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School of Mechanical Engineering

2014

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Field Evaluation for Air-source Transcritical CO_2 Heat Pump Water Heater with Optimal Pressure Control

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ABSTRACT

Several optimal discharge pressure correlations for transcritical CO_2 heat pump in literature have been discussed in this paper. Most of them are related to the ambient temperature, the evaporation temperature and the gas cooler outlet temperature. A revised optimal discharge pressure model as the function of ambient temperature and water outlet temperature is developed based on earlier experimental results. To validate the model developed and control effect in practical applications, field tests are conducted to evaluate the performance of an air-source transcritical CO_2 heat pump water heater in practical application. The results show that the coefficient of performance (COP) can achieve 3.76 in the nominal test condition with 15℃ water inlet temperature and 80℃ hot water supply temperature. Even when the hot water temperature is higher than 90℃, the COP remains at 3.21 with 20℃ dry-bulb temperature and 15℃ wet-bulb temperature. Under -15℃ low ambient air temperature condition, the COP was 2.19 with the hot-water supply temperature of 60℃. Comparison between the field test results and the model predictions show that the maximum relative error of discharge pressure control was 3.2% in the high temperature of water outlet condition. The overall ATHP system performance based on the revised model and PLC control strategy can meet the requirements in practical applications.

Keywords: Transcritical, Field Test, CO_2, Heat Pump, Optimal Discharge Pressure, COP

1. INTRUDUCTION

As a natural refrigerant, carbon dioxide is a kind of promising replacement substance for traditional refrigerants in air conditioning and heat pump systems. However, the critical temperature point of carbon dioxide is low and corresponding operating pressure is high compared with other refrigerants (Lorentzen, 1994; Lorentzen, 1995). In practice, the components of CO_2 refrigeration system must be able to undertake high pressure and meet the safety requirements. So at one time, carbon dioxide was replaced by synthetic refrigerants and stepped down from the stage of history gradually. As far as the perfect match of the refrigerant temperature glide and the increasing water temperature in the gas cooler is concerned, transcritical CO_2 refrigeration systems have incomparable advantage over other heating methods (Riffat *et al.*, 1996). Air-source transcritical CO_2 heat pump water heater (ATHW) can supply hot water from 60℃ to 90℃ at high efficiency for commercial and residential applications (Yokoyama *et al.*,2007). In theoretical simulation and practical operating, the optimal heat rejection pressure of transcritical CO_2 heat pump system is the most significant factor to be considered to improve the overall system performance.

Many scholars have conducted investigations on the optimal heat rejection pressure of transcritical CO_2 heat pump system and made great achievements. Inokmy (1923) presented a graphical method to find the optimum pressure for trans-critical CO_2 refrigeration cycle. Kauf (1999) conducted a simulation model to find the optimal heat rejection pressure for the maximum COP and get the correlation of optimum high pressure and gas cooler outlet temperature or ambient temperature. Liao *et al.* (2000) developed a correlation of the optimal heat rejection pressure in terms of appropriate parameters, such as gas cooler outlet temperature, evaporation temperature and isentropic efficiency of the compressor for specific conditions. Sarkar *et al.* (2004) conducted an investigation of transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications. Finally, Expressions for maximum COP and optimum discharge pressure with relation of cycle parameter have been developed (Sarkar *et al.*, 2006). Chen (2005) performed an analysis on the relationship between the optimum high pressure and other systematic parameters. Cecchinato *et al.* (2010) critically discussed the optimal pressure expressions of several authors, and finally concluded that an approximated correlation should be critically evaluated before it applied to practical. Qi *et al.* (2013) presented an experimental investigation of the optimal heat rejection pressure for a transcritical CO_2 heat pump water heater and obtained a simple correlation of the optimal heat rejection pressure in terms of gas-cooler outlet refrigeration temperature. Zhang *et al.* (2013) carried out the simulation and experimental investigations on the relationships between optimum heat rejection pressures and related operating parameters for a transcritical system using R744/R290 mixture as a refrigerant. Finally, a correlation of the optimal heat rejection pressure with respect to the mass fraction, the outlet refrigerant temperature of the gas cooler, the evaporation temperature is obtained under the specific conditions.

