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Analysis of the mixture refrigerant R600/R245fa using in high temperature heat pump

Xiaohui Yu*, Chang Xu

School of Energy and Environment Engineering, Hebei University of Technology, Tianjin 300401, PR China

Abstract

High temperature heat pump (HTHP) is an effective equipment to recovery industrial waste heat and it has a wide application potential. The choice of refrigerant has important influence on the efficiency of HTHP. In this paper, the performance parameters of the mixture refrigerant R600/R245fa using in HTHP were tested. Compared with the R245fa, the mixture refrigerant R600/R245fa has higher heating capacity and lower COP. But the COP of the HTHP system also can maintain above 3.0 when the temperature of condenser outlet water is at 100 ℃ and 35 ℃ temperature lift. Analysis shows that R600/R245fa is suitable for HTHPs with higher condenser outlet water temperature and heating capacity demand. And its low GWP also has good environmental benefits.

Keywords: mixture refrigerant; High temperature heat pump; cycle performance

Nomenclature		**Greek symbols**	
COP	coefficient of performance	η	efficiency (%)
C_p	specific heat capacity (kJ/(kg.K))	Subscripts	
HTHP	high temperature heat pump	1,2,3,...,6	state point
GWP	global warming potential	a	available
ODP	ozone depletion potential	b	boil
h	enthalpy (kJ/kg)	con	condenser
m	mass flow rate (kg/s)	h	high temperature
T	temperature (°C)	i	inlet
s	entropy (kJ/(mol.K))	l	low temperature
P	power (kW)	o	outlet
Q	heating (kW)	r	refrigerant

1. Introduction

The problem of energy and environment has become a serious problem facing all countries in the world. The environmental pollution caused by energy consumption has become an important obstacle to global

* Corresponding author. Tel: +86 15102282362
E-mail address: 2018133@ hebut.edu.cn

economic growth and social progress. Industrial energy consumption accounts for nearly half of the world's total energy consumption [1]. Many industrial processes produce a large amount of waste heat and directly discharge to the environment, resulting in energy waste and environmental pollution. Therefore, waste heat utilization is an effective way to energy saving. On the one hand, the utilization of waste heat can reduce the thermal pollution. On the other hand, it can reduce the consumption of high-grade energy and achieve multiple purposes. HTHP is one of the reliable waste heat recovery devices. It can improve the quality of low-grade heat energy to higher quality by producing high temperature water or steam. Metallurgy, chemical industry, food, paper and other industries have a large demand for 100-130℃ hot water [2]. Therefore, HTHP has a good application prospect and has some new researches in recent years.

The research on refrigerant selection was also widely carried out, which lays a solid foundation for the selection of HTHP working medium. Guido et al. [3] conducted extensive screening on the working medium of steam compression high temperature heat pump, there were 27 kinds of preliminary selection under the environmental protection requirements and technical restrictions. Bobelin et al. [4] verified that a new mixed refrigerant ECO3 and it could effectively raise the outlet water temperature of HTHP to 125 ℃ . Chamoun et al. [5] designed and tested a new type of heat pump which raised the outlet water temperature to 130~140 ℃ when using R718 as refrigerant. Bamigbetan et al. [6] proposed that pasteurization process, drying, distillation and so on all need the working temperature of 100-125 ℃. After screening, R600 and R1233zd(E) have better performance and basically suitable for the mainstream compressor. In addition to the pure substance, the research of mixed refrigerant has gradually become another direction. Yu et al. [7] conducted a binary near-azeotropic mixture named BY- 4. The COP was 2.8-3.6 when the temperature of outlet water was 90-110 ℃ and 40 ℃ temperature rose. Guo et al. [8] analyzed a series zoetropic mixture, proposed that N-hexane/propene and R4310mee/R32 are superior alternative to CO2 heat pump.

