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The Influence of Internal Heat Exchanger on the Performance of Transcritical CO_2 Water Source Heat Pump Water Heater

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Abstract: The characteristics of the transcritical CO_2 heat pump water heater (HPWH) system are; a lower inlet hot water temperature ($T_{i\text{-}hw}$) (sometimes this is lower than the water source temperature), and an outlet gas cooler temperature ($T_{o\text{-}gc}$) which is affected by the $T_{i\text{-}hw}$ and often lower than the critical temperature. In order to study the effects of the internal heat exchanger (IHX) on the operational performance of the transcritical CO_2 HPWH when $T_{o\text{-}gc}$ is low, a transcritical CO_2 water source HPWH experiment platform is established to conduct experimental research and comparative analysis on the operational performance of the transcritical CO_2 water source HPWH, with or without IHX. It is found that, if only the coefficient of performance (COP) and heating at the optimal exhaust pressure of the transcritical CO_2 water source HPWH were considered, COP and the heating of the non-IHX system would be slightly higher than those of the IHX system at the lower hot water flow and water source temperature, and this increase was not obvious. At the higher hot water flow rate and water source temperature, COP and the heating of the non-IHX system were also higher than those of the IHX system, and the increase was obvious. The experiment results showed that, near the optimal exhaust pressure, the variation range of COP and heating of the IHX system is relatively small, and the system has a relatively high stability.

Keywords: transcritical CO_2; heat pump water heater; internal heat exchanger

1. Introduction

Traditional refrigerants, such as CFC and HCFC, have environmental problems such as destroying the ozone layer and causing the greenhouse effect. As a natural refrigerant, CO_2 has great advantages in environmental performance (ozone depletion potential (ODP) = 0, Global warming potential (GWP) = 1), safety performance, effective heat transfer, stable flow performance, abundant source and low cost. At present, CO_2, as a working medium, has been applied in various fields of refrigeration and air conditioning, including automobile air conditioning, water heater, heat pump, cryogenic cascade refrigeration and other fields.

The exhaust temperature of a CO_2 heat pump compressor is high, and its exothermic process is carried out in the supercritical region without phase transformation [1]. The larger temperature slip matches the distribution trend of hot water temperature, which makes it exhibit a unique advantage in the heat pump water heater (HPWH). Loerentzen and co-workers, of NTNU-SINTEF laboratory in Norway, first proposed the CO_2 heat pump system [2–4]. Steven Brown [5] discussed the refrigeration technology, including transcritical carbon dioxide refrigeration. Ignacio López Paniagua et al. [6]

conducted a comparative study between the transcritical CO_2 HPWH and the R410A HPWH, and found that the CO_2 HPWH had better performance and broader prospects. Xiufang Liu et al. [7] built a transcritical CO_2 HPWH experimental bench and found that coefficient of performance (COP) was positively correlated with water flow and water temperature.

However, the most serious disadvantage of a transcritical CO_2 cycle is the low COP caused by the large energy loss of the expansion valve. Generally, the COP can be improved through a two-stage compression [8], the use of the internal heat exchanger (IHX), the use of the absorption chiller [9] and the replacement of throttle valve with expansion equipment such as injector [10], vortex tube [11] and expander [12]. Among them, the most widely used one is the regenerative cycle, and many researchers have conducted relevant studies.

Rigola et al. [13] conducted experimental tests, and simulated the influence on system COP when the ambient temperature was 35 °C and 43 and the IHX length was 0~4.5 m, respectively. The results showed that the proper IHX length improved the COP of the system, and the higher the ambient temperature, the more effective the COP. Torrella et al. [14] experimentally studied the influence of the IHX, and found that the COP could be increased up to 12% with the IHX. Sanchez et al. [15] experimentally studied the system performance when the IHX was installed in different positions. The experimental results showed that the COP increased by up to 13% after using the IHX, and the exhaust temperature of the compressor increased by nearly 20 °C.

Some scholars studied the performance of the IHX system under different application backgrounds. Boewe et al. [16] studied the influence of the IHX on the system performance for automotive air conditioning, and found that the IHX could improve the cooling COP by up to 25%, where the air inlet temperature on the evaporator side was 26.7 °C, the air relative humidity was 40%, and the flow rate was 7.08 $m^3 \cdot min^{-1}$. Cho et al. [17] experimentally studied the system performance for household air conditioning. It was found that the COP was increased by 7~9% after using IHX. Aprea et al. [18] experimentally studied the influence of the IHX on the performance of the transcritical CO_2 air conditioner. The air temperature on the side of the gas cooler was 25~40 °C and the evaporation temperature was 5 °C. The results show that the COP can be increased by up to 10% after using the IHX.

In addition, many scholars considered using IHX and an ejector together, and verified the system performance through theoretical methods and experimental methods, respectively. In a wide range of simulation results, Zhang et al. [19] found that the ejection rate of the system of the IHX and injector would increase with the introduction of the IHX, and the pressure increase would decrease, so the overall COP value of the system was not necessarily changed. However, in the experimental study, Masafumi et al. [20] found that both the efficiency of the injector and the COP of the overall system would be improved with the introduction of the IHX. The difference between theoretical and experimental results is attributed to the adoption of a large number of hypotheses and simplifications in theoretical research [21].

At the same time, many scholars are committed to using numerical research method to study the performance of the transcritical CO_2 cycle with IHX. Chen et al. [22] established the theoretical model, and they studied the influence of IHX on the system performance. They pointed out that IHX could improve the COP, and the higher the ambient air temperature, the greater the improvement effect. Yudonago et al. [23] found that inlet temperature and mass flow could significantly affect the working efficiency of the IHX, especially the inlet temperature and mass flow on the high-pressure side, which was caused by the special physical properties of the CO_2 working medium in the supercritical zone.