There are many specific experimental and theoretical developments on transcritical CO_2 heat pump system used to supply hot water with high temperature. Jørn Stene (2005) carried out theoretical and experimental studies on air-source transcritical CO_2 heat pump system combined space heating and hot water heating. Yokoyama *et al.* (2010) numerically analyzed the performance of an air-to-water CO_2 heat pump water heating system and the influence of a daily change in a standardized hot water demand on the system performance. Yamaguchi *et al.* (2011) developed a simulation model for a CO_2 heat pump water heater and validated it with experimental results. Wang *et al.* (2013) analyzed the main affecting factors on a transcritical CO_2 heat pump water heater at a fixed water inlet temperature, and conducted an experimental research on a prototype in different working conditions. White *et al.* (2002) constructed a prototype transcritical CO_2 heat pump for heating water to temperatures greater than 65℃. When the evaporation temperature and hot water temperature were 0.3℃ and 77.5℃, the heating coefficient of performance of the prototype was 3.4.

Although a lot of optimal discharge pressure correlations for transcritical CO_2 heat pump have been proposed and many specific experimental and theoretical developments have been achieved in the past few years. The optimal pressure control method for transcritical CO_2 heat pump is still a hard question in practical application. Silvia Minetto (2011) described the development of a CO_2 air/water heat pump for the production of tap hot water in a residential building and developed a new control method for the upper cycle pressure to maximize the COP of tested heat pump system. W. Zhang and C. Zhang (2011) proposed a novel correlation-free on-line optimal control method, which use the on-line correction formula to track the optimal pressure set point for CO_2 transcritical refrigeration systems. Cecchinato *et al.* (2012) developed a real-time model-based optimization algorithm for the optimal or quasi-optimal pressure determination as a more efficient and robust solution than literature approximated ones. The proposed algorithm was dynamically tested by simulation, considering the performance of a supply water temperature controlled carbon dioxide heat pump. Ciro Aprea and Angelo Maiorino (2009) developed an implementable procedure on a cheaper electronic controller to drive an electronic back pressure valve with a simple optimal pressure model based on Liao *et al.*'s correlation.

In this paper, a revised model for optimal discharge pressure was developed based on experimental results and validated with an air-source transcritical CO_2 heat pump water heater in practical application. The field evolutions of optimal discharge pressure control effect were conducted to evaluate the system performance at three diffident operating conditions. With the reasonable agreement observed between the field test results and the model prediction. It is reasonable and effective to model the optimal discharge pressure as the function of the ambient temperature and the water outlet temperature. The overall ATHP system performance based on the revised model and PLC control strategy can meet the requirements in practical applications.

2. THE OPTIMAL DISCHARGE PRESSURE

2.1 Problem Definition

The optimal discharge pressure of a transcritical CO_2 refrigeration system is defined as: under certain operating conditions (the fixed gas cooler outlet temperature, evaporation temperature and other system parameters), the increase or decrease of compressor outlet pressure would cause COP reduction of the system, then the right compressor outlet pressure is called optimal discharge pressure on this conditions.

Hence there exists an optimal discharge pressure where the system reaches the best COP and the knowledge of the optimal operating conditions corresponding to the maximum COP is a very important factor in the design of a transcritical CO_2 refrigeration system. The gas cooler outlet temperature is dependent on external fluid inlet temperature; hence, at any discharge pressure, gas cooler outlet temperature will be fixed for a certain fluid inlet condition. Under certain environment temperature condition, the evaporation temperature is almost stable because of the constant heat transfer temperature difference. The existence of an optimal discharge pressure for fixed operating conditions can be supported by the following argument. For transcritical CO_2 cycle 1–2–3–4–5–6–1 (Figure 1), COP for the heating mode is given by:

$$\mathrm{COP}_{heating} = \frac{h_2 - h_3}{h_2 - h_1} \tag{1}$$

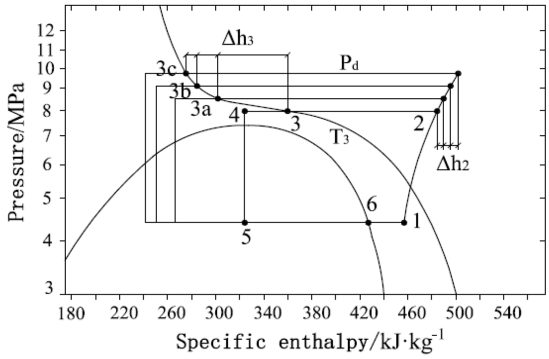

Figure 1: P-h diagram of transcritical CO_2 cycle for various high pressures

With increase of compressor discharge pressure for a constant cooler outlet temperature of T_3; the heating coefficient of performance expression gets modified as:

$$\mathrm{COP'}_{heating} = \frac{(h_2 - h_3) + \Delta h_2 + \Delta h_3}{(h_2 - h_1) + \Delta h_2} \tag{2}$$