R245fa has been theoretically calculated or experimentally verified in some literatures to show that it is a kind of HTHP refrigerant with good performance, safety and stability [9],[10]. R245fa is widely used because of its high COP, low compressor discharge temperature and discharge pressure. The obvious disadvantage of R245fa is that its volumetric heating capacity is lower compared with the conventional heat pump refrigerant [11]. R600 is a recognized refrigerant which is suitable for HTHP, but its application is greatly restricted due to its inflammable and explosive characteristics [12]. R1234ze(Z) and R1233zd(E) are considered as potential substitutes for R245fa in some studies from the perspective of pure refrigerant [13]. In this paper, from the perspective of mixture refrigerant, a binary mixture substance R600/R245fa is proposed to provide more refrigerant choices for users of HTHPs. Most studies on the refrigerants of HTHP adopt the method of theoretical calculation, while in this study, the mixture refrigerant R600/R245 is tested on the experimental platform. For comparison, pure refrigerant R245fa with the same injection mass is also tested at the corresponding working conditions. According to the research of Zhou et al. [14], the theoretical cycle calculation results show that the mixture refrigerant prepared on R245fa has a higher COP when the mass ratio of R245fa is 0.5-0.7. Therefore, the ratio of R600/R245fa is defines as 0.3/0.7 in this experiment. The experimental results show that the two refrigerants R600 and R245fa can complement each other. After the two constitute a mixture refrigerant, the COP of the HTHP system decreases and the heating capacity increases. Compared with pure refrigerant R245fa, the compressor discharge temperature and discharge pressure of mixture refrigerant increases, the compression ratio decreases obviously. The GWP of the mixture refrigerant decreases compared with the pure refrigerant R245fa due to the low GWP of R600. The inflammable and explosive properties of R600 are also weakened because of its low proportion in the composition. The experiment setup are presented in Section 2. The thermodynamic analysis methods are described in Section 3. The results and discussions are arranged for Section 4. Conclusions are summarized in the last section.

2. Experiment setup

A HTHP system was designed and built for testing the performance of the proposed mixture refrigerant R245fa/R600. The schematic diagram of the HTHP system is shown in Fig. 1. The main equipment of the system is evaporator, condenser, electronic expansion valve and scroll compressor. A vapor-liquid separator is installed between the evaporator and the compressor to prevent the refrigerant liquid from entering the compressor and causing liquid hammer. The electronic expansion valve and the condenser are equipped with dry filter and liquid storage tank. Some other accessories together with the main equipment constitute the whole HTHP system. The heat source of the system is an adjustable electric heating tank. The temperature in the heating water tank can be maintained constant through the control cabinet to satisfy the requirements of the evaporator's stable inlet temperature. The heat sink side of the system uses the fan-coil unit to discharge heat to outdoor air and maintain the heat balance of the system.

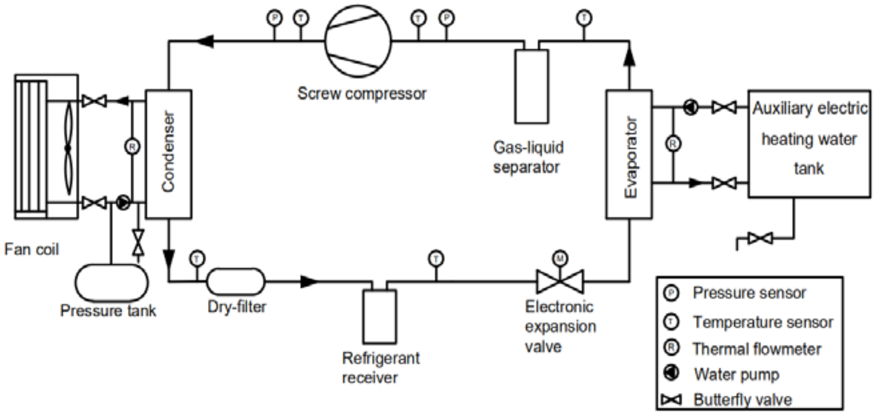


Fig. 1. Schematic diagram of the HTHP system

The detailed parameters of main equipment of HTHP are listed in Table 1. And the information of measuring instruments are listed in Table 2. Some parameters of the refrigerants are listed in Table 3.