Besides, Zhang et al. [24] theoretically studied the influence of IHX and found that when the exhaust pressure was lower than the turning point pressure and the outlet gas cooler temperature ($T_{o\text{-}gc}$) was higher than the turning point temperature, the COP could be improved by using IHX. The $T_{o\text{-}gc}$ was kept at 37.5 °C during the experiment and the results showed that only when the exhaust pressure was lower than the turning point pressure could the COP be improved by using the IHX, and the increase was not evident. Similar results were also found in the study of Kim et al. [25], who proved that the system had a positive effect when the IHX was introduced at low exhaust pressure, but not at high exhaust pressure.

Notably, Rodrigo Llopis et al. [26] found that the internal heat exchanger does not improve the performance of the subcritical cycle, but it could improve the energy performance if it is used inside a cascade refrigeration system. Shariatzadeh et al. [27] found that even though the introduction of IHX always has a positive impact on the performance, IHX definitely has a negative impact on the exergy efficiency. Nilesh Purohit et al. [28] studied the effects of IHX on system performance from an energetic and exergetic perspective. The experimental results demonstrate the advantage of the adoption of IHX in a high ambient temperature for a chiller application.

The results of these studies indicate that whether it is beneficial to use IHX in the transcritical CO_2 HPWH needs further research and analysis, to improve the operating performance and provide reliable technical support for the optimal design and efficient operation of the transcritical CO_2 HPWH.

2. Experiment Test

The experimental bench of the transcritical CO_2 water source HPWH system is designed and established, as illustrated in Figure 1. The experimental test system is composed of the transcritical CO_2 source HPWH system, the water system for heat exchange with the transcritical CO_2 source HPWH system, and the data measurement and acquisition system. The transcritical CO_2 HPWH system is mainly composed of the fully enclosed rolling rotor compressor, the gas cooler, the electronic expansion valve (EEV), the evaporator, the gas-liquid separator and the IHX. The data measurement and acquisition system is mainly composed of the temperature sensor, pressure sensor, flowmeter, power meter, paperless recorder and DC power supply.

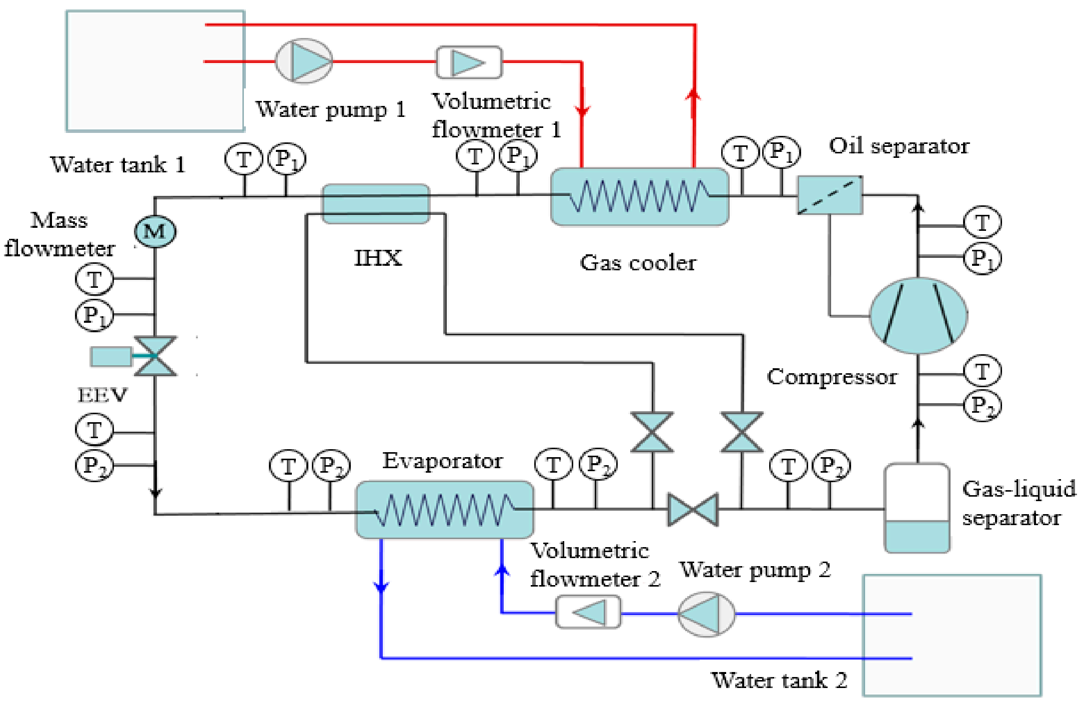


Figure 1. Schematic diagram of a transcritical CO_2 water source heat pump water heater (HPWH) system.

T-type thermocouples were adopted for temperature measurement, and the standard platinum resistance was used for the calibration of all thermocouples. The precision after calibration was 0.2 °C, and the measurement range was from −10 °C to 150 °C. A diffused silicon pressure transmitter was used to measure the pressure. According to the working pressure range of the HPWH system, two specifications of pressure transducer were used. The measurement range of specification 1 was 0~6 MPa, and that of specification 2 was 0~16 MPa, with the measurement accuracy of ±0.25%.

The compressor power was equipped with the QZ8716C1 power meter (Qingzhi Instruments, Qingdao, China), and the precision of the power meter was 0.1% range +0.4% reading error. The flow rate of water was measured by the turbine volume flowmeter, and the model of this was LWGYC (Beijing Flowmeter Factory, Beijing, China). According to the different water flow rate, two kinds of turbine volume flowmeters were adopted. The accuracy of both volume flowmeters was ±0.5 %, and the measuring range was 0.06~0.6 $m^3 \cdot h^{-1}$ and 0.6~6.0 $m^3 \cdot h^{-1}$, respectively. The mass flow rate of CO_2 was measured by the SITRANS F C coriolis mass flowmeter of Siemens, with a range of 0~5600 $kg \cdot h^{-1}$ and an accuracy of ±0.1%.

