EXPERIMENTAL EVALUATION OF THE THERMAL CHARACTERISTICS OF R417A USED IN AIR SOURCE HEAT PUMP WATER HEATERS

by

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In this paper, experiments were conducted to compare the thermodynamic performance of R417A used in air-source heat pump water heaters under two ambient temperature conditions of 20 ℃ and 7 ℃, including discharge pressure, input power and COP, and the experimental results were analyzed. The results showed that the operation was reliable and that R417A is very suitable for heat pump water heaters. This study provides a promising alternative refrigerant to freon.

Key words: *pyrology, R417A, heat pump water heater, refrigerants alternation*

Introduction

Air-source heat pump water heaters [1-4] have been rapidly developed and promoted in recent years due to their advantages such as good safety, energy saving and environmental protection. However, at present, most air-source heat pump water heaters still use HCFC R22 as the refrigerant, which causes serious damage to the ozone layer and increases the greenhouse effect, and according to the Montreal Protocol on the depletion of the ozone layer signed in Canada (1987), it will be banned in developing countries in the next decade. Research into alternative refrigerants to R22 is a key issue currently affecting the development of heat pump water heaters.

Scientists are currently researching the replacement of R22 with new refrigerants for heat pump water heaters [5]. The mixture of refrigerants R407c and R410a in heat pump water heaters has great potential as an alternative refrigerant to R22 [6]. However, the refrigeration capacity per unit volume and the discharge pressure of the almost azeotropic mixture refrigerant R410a are relatively high. As an alternative refrigerant, it requires equipment modification and cannot be used as a direct injection refrigerant replacement [7]. Although R407c has similar thermal performance to R22, it must be replaced with lubricating oil [8, 9].

The R417A is an environmentally friendly refrigerant developed by the French company Rodier. As an ideal alternative to R22, it also does not require the change of lubricating oil. At present, some researchers have conducted research on its application in heat pump water heaters, such as the high slip temperature of the near azeotropic mixture R417A may lead to a decrease in capacity and efficiency due to changes in its composition after iso-

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thermal steam leakage/refilling process. Cao *et al.* [10] conducted an experimental analysis of the mass composition of R417A in a heat pump water heater in the presence of 0%-50% isothermal steam leakage/refill. Chen *et al.* [11] experimentally compared the performance of R417A, R290, and R410A in the same heat pump/air conditioning water heater combination, including heating capacity, cooling capacity, power consumption, COP, discharge pressure, suction pressure, discharge temperature, and analyzed the performance of the heat pump/air conditioning water heater combination under standard air conditions and different inlet temperatures. Overall, there is relatively little research on the use of R417A for heat pump water heaters. The fractal thermodynamics proposed by Zhao, *et al.* [12] offers a new idea to study heat pump systems.

In this paper, comparative experiments were carried out to investigate the thermodynamic performance of R417A used in air source heat pump water heaters under two ambient temperature conditions of 20 °C and 7 °C, including heating capacity, cooling capacity, power consumption, COP, discharge pressure, suction pressure and discharge temperature, and the theoretical analysis is also carried out. We expect to support the promotion of R417A in the future.

Experimental set-up and working fluid characteristics

Description of the experimental set-up

The principle of air-source heat pump water heater is shown in fig. 1. It is designed for two modes of operation (*i.e.*, water heating mode and defrosting mode). The picture of the experimental equipment is shown in fig. 2 and the log*P-h* diagram of theoretical cycle is shown in fig. 3.