Table 1. The main equipment parameters of the HTHP

Equipment	Type	Specification
Evaporator	EB12P3823	Heat transfer rate: 13.3 kW
Condenser	CB12P3823	Heat transfer rate: 14.7 kW
Electronic expansion valve	DPF(Q01)2.4	Diameter: 2.4 mm
Screw compressor	HRH049U4LP6	Rated power: 4.04 kW Nominal refrigeration capacity: 12110 W
Water pump	PUM200	Rated flow: 2.5 m^3/s; Pump head: 12 m; Power: 0.25 kW.
Fan coil	KFR-35GW/MHAB1	Specification: 1280×195×295 mm
Auxiliary electric heating water tank	SX-1000	Volume: 1 m^3

Table 2. The information of measuring instruments

Name	Type	Accuracy grade

Pressure transmitter	MIK- P50- 2 (low- pressure side) MIK- P100- 8 (high- pressure side)	± 0.5% FS included nonlinearity hysteresis and temperature drift
Digital power meter	RK9900N	± 0.5% FS
Turbine flow meter	LWY-25	± 1.0L
Pressure gauge	YTZ-150	± 1.5% FS
Temperature sensor	PT1000	A

Table 3. Some parameters of the refrigerants

Substance	Molar mass (g/mol)	Tb (°C)	Tc (°C)	Pc (MPa)	ODP	GWP (100yr)
R600	58.12	- 0.49	151.98	3.796	0	4
R245fa	134.05	15.14	154.01	3.651	0	858
R600/R245fa (3:7)	96.31	/	132.74	3.557	0	/

3. Analysis method

The circulating principle of high temperature heat pump is reverse Carnot cycle, as shown in Fig. 2. The blue and red lines represent the temperature changes of the circulating water in the evaporator and the condenser respectively. The whole refrigerant cycle includes four thermodynamic processes: endothermic process (5-1) , compression process (1-2) , condensation process (2-4) , throttling process (4-5) . In order to simplify the calculation, the following assumptions are made as follows.

(1) All components of the system are operated under steady- states and steady- flow conditions.

(2) The changes in kinetic energy and potential energy of the refrigerant are ignored.

(3) The temperature loss and pressure loss of the refrigerant during pipeline circulation are ignored.

(4) The evaporation and condensation processes are isobaric.

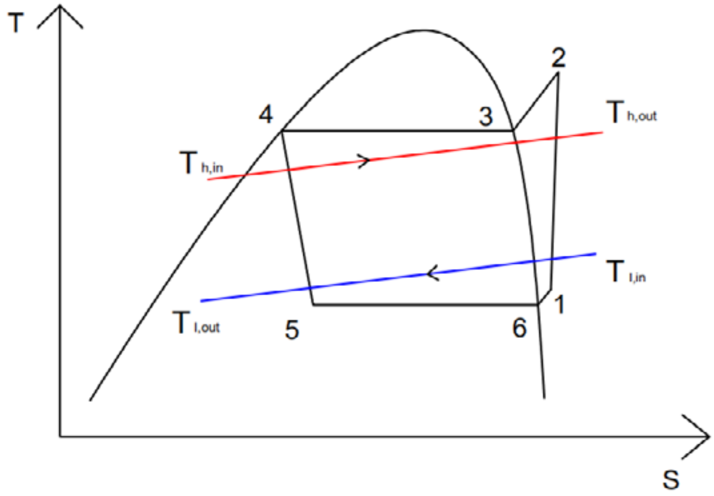


Fig. 2. T-s diagram of the HTHP

The energy equation of some components of the system, based on the first law of thermodynamics, can be calculated as follows.

(1) Condenser

The heating release of the refrigerant in the condenser can be calculated as follows：

$$Q_{con} = m_r(h_2 - h_4) = m_{hw}C_{p,hw}(T_{h,in} - T_{h,out}) \tag{1}$$

Where h2 is the enthalpy of the refrigerant at the condenser inlet, h4 is the enthalpy of the refrigerant at the condenser outlet, mr is the mass flow rate of the refrigerant, mhw is the mass flow rate of the high temperature water passing through the condenser, Th,in is the temperature of high temperature water at the condenser inlet, Th,out is the temperature of high temperature water at the condenser outlet, Cp,hw is the constant pressure specific heat capacity of the high temperature water.