3. Results and Analyses

3.1. Performance Comparison of HPWH under Different Working Conditions

Keeping the inlet flow and temperature of water source and inlet hot water temperature ($T_{i\text{-}hw}$) and flow of hot water unchanged, the opening of EEV was realized by adjusting the input voltage of the pulse signal converter, which is 10–100%, and obtained the comparison of performance parameters of the heat pump system, with or without IHX, under different exhaust pressures. Specific experimental conditions are shown in Table 1.

Table 1. Experimental test condition.

Project	Condition
Water inlet temperature/°C	20
Water inlet flow/$m^3 \cdot h^{-1}$	1.5
Hot water inlet temperature/°C	15
Hot water inlet flow/$m^3 \cdot h^{-1}$	0.25

Figures 2–4 show the comparison of performance parameters, such as suction pressure, mass flow rate of CO_2, $T_{o\text{-}gc}$ of CO_2, outlet temperature of hot water ($T_{o\text{-}hw}$), heat production, compressor power and COP of HPWH, with or without IHX.

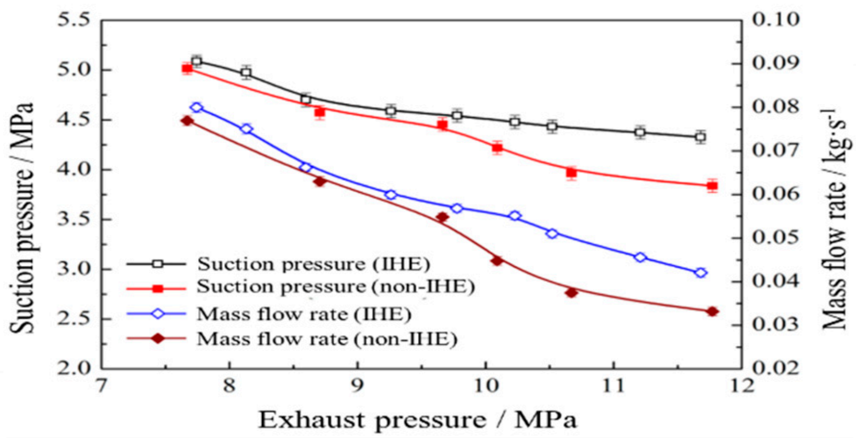


Figure 2. Variation of suction pressure and CO_2 mass flow.

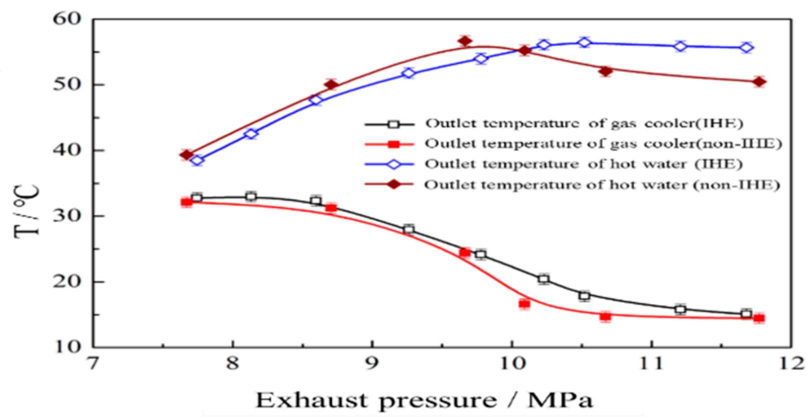


Figure 3. Outlet gas cooler temperature ($T_{o\text{-}gc}$) and outlet hot water temperature ($T_{o\text{-}hw}$) change with exhaust pressure.

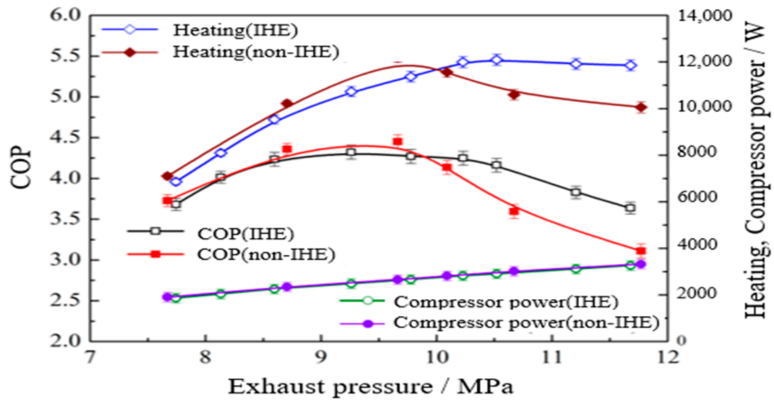


Figure 4. Comparison of heating, compressor power and coefficient of performance (COP).

Figure 2 shows that the suction pressure of the non-IHX system is lower than that of the IHX system. When the exhaust pressure is low, the difference of inspiratory pressure is small, and with the increase of the exhaust pressure, the difference is increasingly larger. This is because after the heat pump system is equipped with the IHX, the internal volume of the system increases and the amount of CO_2 charge increases. Under the same external operating conditions, when the exhaust pressure is the same, the CO_2 accumulation in the evaporator of the IHX system is more than that of the non-IHX system. Therefore, the evaporative pressure is higher than that of the non-IHX system, resulting in a suction pressure higher than that of the non-IHX system. When the exhaust pressure continues to rise, the accumulation of CO_2 in the gas cooler increases, which leads to the relative difference of CO_2 accumulation in the evaporator, with or without the IHX system, so that the difference of inspiratory pressure becomes increasingly larger.