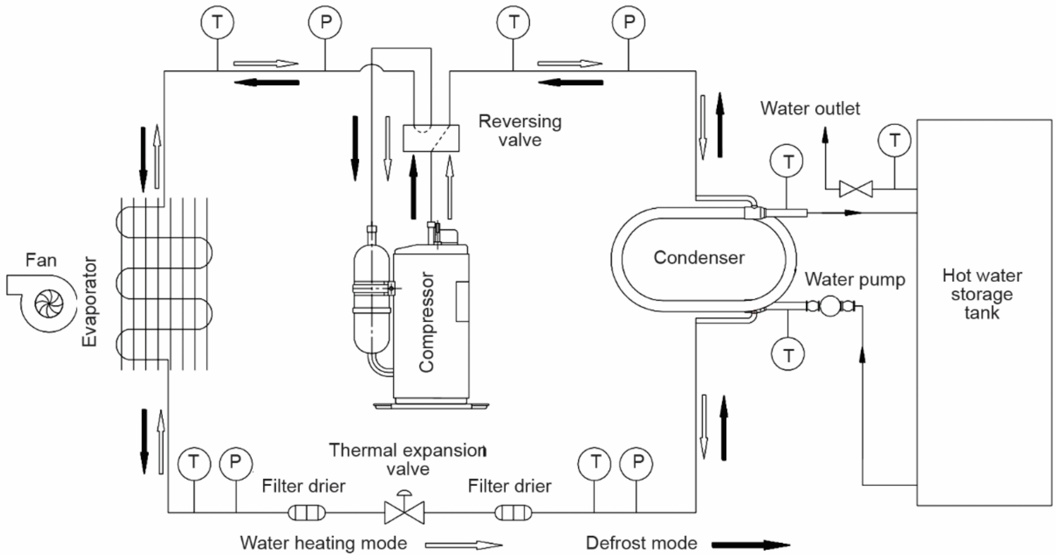

Figure 1. Principle of air source heat pump water heater

The test units consist of an air source heat pump water heater and a water tank. The specifications of the main components of the testing equipment are shown in tab. 1.

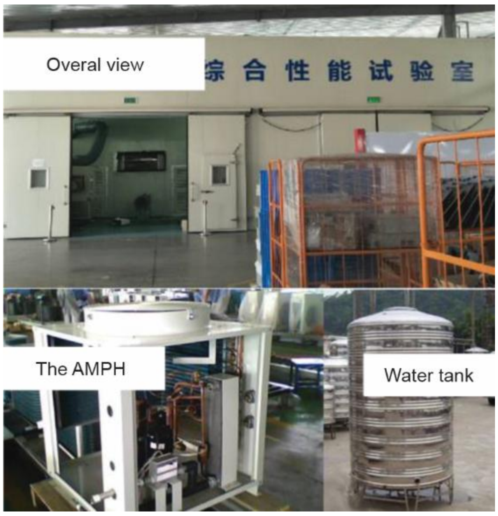

Figure 2. Experimental equipment

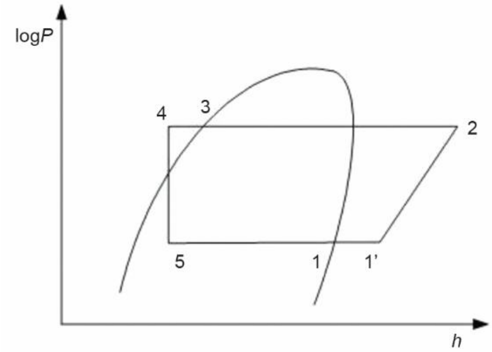

Figure 3. The log*P-h* diagram of theoretical cycle

Table 1. Specifications of the main components of the testing equipment

Air-source heat pump water heater	Quantity	Specification
Nominal water heating capacity		25 kW
Compressor	2	Copeland: ZR47KC-TFD
Working fluid		R417A
Air-source evaporator and condenser	1	Hydrophilic film corrugated aluminum fins and inner grooved copper tubes (axial fan rated input power, YDK139-120-4) Inner grooved copper tube (Tube pitch: 25.1 mm, pttra: 25.1 mm, OD: 9.52, δ: 0.5 mm) Hydrophilic film corrugated aluminum fin (Pitch of fins: 1.34 mm; δ: 0.13 mm)
Thermostatic expansion valve	2	BAE 7
Double-pipe heat exchanger	1	Tailored
Inner tube	3	External thread copper (ID: 16.3 mm; OD: 19 mm; Length: 4680 mm)
Outer tube	3	Seamless steel tube (ID: 25 mm; OD: 28 mm; Length: 4630 mm)
Insulation layer		Black waterproof rubber thermal board
Water tank	1	615 L
Liner		Stainless steel
Insulation layer		Polystyrene foamed plastic (Suitable temperature range: –50~125 °C; Thickness: 50 mm; conductivity: 0.034 W/mK)

OD: out diameter; ID: inner diameter.