(2) Scroll compressor

Scroll compressor is suitable for small units because of its high efficiency and strong stability. The efficiency of compressor can be calculated as follows:

$$\eta = \frac{P_a}{P_{in}} \tag{2}$$

Where η is the compression efficiency, Pin is the input power of the compressor which can be measured directly, Pa is the available power which can be expressed as:

$$P_a = m_r(h_2 - h_1) \tag{3}$$

Where h1 is the enthalpy of refrigerant at the evaporator outlet.

In conclusion, the COP of the system can be calculated as follows:

$$COP = \frac{Q_{con}}{P_{in}} \tag{4}$$

4. Results and discussions

The evaporator inlet water temperature of the experiment ranged from 45 °C to 65 °C and the temperature interval is 5 °C . The temperature lift between evaporator inlet water and condenser outlet water remains constant at 35 °C . When the operating condition is stable, the HTHP performance is tested and investigated under different operating conditions. For the convenience of the following description, R600/R245fa is denoted by R1 and R245fa is denoted by R2.

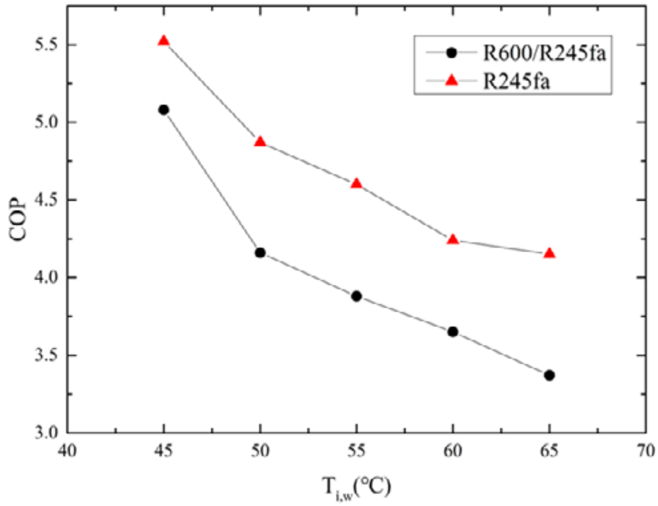
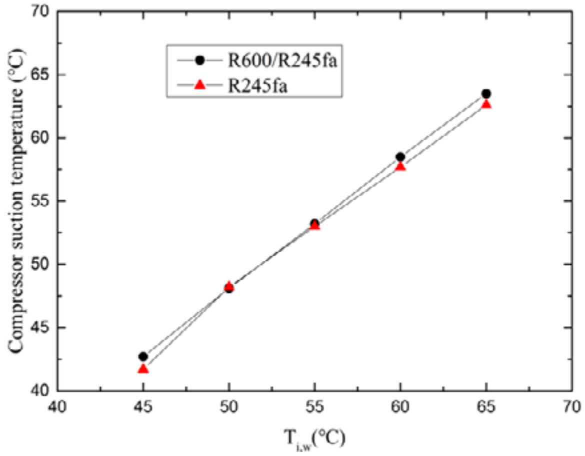
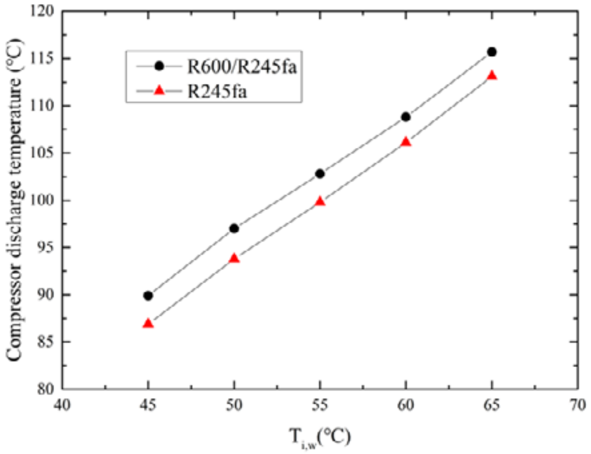


