value, thus resulting in COP attenuation. In this situation, the average COP loss of fixed-charge system compared with optimum system is 3.51%.

Besides, COP loss could be further restrained by optimizing the charge of fixed-charge system. As shown in Fig. 9B, optimal charge exists for fixed-charge system, where average COP loss is only 2.82%.



A (optimum system) B (fixed-charge system)

Fig. 8 Impact of ambient temperature on charge distribution after re-sizing heat exchangers



A B

Fig. 9 COP comparison between optimum system and fixed-charge system / Impact of refrigerant charge on COP Loss in fixed-charge system

5.2. Additional refrigerant reservoir

However, the performance improvement achieved by re-sizing heat exchangers is limited for fixed-charge system. As analyzed in Section 4, change in evaporator is the predominant factor affecting optimum charge of system. If change in evaporator can be restrained, the optimum charge will remain constant in most situations. Accordingly, adding additional refrigerant reservoir is proposed in this work to make up for the change in evaporator.

Similarly, ambient temperature is chosen as independent variable. The root cause of change in evaporator is the density change affected by the evaporating temperature. Based on parametric analysis, with the increase of ambient temperature, the charge in evaporator will also increase. To offset the change in evaporator, the possible location for an additional component is the inlet and outlet of gas cooler. **오류! 참조 원본을 찾을 수 없습니다.** displays the impact of ambient temperature on density at different locations. With the increase of ambient temperature, the refrigerant density at the inlet of gas cooler and evaporator both increase. However, at the outlet of gas cooler, refrigerant density is rather large and slightly decreases. Therefore, if a reservoir is added at the outlet of gas cooler, overall charge of system will increase that change in evaporator will be weakened. Moreover, the slight decrease of density can make up for the increase in evaporator.

Table 6 Impact of ambient temperature on refrigerant density at different locations

Ambient temperature /°C	Density of CO_2 /kg·m^{-3}		
	Evaporator	GC inlet	GC outlet
-25	37.9	121.9	864.7
-15	50.5	142.2	863.5
-5	66.8	164.4	861.0

5	88.2	189.0	857.8
15	119.7	219.5	853.5
25	171.5	265.0	856.6

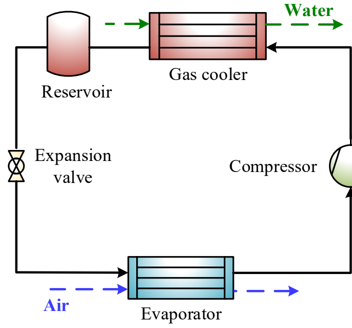


Fig. 10 CO_2 HPWH with additional reservoir at the outlet of gas cooler

Therefore, we added a reservoir at the outlet of gas cooler, as shown in Fig. 10. By changing the volume of the reservoir, we compared the system performance of optimum system and fixed-charge system. As shown in Fig. 11, when volume of high-pressure reservoir increases, the relative change of optimum charge decreases from 72% to 11%, and the average COP loss of fixed-charge system decreases from 6.35% to 0.76%. This also proved that with the decrease of the change of optimum charge, optimal system performance can be ensured by fixing refrigerant charge.

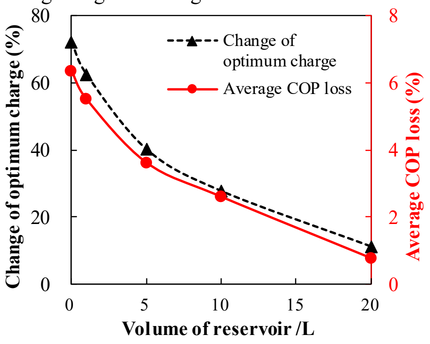
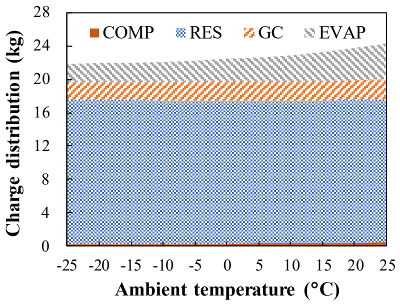
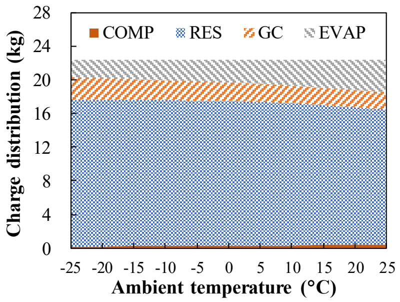


Fig. 11 Impact of reservoir volume on COP Loss and change of optimum charge

Fig. 12 illustrates when the volume of reservoir is 20L, the change of refrigerant charge in optimum system and fixed-charge system. In optimum system, with the increase of ambient temperature, charge in evaporator increases while charge in refrigerant reservoir is in decline. Therefore, the relative change of total charge decreases significantly to only 11.0%. In fixed-charge system, the change in evaporator will be offset by refrigerant reservoir, thus reducing its impact on gas cooler. In this situation, the average COP loss of fixed-charge system is only 0.76%. Moreover, with the increase of volume of reservoir, the average COP loss will be further reduced.



A (optimum system)



B (fixed-charge system)

Fig. 12 The change of refrigerant charge after adding additional opponent

In summary, with reasonable charge management methods, the change of optimum charge can be effectively restrained. Then traditional high-pressure control methods can be replaced by fixing charge for robust and efficient control.

6. Conclusions

In this work, new charge management method is proposed to approach optimal high pressure rather than pressure control in CO_2 HPWH. By developing simulation model on a running CO_2 HPWH, the relation of discharge high pressure and refrigerant charge is thoroughly investigated. At last, two charge management methods are presented. Main conclusions can be drawn as follows.

The high pressure control in practical systems is fundamentally to manage refrigerant charge optimum. If optimum charge remains constant under any working condition, then the system can be guaranteed at optimal high pressure with fixed refrigerant charge.

Ambient temperature and water entering temperature both have noteworthy influence on optimum charge. Among all components, the change in evaporator has decisive effect on the change of optimum charge, accounting for 90% and 97%, respectively.

Two system charge management methods aimed at restraining the variation of optimum charge are proposed. After re-sizing heat exchangers, the change of optimum charge is 13.7% less than original system. The minimum average COP loss of fixed-charge system verses optimum system is 2.82%. After adding a high-pressure reservoir at the outlet of gas cooler, the change of optimum charge is 58.6% less than original system. The minimum average COP loss is only 0.76%.

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