– The ASHPWH with a rated cooling capacity of 25 kW mainly consists of the following primary components: two compressors (ZR47KC, Copeland, Taiwan, China), a tube-in-tube condenser insulated from the ambient air by using black waterproof rubber thermal board with a thickness of 10 mm, a finned tube evaporator with a total length of 121 m, and two expansion valves, while the amount of refrigerant was determined according to the actual needs.
– A pressurized water tank with a volume of 615 L and a maximum working pressure of 0.6 MPa was designed, and the tank was insulated from the ambient air by using 50 mm thick polystyrene foam plastic with a stainless steel plate.

Experimental cases

The experiment was conducted in an enthalpy laboratory (GB/T17758-2010) that simulates the conditions of the experimental environment. Two typical working conditions were tested: 7 °C/6 °C (dry bulb temperature 7 °C, wet bulb temperature 6 °C) and 20 °C/15 °C. The data were recorded continuously at approximately 5 °C intervals, and the experiment was stopped when the hot water temperature reached 55 °C from 15 °C and the hot water flow rate was 5.2 m^3/h. The experimental measurements are listed in tab. 2, and the data were then transferred to a computer. The data acquisition system consisted of a Keithley data acquisition instrument, temperature sensors, thermometers, and pressure gauges. The test instruments are described in tab. 2.

Two Rotronics (HygroFlex) termometers were used to measure dry and wet bulb air temperatures. The outdoor dry and wet bulb temperatures, similar to the condenser inlet air temperature, were quantified using two thermometers. Four Pt100 RTD (Omega) termometers were installed at four different positions in the water tank to measure water temperature variations, and the average of these four positions was taken as the water temperature. Two of these were installed at the inlet and outlet of the condenser to measure hot water temperature variations. Two Bourdon pressure gauges with different measuring ranges were used to measure the refrigerant level at the compressor inlet and outlet.

A FLUKE 39 power meter with a range of 0 kW to 10 kW (±2%) was used to record instantaneous and total power consumption.

All experimental data was monitored and collected throughout the process using a desktop computer.

Table 2. Summary of the measuring instruments

Parameters	Instrument	Model	Range	Accuracy
Air temperature and humidity	Thermometer	ROTRONIC HygroFlex	–40 ~ 85 °C	±0.1°C ±1% RH
Supply air flow rate	Thermal anemometer	E+E 70-VT62B5	0 ~ 5 m/s	±2% F.S
Water temperature	Pt100 RTD	Omega 1/10 DIN	–100 ~ 400 °C	±0.05 °C
Refrigerant pressure	Pressure gauge	Bourdon	–0.1 ~ 1.2 MPa –0.1 ~ 3.4 MPa	±0.3% F.S.
Power consumption	Power meter	FLUKE 39	0 ~ 10 kW	±2%

RH: relative humidity; F.S.: full scale

Performance Indicators

The water heating capacity, Q_{wh}, can be defined:

$$Q_{\mathrm{wh}} = \frac{\rho c_p V (t_{\mathrm{w}2} - t_{\mathrm{w}1})}{\Delta t} \tag{1}$$

where ρ is the water density, (= 1000 kg/m^3), c_p – the specific heat of hot water at constant pressure, (= 4.186 kJ/kgK), V – the volume of the water tank, (= 0.15 m^3), $t_{\mathrm{w}1}$, $t_{\mathrm{w}2}$ [°C] – the mean temperature of the water before and after heating in the tank at a unit time interval, and Δt [hours] – the time interval.

The COP for water heating, COP_{wh}, is calculated using:

$$COP_{\mathrm{wh}} = \frac{Q_{\mathrm{wh}}}{W} \tag{2}$$

where W [kW] is the power consumption.

Working fluid characteristics

The R417A is a near azeotropic blend refrigerant consisting of R125, R134a, and R600 in proportions of 47%, 50%, and 3%. Its performance is similar to that of R22. It has the advantage of being a non-zeotropic working fluid. As it evaporates or condenses at different temperatures, the composition of its gas and liquid phases changes, and its temperature changes continuously from a saturated liquid to a two-phase region until complete evaporation. As an alternative working fluid for fluorocarbon refrigerants, it can compensate for the shortcomings of using pure refrigerants.