In Figure 2, the mass flow of CO_2 of the IHX system is larger than that of the non-IHX system, mainly due to the fact that the suction pressure of the IHX system is higher under the same exhaust pressure. For the transcritical CO_2 HPWH system of the constant frequency compressor, the suction pressure is higher and the specific capacity of suction is smaller, so the mass flow rate of the compressor is larger when the same exhaust pressure is used.

In Figure 3, for the non-IHX system, $T_{o\text{-}gc}$ decreases continuously as the exhaust pressure increases. When it descends to a certain point, it basically stays the same. Since $T_{i\text{-}hw}$ remains unchanged at 15 °C, when $T_{o\text{-}gc}$ approaches $T_{i\text{-}hw}$, it can only approach 15 °C and cannot continue to decrease. For the IHX system, $T_{o\text{-}gc}$ is decreasing, and as it is reduced to a certain degree, it is also basically unchanged, and the variations of the basic system and the non-IHX system are consistent. The difference is that, when the exhaust pressure is low, $T_{o\text{-}gc}$ in the IHX system is slightly higher than that of the non-IHX heat pump system. This is due to the fact that the CO_2 mass flow of the IHX heat pump system is slightly higher than that of the non-IHX heat pump system.

In Figure 4, it can be seen that with the increase of the exhaust pressure, the compressor power increases gradually in both the basic cycle and the IHX cycle, and the rising trend is close to a linear relationship. This is caused by the reduced mass flow of CO_2 and the increased differential pressure between the exhaust pressure and the suction pressure. The compressor power of the non-IHX system is slightly higher than that of the IHX system. Heating is gradually increased to the extreme value, and gradually decreased with the increase of the exhaust pressure. The extreme value of heating in the basic cycle is higher than that in the IHX cycle, since $T_{o\text{-}gc}$ decreases continuously as the exhaust pressure rises. The heating produces a relatively significant reduction when the heating reaches the maximum value in the non-IHX system. This is due to the fact that the evaporation temperature and refrigerant flow continue to decrease as the cooling outlet temperature is close to $T_{i\text{-}hw}$ (15 °C). However, after the heating of the IHX system reaches the maximum value, it slowly decreases and the exhaust pressure at the maximum value is higher than that of the non-IHX system. The reason is that the $T_{o\text{-}gc}$ is slowly approaching $T_{i\text{-}hw}$ (15 °C) in the IHX system.

In addition, for the basic cycle, the COP of the system reaches the maximum at the optimal exhaust pressure, and the system heating also reaches the maximum at the optimal exhaust pressure. For the IHX cycle, at the optimal exhaust pressure, the COP of the system reaches the maximum value, but the heating of the system is not the maximum value. As the exhaust pressure increases, the heating and compressor power consumption increases to different degrees.

The optimal COP of the basic cycle is higher than that of the IHX cycle. With the increase of exhaust pressure, the mass flow of CO_2 decreases continuously, and $T_{o\text{-}gc}$ decreases continuously until approaching $T_{i\text{-}hw}$ (15 °C). The difference is that the $T_{o\text{-}gc}$ drops faster in the basic cycle, resulting in a higher supercooling at the optimal exhaust pressure, which leads to larger heating and higher COP. Therefore, at the optimal exhaust pressure, the IHX cycle reduces the COP. When the exhaust pressure of the basic cycle is higher than the optimal exhaust pressure, the evaporation temperature decreases obviously and is lower than that of the IHX cycle. A decrease in the evaporation temperature leads to a decrease in the COP, so the COP of the basic cycle starts to be lower than that of the IHX cycle as the exhaust pressure increases.

Table 2 shows the comparison of the optimal exhaust pressure, COP and differential pressure fluctuation values of the IHX system. Among them, the fluctuation values of pressure difference refer to the difference values of the exhaust pressure, as the optimal COP decreases by 5%. When the optimal COP is reduced by 5%, there is a pressure value on each side of the optimal exhaust pressure. Additionally, the difference from the optimal pressure is the right differential pressure and the left differential pressure, respectively. The value directly reflects the fluctuation situation of COP when the exhaust pressure changes near the optimal exhaust pressure, that is, the stability performance of the system near the optimal exhaust pressure.

Table 2. Comparison of optimal exhaust pressure and COP of a heat pump with or without internal heat exchanger (IHX).

Project	IHX	Non-IHX	Contrast
The optimal pressure/MPa	9.3	9.7	–0.4
The optimal COP	4.32	4.45	3%
Fluctuation value of left differential pressure/MPa	0.9	1.1	–0.2
Fluctuation value of right differential pressure/MPa	1.4	0.3	1.1
Total differential pressure fluctuation value/MPa	2.3	1.4	0.9

It can be seen from Table 2 that for the IHX system, when the optimal COP decreases by 5%, the corresponding exhaust pressure changes by 2.3 MPa. In contrast, for the non-IHX system, the corresponding exhaust pressure changes by 1.4 MPa when the optimal COP decreases by 5%. Therefore, near the optimal exhaust pressure, the operating stability of the non-IHX system is much worse than that of the IHX system, and it can also be clearly seen from Figure 4 that the COP of the non-IHX system decreases significantly near the optimal exhaust pressure.