Due to the unique behavioral pattern of carbon dioxide properties around the critical point and beyond, the slope of the isotherms is quite modest for a specific pressure range; at other pressures above and below this range, the isotherms are quite steep. As high pressure of the cycle increases, the state point of gas cooler outlet changes correspondingly. Just as indicated by 3a, the quantity Δh_3 is large compared to Δh_2; as is evident from Figure 1, and this causes an increase in the modified COP value as can be observed from Eq. (2). Along with the high pressure rise, the state point of gas cooler outlet can reach to 3b, and then the increasement of Δh_3 gets much smaller compared to 3a. At a particular pressure, the COP attains a maximum value. With further increase in pressure, just as 3c, Δh_3 does not produce the required gain over Δh_2 and thus the COP begins to fall. The pressure range where the isotherms are fairly flat and where this beneficial gain in COP occurs varies considerably with gas cooler outlet temperature. For a CO_2 refrigeration system, the gas cooler outlet temperature is dependent on external fluid inlet temperature (Kauf, 1999; Chen and Gu, 2005). Hence the fluid inlet temperature plays an influential role in determining the optimum operating conditions for the cycle.

2.2 Literature Correlations

Inokmy (1923) presented a graphical method to find the optimum pressure for trans-critical CO_2 refrigeration cycle used in automobile A/C systems. However, modern refrigeration systems have to fulfil many requirements, which include a rapidly growing number of applications. Professor Lorentzen (1994) in Norwegian University of Science and Technology (NTNU) discussed the trans-critical CO_2 refrigeration systems in motor car air conditioning, distinct heating and commercial refrigeration combined with hot tap water. Finally it is concluded that the optimal discharge pressure was mainly determined by the CO_2 gas cooler outlet temperature. Kauf (1999) found the graphical method is too time-consuming to determine the optimum high pressure if several operating conditions are investigated. The correlation of optimum high pressure and gas cooler outlet temperature or ambient temperature was introduced by simplifying the simulation of trans-critical CO_2 refrigeration cycle. Based on the cycle simulations, Liao *et al.* (2000) obtained the correlation of optimal heat rejection pressure in terms of gas cooler outlet temperature, evaporation temperature and isentropic efficiency of the compressor for specific conditions. Under a certain range of conditions, the isentropic efficiency is a constant or independent of the heat rejection pressure. It can be concluded that the optimal heat rejection pressure is a function of the evaporation temperature and the outlet temperature of gas cooler. The effectiveness of the internal heat exchanger was first considered to be an influencing factor to system performance by Sarkar *et al.* (2004). Finally it was concluded the effects of evaporator temperature and gas cooler outlet temperature were more predominant compared to internal heat exchanger effectiveness in determining the optimal high pressure correlations. Sarkar *et al.* (2006) introduced the compressor speed and inlet temperature of the fluid to be heated in gas cooler to invest the relationship of the optimal COP and system parameters. The effects of heat exchanger area ratio of gas cooler and evaporator on system performance were presented as well. The correlation of optimal high pressure related to fluid inlet temperature is obtained based on the fixed water outlet temperature, specified compressor displacement volume and heat exchanger area ratio. The mathematical correlation between the optimum high pressure and environmental temperature or gas cooler outlet temperature was established by Chen and Gu (2005). Qi *et al.* (2013) obtained a simple correlation of the optimal heat rejection pressure in terms of gas-cooler outlet refrigeration temperature for a transcritical CO_2 heat pump water heater. All the above expressions are based on several assumptions that differ one from another and also vary with the experimental behavior of the facility.

Table 1: Comparison of literature correlations

	correlations	conditions
Kauf	$P_{opt,Karf} = 2.6t_{air} \approx 2.6t_{gc,out} + 7.54$	$30℃ \le t_{air} \le 50℃$
Liao et al.	$P_{opt,Liao} = (2.778 - 0.0157t_e)t_{gc,out} + 0.381t_e - 7.54$	$-10℃ \le t_e \le 20℃$; $30℃ \le t_{gc,out} \le 60℃$
Sarkar et al. 2004	$P_{opt,Sarkar} = 4.9 + 2.256t_{gc,out} - 0.17t_e + 0.002t_{gc.out}^2$	$30℃ \le t_{air} \le 50℃$; $-10℃ \le t_e \le 10℃$
Sarkar et al. 2006	$P_{opt,Sarkar} = 85.45 + 0.774t_{w,in}$	$20℃ \le t_{w,in} \le 40℃$
Chen and Gu	$p_{opt,Chen} = 2.68t_{air} + 0.975 = 2.68t_{gc.out} - 6.797$	$30℃ \le t_{air} \le 50℃$
Qi et al.	$p_{opt,Qi} = 132.3 - 8.4t_{gc.out} + 0.3t_{gc.out}^2 - 2.68t_{gc.out}^3$	$20℃ \le t_{gc.out} \le 45℃$