When R417A is used in heat pump water heaters, it can reduce the compression ratio caused by the increase of water temperature and environmental temperature changes, enabling single-stage compression to achieve lower evaporation temperatures, increase the capacity of the refrigeration unit, realize non-isothermal refrigeration, reduce energy consumption, and improve the refrigeration coefficient. In addition, as a nearly azeotropic working fluid, the difference in gas, and liquid composition is relatively small. If the refrigeration system leaks, it will not cause excessive changes in the composition of the mixture. Therefore, using the original ratio for charging will not cause significant changes in machine performance, overcoming the drawbacks of non-zeotropic blends. The R417A will be an ideal alternative to R22 as a refrigerant for heat pump water heaters, and the theoretical comparison of characteristics between R417A and R22 is shown in the tab. 3.

Results and discussion

Figure 4 shows the variation of hot water temperature as a function of operating time for outdoor temperatures of 20 °C and 7 °C. It can be seen that the water temperature increases rapidly at the beginning and then the upward trend gradually slows down. This indicates that the increase in water temperature reduces the transfer efficiency of the heat exchanger as a function of the difference between the hot water and refrigerant temperatures.

It takes longer to heat the hot water to the desired temperature as the ambient temperature decreases. It takes approximately 73.109 minutes to heat 150 L of hot water from 15 °C to 55 °C at ambient temperatures of 20 °C and 7 °C, respectively.

Figure 5 shows the changes of suction pressure and discharge pressure with time. It can be seen that the suction pressure does not obviously change, while the discharge pressure

Table 3. Comparison of properties of R417A and R22

Refrigerant	R417A	R22
Dew point temperature, 1 atmosphere, [°C]	–36.7	25 kW
Boiling point temperature, 1 atmosphere, [°C]	–41.8	–40.8
Critical temperature [°C]	89.9	96.24
Critical pressure [MPa]	4242	4980.71
Liquid density, 25 °C, [kgm^{-3}]	1062.4	1194.68
Liquid specific heat, 25 °C, [kJkg^{-1}°C^{-1}]	1.44	1.24
Gas specific heat, 25 °C, 1 atmosphere, [kJkg^{-1}°C^{-1}]	1.06	0.685
Steam pressure, 25°C, [kPa]	985	1043.1
Liquid viscosity, 25 °C×10^{-4} Pa·s	1.62×10	1.59×10
Gas viscosity, 1 atmosphere ×10^{-4} Pa·s	1.21	1.30
ODP (R11= 1 .0)	0	0.05
GWP (CO_2 = 1.0)	1950	1700
Lubricating oil	MO, AB, POE	MO, AB, POE
Toxicity	NO	NO

gradually increases with the operating time due to the increase in hot water temperature, resulting in an increase in compression ratio. The increase in discharge pressure is due to the fact that the heat exchanger is designed for standard operating conditions. As the hot water temperature increases, its heat exchange surface gradually becomes insufficient, resulting in an increase in condensing temperature and pressure. The reason for the apparent variation in suction pressure can be explained by the fact that the increase in pressure ratio causes an increase in the mass flow of refrigerant entering the evaporator, resulting in a decrease in refrigerant superheat. To maintain a constant superheat of the refrigerant, the opening of the expansion valve is reduced, thus limiting the flow of refrigerant into the evaporator.

In addition, the decrease in gas transmission coefficient caused by the increase in pressure ratio results in a decrease in refrigerant recirculation flow.

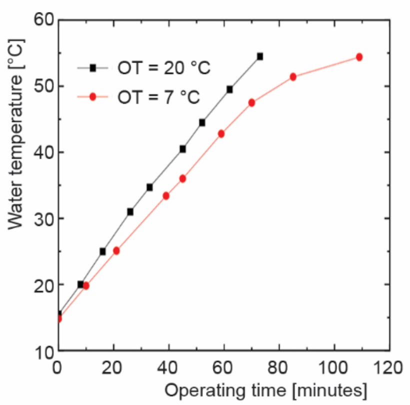

Figure 4. Variations in hot water temperature with operating time

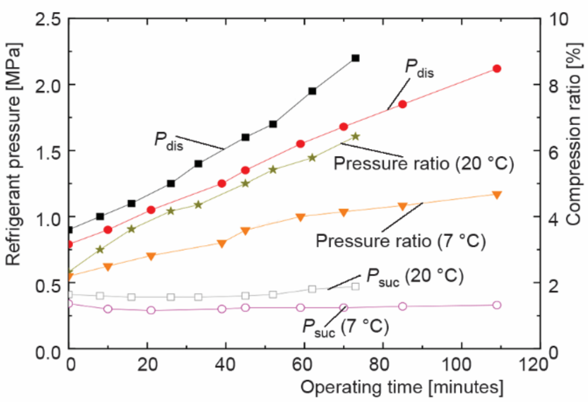

Figure 5. Variations in the discharge and suction pressure with operating time

Figure 6 illustrates the variation of hot water output with operating time. It can be observed that the refrigerant flow will decrease due to the increase in pressure ratio with hot water temperature, resulting in a decrease in the transient hot water output.