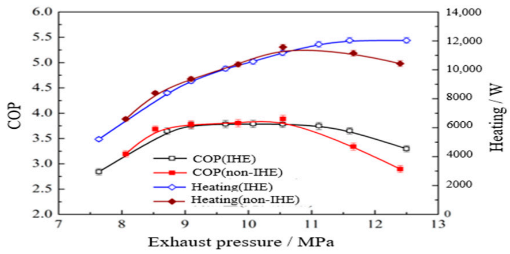
3.2. Performance Comparison of HPWH with Different Hot Water Flow

By keeping the inlet flow, the temperature of water, and $T_{i\text{-}hw}$ unchanged, we changed the inlet flow of hot water and the opening of EEV, and made comparison of various performance parameters of the heat pump system, with or without IHX under different inlet flows of hot water. Table 3 shows the experimental test conditions

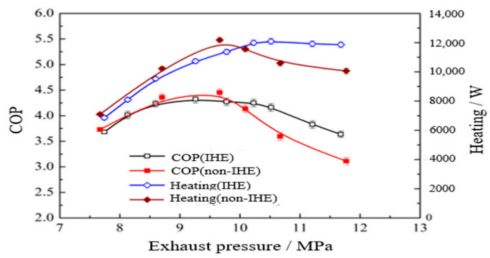
Table 3. Experimental test condition.

Project	Condition
Water inlet temperature/°C	20
Water inlet flow/$m^3 \cdot h^{-1}$	1.5
Hot water inlet temperature/°C	15
Hot water inlet flow/$m^3 \cdot h^{-1}$	0.20, 0.25, 0.30

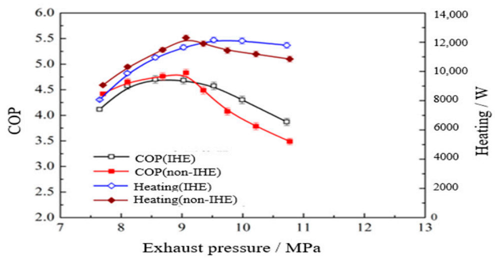
Figures 5 and 6 show the comparison of heat production, $T_{o\text{-}hw}$ and COP of HPWH, with or without IHX, under different hot water inlet flow.



(**a**) Hot water flow is 0.20 $m^3 \cdot h^{-1}$.

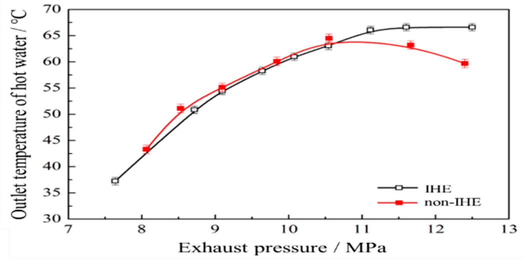


(**b**) Hot water flow is 0.25 $m^3 \cdot h^{-1}$

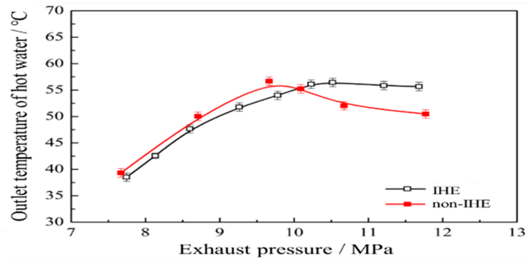


(**c**) Hot water flow is 0.30 $m^3 \cdot h^{-1}$

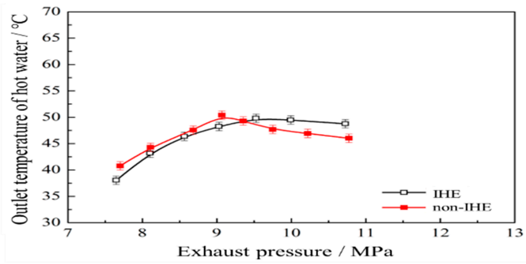
Figure 5. Comparison between heating and COP with and without IHX at different hot water flow rates.



(**a**) Hot water flow is 0.20 $m^3 \cdot h^{-1}$



(**b**) Hot water flow is 0.25 $m^3 \cdot h^{-1}$



(**c**) Hot water flow is 0.30 $m^3 \cdot h^{-1}$

Figure 6. Comparison of outlet temperature of hot water with and without IHX at different hot water flow rates.

From Figure 5, as the exhaust pressure keeps rising, the heating of both the IHX system and the non-IHX system keeps rising, reaching the maximum, and then gradually decreases. After the heating of the IHX system reaches the maximum value, it slowly decreases with the increase of the exhaust pressure. However, the heating of the non-IHX system decreases significantly with the increase of the exhaust pressure when it reaches the maximum value. In the non-IHX system, when the heating reaches the maximum, $T_{o\text{-}gc}$ is close to $T_{i\text{-}hw}$, and thereafter, the suction pressure continues to decrease when the exhaust pressure continues to rise. Due to the role of the IHX, heat exchange occurs between the CO_2 high-pressure working medium at the outlet of the gas cooler, and the low-pressure steam at the outlet of the evaporator when $T_{o\text{-}gc}$ is close to the $T_{i\text{-}hw}$, thus resulting in the process where the heat production is slowly reduced with the increase of exhaust pressure.

As the exhaust pressure keeps rising, the COP of both the IHX system and the non-IHX system keeps rising, reaching the maximum, and then gradually decreases, indicating that there is an optimal COP. The difference is that when COP of the non-IHX system reaches the optimal value, it decreases significantly with the increase of the exhaust pressure. This is because in the non-IHX system, $T_{o\text{-}gc}$ is close to $T_{i\text{-}hw}$ when COP reaches the optimal value; after that, $T_{o\text{-}gc}$ is almost unchanged and the inlet pressure continues to decrease as the exhaust pressure continues to rise.