where： p_{opt} was the optimal high pressure, t_{air} and $t_{gc,out}$ were the ambient temperature and gas cooler out temperature, respectively. T_e and $t_{w,in}$ were the evaporating temperature and water inlet temperature, respectively.

2.3 Optimal Discharge Pressure Correlation

Based on the experimental plant, CO_2 gas cooler outlet temperatures and evaporator temperatures are variables with change of discharge pressures (Wang *et al.*, 2013). Meanwhile, variation trends of CO_2 gas cooler outlet temperatures and evaporating temperatures are different with variation of discharge pressure at different ambient temperatures and water outlet temperatures. That means, the correlations presented in literatures for the optimal discharge pressure as the function of gas cooler outlet temperatures or fluid inlet temperature and evaporating temperatures is not suitable for control system, especially at the process of seeking the optimal status in a real plant. Essentially, as steady-state simulation of literatures, the correlations for the optimal discharge pressures only can be used to predict the optimal discharge pressures at certain gas cooler outlet temperature and evaporating temperature.

Therefore, based on more experimental results from the production unit, we obtained the revised correlations for the optimal discharge pressure as the function of ambient temperatures and water outlet temperatures, as shown below. The revised correlations are suitable for ambient temperatures from -15℃ to 35℃ and water outlet temperatures from55℃ to 95℃. The water inlet temperature is between10℃ to 15℃ for all the conditions.

$$P_{opt} = 1.097995 + 0.106442t_{w,out} + 0.101404t_{air} - 0.001216t_{air}^2 \qquad 5℃ \le t_{air} \le 35℃ \tag{3a}$$

$$P_{opt} = 2.468391 + 0.122379t_{w,out} - 0.000407{t_{w,out}}^2 + 0.016207t_{air} \qquad -15℃ \le t_{air} < 5℃ \tag{3b}$$

Where temperature is with the unit of ℃ while pressure is with the unit of MPa. Compared with the correlations given by Shouguo Wang (2013), the revised correlations are more reliable for high water outlet temperature conditions.

For application of the revised correlations for the optimal discharge pressure in an production units, firstly, according to ambient temperatures and required water outlet temperatures, the optimal discharge pressure can be calculated with the correlation, and then based on real-time discharge pressures measured by pressure transducers, control system can keep discharge pressure at the predicted optimal value by regulating the opening of expansion valve.

3. FIELD TEST

3.1 Application Introduction

To verify the control precision of the system optimal discharge pressure and evaluate the system performance of optimal pressure control strategy, an air-source transcritical CO_2 heat pump water heater of practical production was tested to get the system performance and optimal pressure control effect. The evaluation test was established in National Quality Supervision and Inspection Center of Compressor and Refrigerator Products, China from May 1st to May 3rd of 2013. ATHW was designed to replace the traditional oil-fired or coal-fired boilers and electric heating water heater in residential and commercial applications. With additional electric heater, an ATHW can supply boiled water around the whole year.