Fig. 3. The COP of HTHP varied with different inlet water temperature of the evaporator

As shown in Fig. 3, the COP of R1 and R2 decreases gradually with the increase of evaporator inlet water temperature. When the inlet water temperature increased from 45 °C to 65 °C , the COP of HTHP using R1 and R2 decreases by 1.71, 1.37 respectively. It means that the COP of the HTHP system decreases by 1.68 % and 1.24% on average with the evaporator inlet water temperature increased by 1 °C . Among them, the COP decreases faster in the temperature interval 45-50 °C , and the other temperature intervals decrease evenly. The average COP of using R1 decreases about 13.9% compared with R2. Considering the experimental error, the R1 also can maintain the average COP not less than 3.0 when the water temperature lift is 35 °C and the outlet water temperature at 100 °C.



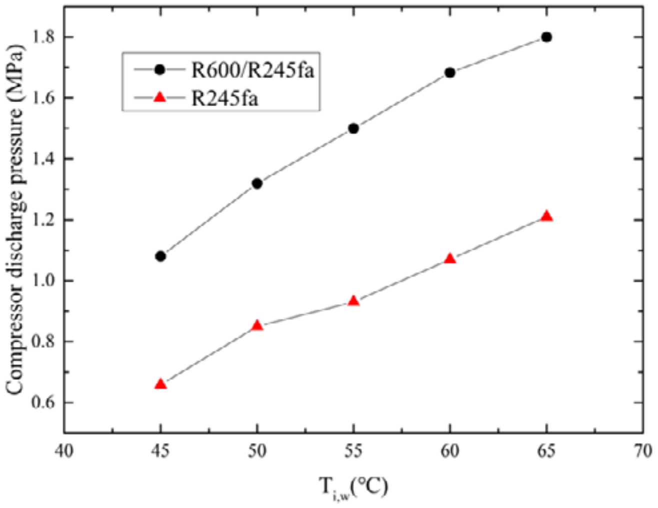
(a) The compressor suction temperature changed with different inlet water temperature of the evaporator



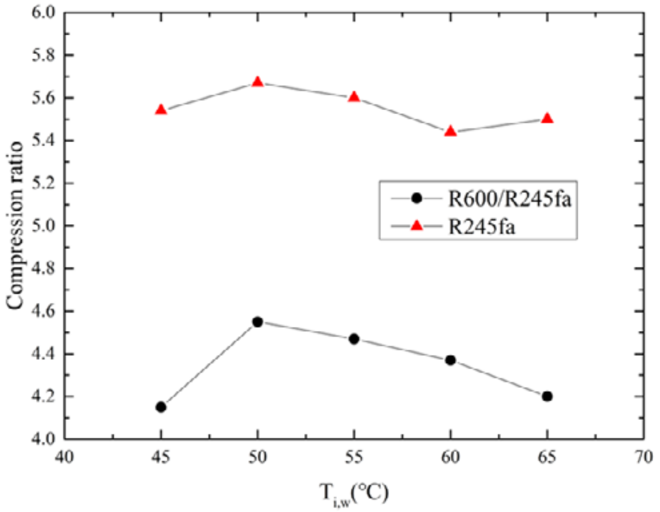
(b) The compressor discharge temperature changed with different inlet water temperature of the evaporator

Fig. 4. The compressor suction temperature and discharge temperature changed with different working conditions

As shown in Fig. 4 (a), the compressor suction temperature varies with different inlet water temperature of the evaporator. R1 has a slightly higher compressor suction temperature than R2 in this experiment. The compressor suction temperature is lower than the evaporator inlet water temperature, which may be due to a little amount of liquid refrigerant evaporates and absorbs heating at the inlet of the compressor. The Fig. 4 (b) shows the change of compressor discharge temperature under different evaporator inlet water temperatures. The compressor discharge temperature corresponding to R1 is significantly higher than that to R2, and the average temperature overshoot is 2.9 °C. The difference between the compressor suction temperature and the discharge temperature will rise with the increase of the evaporator inlet water temperature in this experiment.