In addition, the optimal exhaust pressure of the IHX system is lower than that of the non-IHX system. The optimal COP of the non-IHX system is slightly higher than that of the IHX system, and the increasing range tends to decrease with the decrease of the hot water inlet flow. The reason is that when the hot water inlet flow decreases, the heat transfer effect at the water side of the gas cooler becomes worse, $T_{o\text{-}gc}$ increasing, and the influence of the IHX on COP reduction begins to be weakened.

Figure 6 shows that, as the exhaust pressure keeps rising, $T_{o\text{-}hw}$ of both the IHX system and the non-IHX system keeps rising, reaching the maximum, and then gradually decreases. After $T_{o\text{-}hw}$ reaches the maximum, the IHX system begins to slowly decrease with the increase of the exhaust pressure, but the non-IHX system decreased significantly. Heat production is affected by the temperature difference and the flow rate of hot water inlet and outlet. When $T_{i\text{-}hw}$ and flow rate of hot water remain unchanged, the trend of $T_{o\text{-}hw}$ is similar to that for heat production.

Table 4 compares the main performance parameters of the HPWH system with and without IHX, at different hot water inlet flows. It can be seen that the optimal exhaust pressure of the HPWH system without IHX is higher than that of the HPWH system with IHX, about 0.4~0.5 MPa higher. The optimal COP of the non-IHX system is slightly higher than that of the IHX system and the increasing rate tends to decrease with the decrease of the hot water inlet flow, but the overall increase rate changes insignificantly. As the hot water inlet flow was 0.3 $m^3 \cdot h^{-1}$ and 0.2 $m^3 \cdot h^{-1}$, the increase was 3.4% and 2.4%, respectively. When the hot water inlet flow decreases, the heat transfer effect at the water side of the gas cooler becomes worse and $T_{o\text{-}gc}$ increases. Therefore, the influence of the IHX on COP reduction begins to be weakened.

Table 4. Comparison of main performance parameters of HPWHs with and without IHX at different hot water inlet flows.

Project	0.3 $m^3 \cdot h^{-1}$		0.25 $m^3 \cdot h^{-1}$		0.20 $m^3 \cdot h^{-1}$	
	IHX	Non-IHX	IHX	Non-IHX	IHX	Non-IHX
The optimal exhaust pressure/MPa	8.6	9.1	9.3	9.7	10.1	10.6
The optimal COP	4.71	4.87	4.32	4.45	3.79	3.88
Fluctuation value of left differential pressure/MPa	0.5	1.0	0.9	1.1	1.4	1.9
Fluctuation value of right differential pressure/MPa	1.2	0.2	1.4	0.3	1.6	0.3
Total differential pressure fluctuation value/MPa	1.7	1.2	2.3	1.4	3.0	2.2
Heating at optimal pressure/kW	11.0	12.3	10.7	12.2	10.6	11.6
To-hw at optimal pressure/°C	46.4	50.4	51.8	56.7	61.0	64.5
Maximum heating/kW	12.2	12.3	12.1	12.2	12.0	11.6
Maximum To-hw/°C	49.8	50.4	56.5	56.7	66.6	64.5
Exhaust pressure at the extreme value of heating/MPa	9.5	9.1	10.5	9.7	11.6	10.6

Near the optimal exhaust pressure, the heating of the non-IHX system is also nearly the maximum. The heating of the IHX system is not the maximum at the optimal exhaust pressure, because with the increase of pressure, both heating and power consumption increase, but the rate of increase is different. In the vicinity of the optimal exhaust pressure, the heating and COP have a relatively small variation range with the exhaust pressure and relatively high stability, which is the advantage of HPWH system using IHX. Secondly, with the decrease of the hot water inlet flow, COP decreased significantly, while the temperature of the hot water outlet increased significantly. In addition, the fluctuation value of the total pressure difference of the HPWH system is greatly increased, that is, the change range of heating and COP is relatively smaller with the exhaust pressure, and the stability is relatively higher.

3.3. Performance Comparison of HPWH at Different Water Source Temperatures

By keeping the inlet flow rate of water source, $T_{i\text{-}hw}$ and the flow rate of hot water unchanged, changing the inlet temperature of water ($T_{i\text{-}w}$) and adjusting the opening degree of EEV, the comparison of performance parameters of the heat pump system with and without IHX can be obtained under different inlet temperatures of water source. The experimental conditions are shown in Table 5.

Table 5. Experimental test condition.

Project	Condition
Water inlet temperature/°C	10, 15, 20, 25
Water inlet flow/$m^3 \cdot h^{-1}$	1.5
Hot water inlet temperature/°C	15
Hot water inlet flow/$m^3 \cdot h^{-1}$	0.25

Figures 7 and 8 show the comparison of heating, $T_{o\text{-}hw}$ and COP of HPWH, with or without IHX at different inlet temperatures of the water source. Figure 7 shows that, as the exhaust pressure keeps rising, the heat output of the system with or without IHX keeps rising, reaching the maximum, and then gradually decreases. After the heating reached the maximum value, the heating of the IHX system began to decrease slowly with the increase of exhaust pressure, but the heating of the non-IHX system decreased relatively significantly.

At the same $T_{i\text{-}w}$, the maximum heating of the IHX system and the non-IHX system are not distinctly different, because the main role of the IHX is to realize heat exchange within the system. As $T_{i\text{-}w}$ increases, the maximum heating of both the IHX system and the non-IHX system increases significantly, because the rising $T_{i\text{-}w}$ leads to a rising evaporation temperature.