3.2 Test Facilities and Method

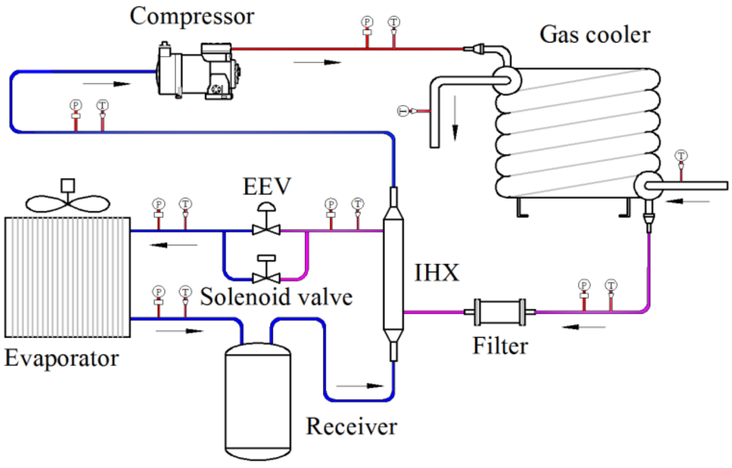

Figure 2: The schematic of a transcritical CO_2 heat pump water heater

The tested system is fully instrumented to evaluate its performance as a whole, and that of its individual components. The schematic of tested transcritical CO_2 heat pump system is shown in Figure 2. The temperatures are measured with T-type thermocouple temperature sensors with an accuracy of ±0.5℃. The pressures in the system are measured with pressure transducers ± (0.2% of full scale, 16MPa accuracy). The mass flow rate of water is measured by an Electromagnetic mass flow meter ± (0.5% of full scale accuracy). The electrical power input is monitored using a digital power meter QT1600 with an accuracy of ± (0.1% of range + 0.05% of reading). All the pressure, temperature, flow rate and power readings are continuously monitored by a calibrated Agilent HP34970 data acquisition system. The pressure and temperature are also collected by Siemens SIMATIC S7-200 Programmable Logic Controller (PLC) to predict the optimal discharge pressure.

Table 2: Characteristics of main components of the system

Components	Characteristics
CO_2 compressor	Semi-hermetic piston, Swept volume: 10.7 $m^3 \cdot h^{-1}$, Input power: 18 kW, Rotational speed: 1450 $r \cdot min^{-1}$
Gas cooler	Tube-in-tube, Counter flow, Stainless steel outer tube(water): Φ64 mm, Copper inner tube(CO_2): Φ7.94mm, Smooth tubes
IHX	Tube-in-tube heat exchanger, Heat transfer area: 0.16 m^2
Evaporator	Fin and tube heat exchanger, Heat transfer area 75 m^2
Fan	Axial flow, Volume flow: 16000 $m^3 \cdot h^{-1}$
EEV	JKV-24D, Electrically operated step motor valve
Receiver	Inner volume 0.125 m^3
Solenoid valve	HPV-825DS, CV: 0.038

The system is mainly comprised of a semi-hermetic reciprocating compressor, a finned-tube evaporator, a receiver, a filter, electronic expansion valves, a tube-in-tube gas cooler and defrosting solenoid value. The detailed characteristics of main components are shown in Table 2. The field test was conducted for three operating conditions: nominal condition, high temperature of water outlet (HTWO) and low ambient temperature (LAT) condition. The detail information for three operating conditions is presented in Table 3.

Table 3: Field test conditions

items	Ambient temperature Dry(Wet or RH) bulb	Water inlet temperature	Water outlet temperature
Nominal condition	25(23) ℃	15℃	80℃
HTWO	20(15) ℃	15℃	90℃
LAT	-15(60%) ℃	12℃	60℃

This test system could either be controlled automatically or manually, which could shorten the regulating time. The tested transcritical CO_2 heat pump water heater is shown in Figure 3.

Figure 3: Picture of the transcritical CO_2 heat pump water heater

3.3 Control Strategy

The control program consists of five subprograms: the startup subprogram, the stopping subprogram, the troubleshooting subprogram, the defrosting subprogram and the optimal pressure control subprogram. As the core part of control system, the optimal pressure control plays a significant role in determining the heating capacity and COP under certain operating condition. The compressor discharge pressure is achieved by adjusting the opening degree of the electronic expansion valve. According to the characteristics of PLC output, the idea of sequential control is adopted for the whole control process. Therefore, the control variables are processed at once after the end of the single scanning cycle.

Figure 4 shows the control strategy of system high pressure in optimal pressure control subprogram. As can be seen that the predicted optimal pressure is calculated after the system parameters are acquired. When the difference between the predicted value and the test high pressure is less than 0.01MPa, the opening degree of the EEV is no longer adjusted; otherwise, the opening degree of EEV will be regulated in the next cycle. The direction of EEV adjustment is decided by the minus of predicted value and test high pressure. If the result is positive, the opening degree will decrease. If the result is negative, the opening degree will increase. The time of scanning is much faster than the time of system parameters changing, which is the most important feature of the optimal pressure control strategy. All of this can reach the requirement of control precision.

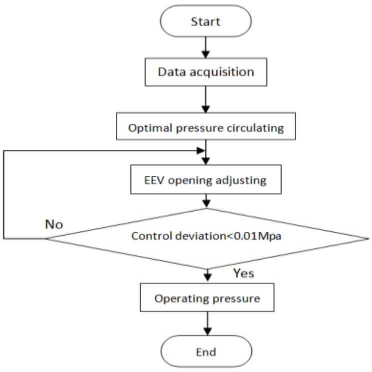

Figure 4: The optimal pressure control strategy

4. RESULTS AND DISCUSSION

Based on test facilities and control strategy, field evaluation was conducted to investigate the field test high pressure and system performance. The field test was conducted for three operating conditions: nominal condition, high temperature of water outlet and low ambient temperature condition. In the following sections, tested results are discussed.