Furthermore, the heat transfer coefficient and total heat transfer of R417A will be slightly reduced due to the temperature slip of R417A as a ternary non-azeotropic mixture, particularly in relation to bubble detachment resistance under nucleate boiling conditions. As a consequence of the augmented compression work per unit mass of refrigerant, the instantaneous input capacity of the system is augmented. The instantaneous input capacity of the system is enhanced due to the increase in compression work per unit mass of refrigerant.

The heating water capacity is significantly reduced in the 7 °C operating condition in comparison to the 20 °C operating condition. In contrast, the input capacity is less affected by the ambient temperature. The reduction in heat output can be attributed to the diminished refrigerant circulation resulting from the elevated compression ratio. The diminished impact of ambient temperature on input power can be attributed to the concurrent reduction in refrigerant flow and the concomitant increase in input power per unit mass.

The calculations indicate that the average water heating power of the system is 23.51 kW under 20 °C operating conditions, with an average input power of 5.47 kW. In contrast, the average water heating power of the system is 15.75 kW under 7 °C operating conditions, with an average input power of 4.83 kW.

Figure 7 illustrates the variations in the COP with respect to operating time under 20 °C and 7 °C conditions. It can be observed that there is a gradual decline in the COP due to a reduction in water heating power and an increase in input power resulting from an elevated hot water temperature.

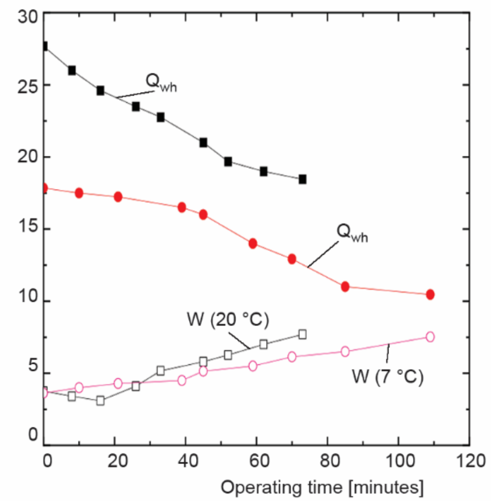

Figure 6. Variations in power with operating time

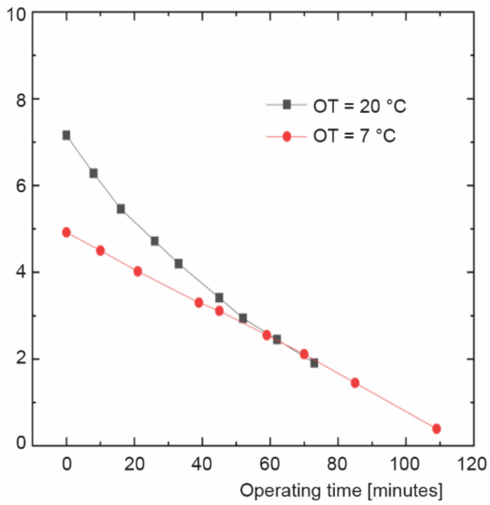

Figure 7. Variations in COP with operating time

Conclusions

- The experimental results indicate that the suction pressure of the system remains relatively constant, while the discharge pressure of the system gradually increases in a linear fashion with increasing hot water temperature. Both are equally susceptible to environmental influences.

- The heating power gradually decreases and the input power gradually increases as the hot water temperature increases with the operating time.
- The COP continuously decreases and the downward trend gradually slows down with the operating time as the hot water temperature increases. This R417A is suitable for applications where the water temperature is not high. Under experimental conditions, the COP was 4.30 and 3.26 under 20 °C and 7 °C operating conditions, respectively.

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