As the exhaust pressure keeps rising, the COP of both the IHX system and the non-IHX system keeps rising, reaching the maximum, and then gradually decreases, indicating that there is an optimal COP. When COP reached the optimal value, COP with IHX system began to decrease slowly with the increase of the exhaust pressure, but the COP of the non-IHX system decreased significantly. Under the four water inlet temperatures, the COP comparison showed similar variations.

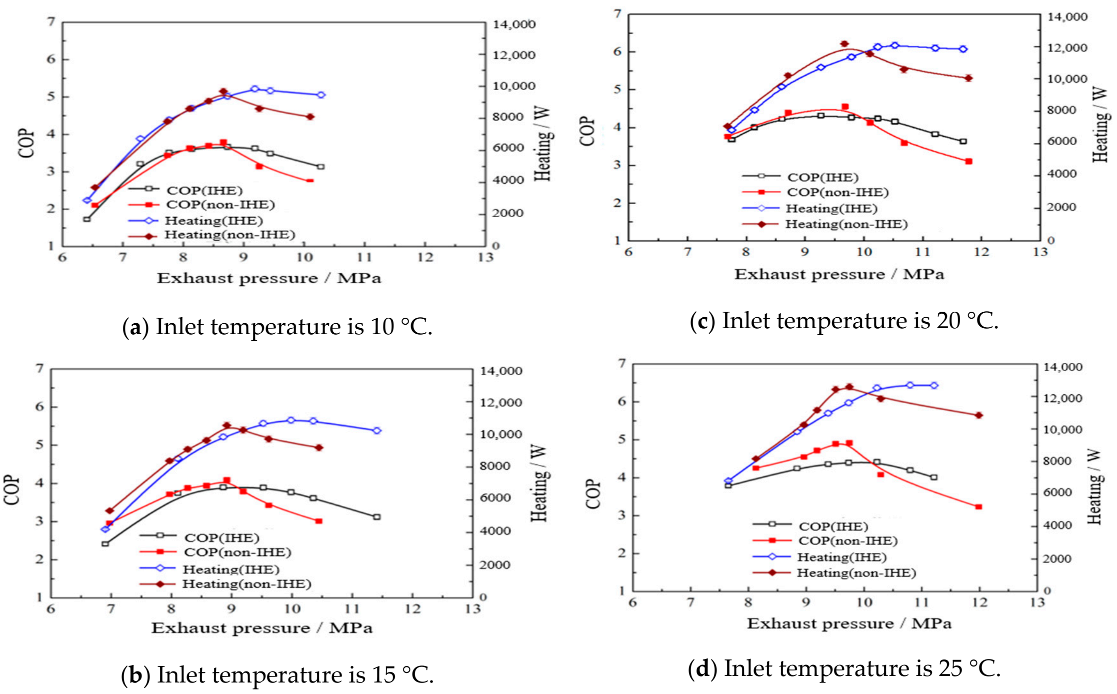
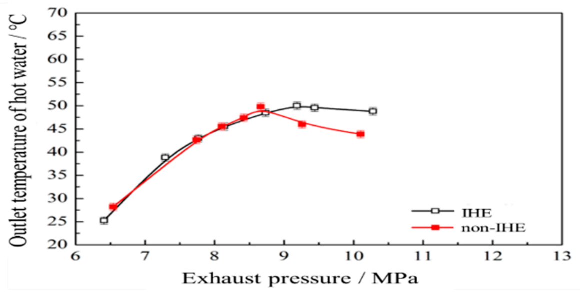
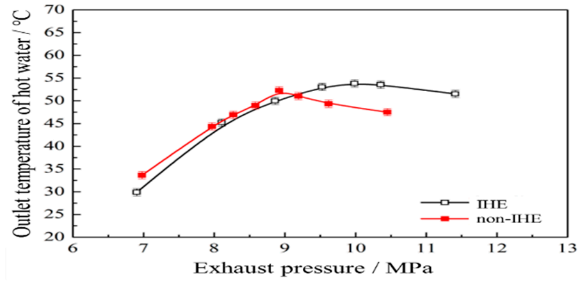


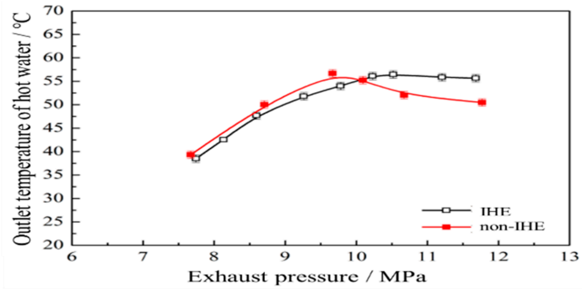
Figure 7. Comparison between heating and COP with and without IHX at different inlet temperatures of water source.



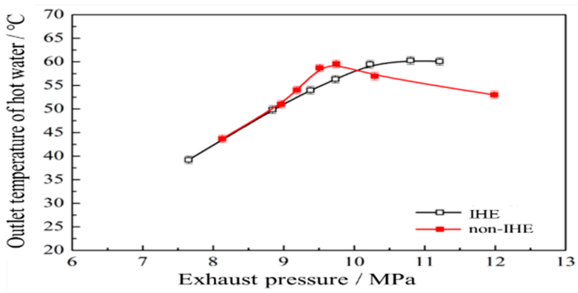
(**a**) Inlet temperature is 10 °C.



(**b**) Inlet temperature is 15 °C.



(**c**) Inlet temperature is 20 °C.



(**d**) Inlet temperature is 25 °C.

Figure 8. Comparison of outlet temperature of hot water with and without IHX at different inlet temperatures of water source.