4.1 Nominal Condition

It can be seen from Figure 5 that the water inlet and outlet temperature as well as the ambient temperature were recorded under nominal condition. At the beginning of the test, the water outlet temperature was adjusted by regulating the water flow rate. The stable test process started at 300s, and then all the temperatures were almost constant in the whole test. The maximum deviation of the water outlet temperature is 2.4%.

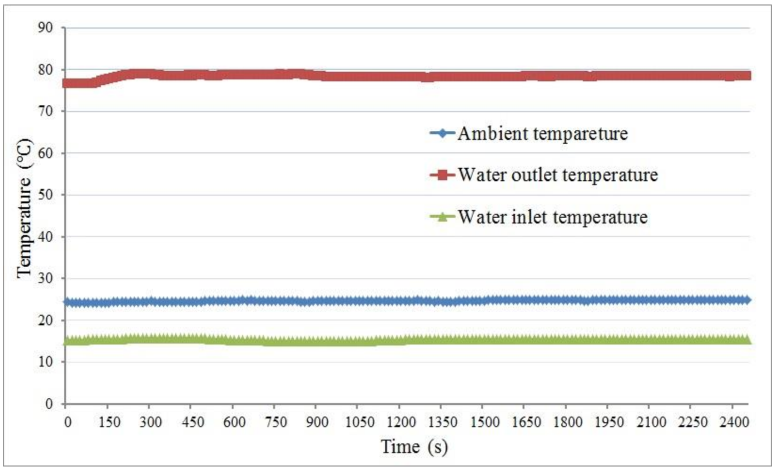

Figure 5: Test temperatures for nominal condition

Figure 6 shows the test pressure and predicted pressure in the nominal test condition, respectively. Based on the temperature recorded, the predicted pressure can be got by correlation (3a). Just as the variation of water outlet temperature at the beginning, the predicted pressure increased gradually to reach the requirement for standard hot water supply. To catch up with the predicted value, the test pressure was adjusted by changing the openings of EEV. When the difference was obvious, the change of openings was dramatic. The reverse is also true. So the test pressure fluctuated several times before it reached to target. In the whole stable test, the deviation of test pressure was 1.8%, which meant the control precision was accurate for nominal test condition.

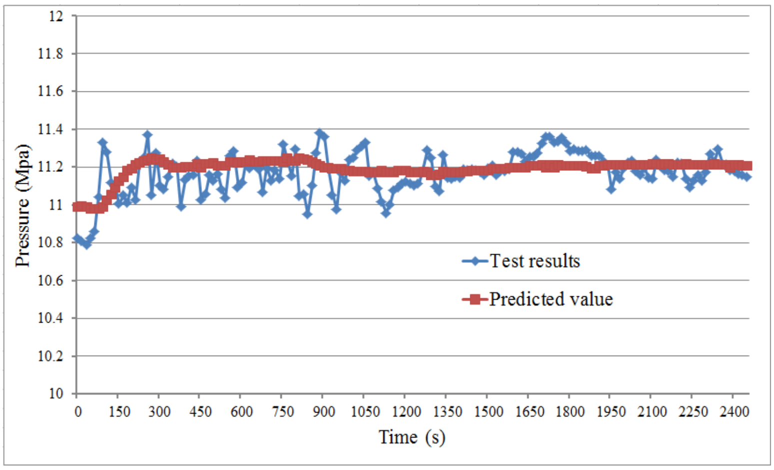

Figure 6: Test and predicted pressures for nominal condition

4.2 HTWO Condition

Figure 7 presents the water inlet temperature, water outlet temperature and ambient temperature for HTWO condition. Just as the nominal condition, the water outlet temperature raised correspondingly with the regulation of water flow rate at the beginning. After that, the water outlet temperature kept stable for the whole test process from 300s to 2100s. For every test condition, the whole stable test period was 1800s. The maximum deviation of the water outlet temperature is 2.6% for HTWO condition.