The optimal COP of the non-IHX system is slightly higher than that of the IHX system, and the increasing range tends to increase with the increase of $T_{i\text{-}w}$. When the inlet water temperature was 10 °C and 25 °C respectively, the optimal COP of the non-IHX system was improved by 3.3% and 11.3%, respectively, compared with the optimal COP of the IHX system. For the HPWH, when $T_{i\text{-}w}$ is high, especially higher than $T_{i\text{-}hw}$, the IHX will increase the EEV inlet fluid temperature and reduce the compressor inlet temperature, resulting in the reduction of system heat production, leading to the reduction of COP. The higher the $T_{i\text{-}w}$, the more obvious the effect. Due to the characteristics of the application site, $T_{i\text{-}w}$ of the HPWH may be higher than $T_{i\text{-}hw}$, sometimes even much higher. This is the difference between HPWH and the normal heat pump.

Since the inlet temperature and flow rate of hot water remain unchanged, the change law of $T_{o\text{-}hw}$ is consistent with that of heating. From Figure 8, as the exhaust pressure continues to rise, $T_{o\text{-}hw}$ in both IHX and non-IHX systems continue to rise, reaching a maximum, and then gradually decrease.

Table 6 shows the main performance parameters of HPWH system with or without IHX under different $T_{i\text{-}w}$. When $T_{i\text{-}w}$ is low, the optimal exhaust pressure of the non-IHX system is slightly higher than that of the IHX system. However, when $T_{i\text{-}w}$ is high, the optimal exhaust pressure of the non-IHX system is lower than that of the IHX system, which is determined by the relationship between $T_{i\text{-}w}$ and $T_{i\text{-}hw}$.

Table 6. Comparison of main performance parameters of HPWH system with and without IHX at different hot water inlet flows.

Project	10 °C		15 °C		20 °C		25 °C	
	IHX	Non-IHX	IHX	Non-IHX	IHX	Non-IHX	IHX	Non-IHX
The optimal exhaust pressure/MPa	8.7	8.8	8.9	9.0	9.3	9.7	10.23	9.8
The optimal COP	3.67	3.79	3.9	4.09	4.32	4.45	4.42	4.92
Fluctuation value of left differential pressure/MPa	1.0	0.7	0.6	0.60	0.9	1.1	1.5	0.6
Fluctuation value of right differential pressure/MPa	0.7	0.1	1.3	0.20	1.4	0.3	0.6	0.2
Total differential pressure fluctuation value/MPa	1.7	0.8	1.9	0.80	2.3	1.4	2.1	0.8
Heating at optimal pressure/kW	9383	9688	9849	10550	10731	12179	12525	12594
$T_{o\text{-}hw}$ at optimal pressure/°C	47.3	48.4	48.9	50.8	51.8	56.7	58.2	57.8
Maximum heating/kW	9849	9688	10871	10550	12091	12179	12698	12594
Maximum $T_{o\text{-}hw}$/°C	49.1	48.4	52.5	50.8	56.5	56.7	58.9	57.8
Exhaust pressure at the extreme value of heating/MPa	9.2	8.8	9.99	8.9	10.5	9.7	10.8	9.8

Near the optimal exhaust pressure, the heating and COP of the non-IHX system vary greatly with the exhaust pressure, especially as the exhaust pressure is slightly higher than the optimal exhaust pressure, indicating that the stability of the system is poor. On the contrary, the IHX system has a relatively small variation range of heating and COP, and its stability is relatively high

Besides, with the increase of the $T_{i\text{-}w}$, the maximum heating and the optimal COP increased significantly. The increase of Ti-w leads to the increase of the evaporation temperature, which leads to the increase of the suction pressure. As the suction pressure continues to rise, the exhaust pressure continues to rise, which leads to the significant increase of heating and COP.

4. Conclusions

In this paper, a transcritical CO_2 water source HPWH experiment bench was established to conduct experimental research and comparative analysis on the operation performance of transcritical CO_2 water source HPWH, with or without IHX. The results showed that:

(1) When the $T_{i\text{-}hw}$ was lower (15 °C), the optimal COP of non-IHX system was slightly higher than that of the IHX cycle. When the exhaust pressure was lower than the optimal exhaust pressure, the COP of the non-IHX system was slightly higher than that of the IHX system. When the exhaust pressure was higher than the optimal exhaust pressure, COP of the non-IHX system decreased significantly and was gradually lower than that of the IHX system. At the optimal exhaust pressure, the non-IHX system has higher supercooling, which leads to higher COP. When the exhaust pressure of the basic cycle is higher

than the optimal exhaust pressure, the evaporation temperature decreases obviously and is lower than that of the IHX cycle. A decrease in the evaporation temperature leads to a decrease in the COP, so the COP of the basic cycle starts to be lower than that of the IHX cycle as the exhaust pressure increases.

(2) For the basic cycle, the COP of the system reaches the maximum at the optimal exhaust pressure, and the heating of the system also reaches the maximum approximately. For the IHX system, the COP reached the maximum value at the optimal exhaust pressure, but the heat generation did not reach the maximum value. Instead, the heat generation reached the maximum value at a point slightly higher than the optimal exhaust pressure. The COP is the ratio of heating to power consumption. With the increase of pressure, both heating and power consumption increase, but the rate of increase is different, so the heating is not the maximum when COP is maximized.

(3) Near the optimal exhaust pressure, the heating and COP of the non-IHX system varied greatly with the exhaust pressure, and the stability was relatively poor. However, the variation range of heating and COP in the IHX system was relatively small, and the stability was relatively high.

(4) If COP and heating at the optimal exhaust pressure of the transcritical CO_2 water source HPWH were only considered, COP and heating of the non-IHX system were slightly higher than that of the IHX system at the lower hot water flow and water source temperature, and the increase was insignificant. At the high hot water flow rate and water source temperature, COP and heating of the non-IHX system were also higher than that of the IHX system, and the increase was more obvious.

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