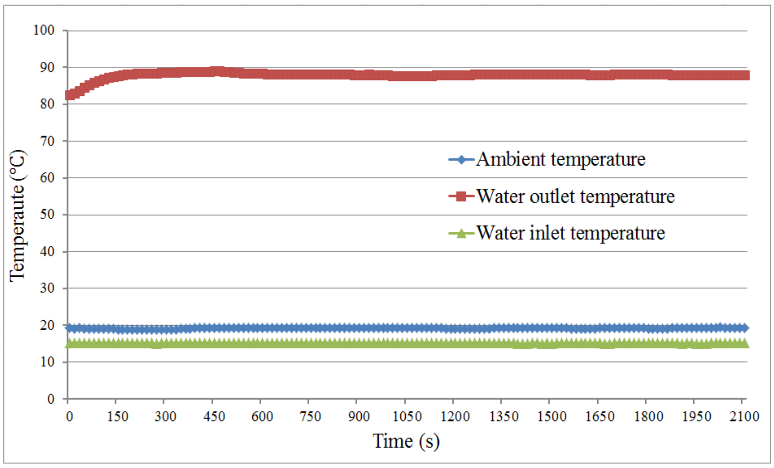

Figure 7: Test temperatures for HTWO condition

The test pressure and predicted pressure in the HTWO test condition is represented in Figure 8. With the constant of ambient temperature, the predicted pressure was just the single function of water outlet temperature. So before the test pressure reached a small neighborhood of predicted one, the openings of EEV changed rapidly to adjusting the actual test pressure. In the second half of the test, the predicted pressure and test pressure matched very well. For the whole 1800s stable test, the average test pressure was 11.64Mpa. To further evaluate the control procedure, the deviation of test pressure and predicted pressure was calculated for HTWO condition. The maximum value was 3.2% for the stable test process.

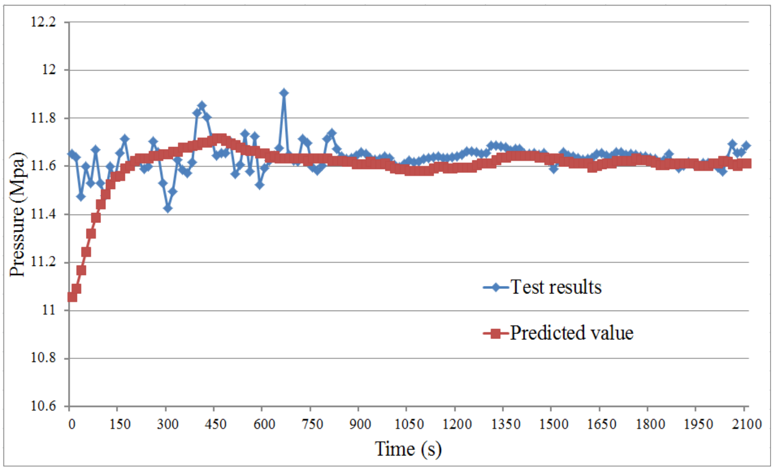

Figure 8: Test and predicted pressures for HTWO Condition

4.3 LAT Condition

As shown in Figure 9, the water inlet temperature, water outlet temperature and the ambient temperature were also obtained for LAT condition. In the first 300s, the water outlet temperature was much lower than 60℃ because of over required water flow rate. While the water inlet temperature and ambient temperature had already meat the test requirement. By reducing the water flow rate, the water outlet temperature increased significantly. For the whole test of neat 1800s, the water outlet temperature was no big difference with 60℃. The maximum deviation of the water outlet temperature is 1.2% for LAT condition.

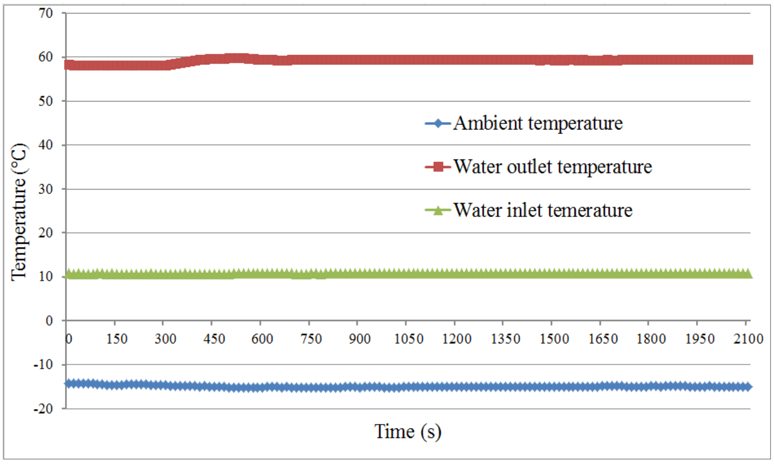

Figure 9: Test temperatures for LAT condition

For LAT condition, the test pressure and predicted pressure were presented in Figure10. Based on the temperature recorded in Figure 9, the predicted pressure can be got by correlation (3b). More difference from the previous test conditions, the test pressure jumped erratically in the first 300s and fluctuated periodically in the rest time. This was mainly because the openings were over the critical point of EEV. When the opening was smaller than the critical point, the test pressure increased sharply. When the opening was bigger than the critical point, the test pressure decreased sharply. For the stable teat process from 300s to 2100s, the periodical variation of test pressure was the result of optimal high pressure control strategy. So the test pressure fluctuated regularly over the predicted pressure in the stable test process. For low temperature test condition, the average test pressure was 8.06Mpa, which was closer to the predicted pressure. Correspondingly, the maximum deviation of the test pressure was 1.8% for control accuracy.

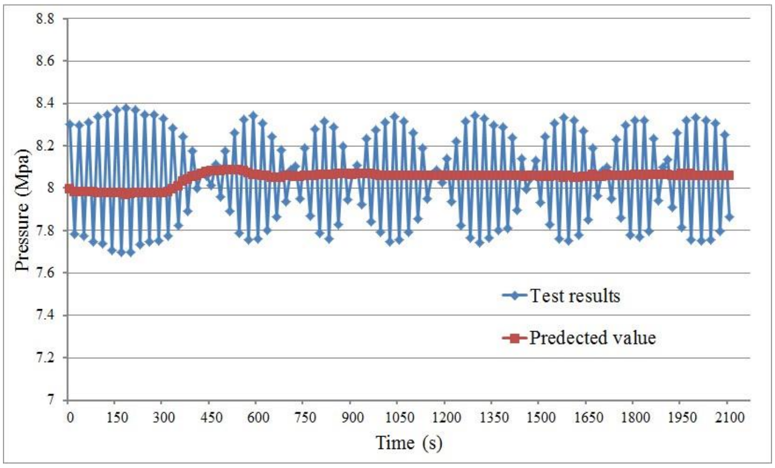

Figure 10: Test and predicted pressures for LAT Condition

4.4 Test Summary

The actual compressor discharge pressure control strategy was applied to air-source transcritical CO_2 heat pump water heater for three different test conditions. Based on the PLC control strategy and data collection, the system operating parameters were described above. To better evaluate the performance of HTHP, a test summary at different test conditions was made and the results were shown in Table 4. All those system performance: the system heating capacity, power consumption and COP were also obtained by the HP34970 data acquisition system.

Table 4: Test summary

items	Heating capacity kW	Power consumption kW	COP
Nominal condition	69.065	18.384	3.76
HTWO	59.664	18.585	3.21
LAT	27.877	12.734	2.19

5. CONCLUSION

The optimal discharge pressure correlations for transcritical CO_2 heat pump under specific conditions were discussed. The optimal discharge pressure as the function of gas cooler outlet temperatures or fluid inlet temperature and evaporating temperatures is not suitable for control system, especially at the process of seeking the optimal status in a real plant. In this paper, a revised optimal discharge pressure related to ambient temperature and water outlet temperature was introduced based on experimental prototype.

To further verify the correctness of the optimal discharge pressure correlation introduced, an air-source transcritical CO_2 heat pump water heater in practical application was tested to get the system performance and optimal pressure control effect. The field test results show the COP can achieve 3.76 in the nominal test conditions with 80℃ hot water supplied. Even when the hot water temperature is higher than 90℃, the COP keeps at 3.21 with 25℃ environment temperature. Under -15℃ low ambient air temperature condition, the COP was 2.19 with the hot-water supply temperature of 60℃. Based on the comparison between field test results and the model predictions, the maximum error of discharge pressure control was 3.2% in the high temperature water outlet conditions. A reasonable agreement between field test results and model predictions was achieved. That means the overall ATHP system performance based on the revised model and PLC control strategy can meet the requirements in practical applications.

NOMENCLATURE

h	Specific enthalpy	(kJ•kg-1)
Δh	Enthalpy difference	(kJ•kg-1)
P	Pressure	(MPa)
t	Temperature	(^{o}C)
Subscript		
opt	optimal	
air	air temperature	
gc	gas cooler	
e	evaporator temperature	
w	water	
in	inlet	
out	outlet	

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ACKNOWLEDGMENT

We are grateful to Jiangsu Baixue Electric Appliances Co., Ltd. and New Century Excellent Talents in University (NCET-12-0449) for funding this research.