(a)



(b)

Fig. 5. The variation of the discharge pressure of the compressor (a) and compression ratio (b) changed with different inlet water temperature of the evaporator

Fig. 5 shows the variation of the compressor discharge pressure and compression ratio under different working conditions. As shown in Fig. 5(a) , the compressor discharge pressure of R1 is 0.533MPa higher than R2 on average. The maximum of the difference between the two pressures occurs when Ti,w is 60 ℃ and the value is 0.613 MPa. Despite the higher compressor discharge pressure of the R1, it can been seen from Fig.

5(b) that the compression ratio is lower. The average compression ratio of R1 is 4.35 and the average compression ratio of R2 is 5.55. The different of the mean value is 1.2. It seems that R1 may be suitable for higher heating temperatures form the trend. In general, R1 is superior to R2 in term of compression ratio.

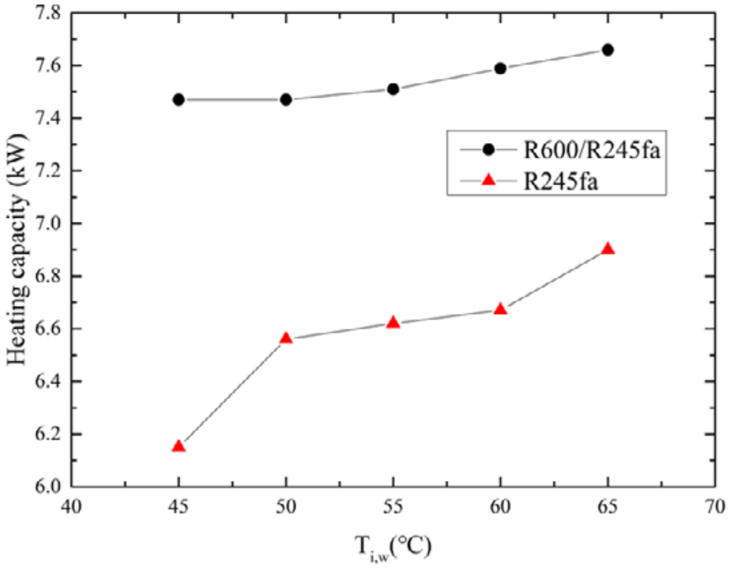


Fig. 6. The heating capacity of HTHP varied with different inlet water temperature of the evaporator

As shown in Fig. 6, the heating capacity of the HTHP using R1 as refrigerant is significantly higher than that of R2. The value of using R1 is on average 14.7% higher than that of R2. This suggests that R1 is better suited to higher heating capacity requirements of HTHPs than R2. The heating capacity of the HTHP is negatively correlated with the compression ratio. In this experiment, with the increase of the evaporator inlet water temperature, the compression ratio of the two refrigerant decreases gradually. It makes the compressor volume efficiency lift, resulting in the increase of the actual compressor gas transmitting volume. Meanwhile the volumetric heating capacity decreases with the rise of condensation temperature, but the product of volumetric heating capacity and refrigerant steam volume at the outlet of the compressor will still increase, that is, the heating capacity of the HTHP will continue to increase.

5. Conclusion

The operating performance of the HTHP system using R600/R245fa and R245fa as refrigerant are obtained by experimental test under different running conditions. We analyzed these data and reach the following conclusions:

1. From the point of the COP, the mixture refrigerant R600/R245fa is less than R245fa and the COP dropped by an average of 13.9%. But the mixed refrigerant R600/R245fa can keep the COP at least 3.0 when the temperature lift is 35 ℃ and outlet water temperature at 100 ℃. Therefore, this kind mixture refrigerant can be accepted for HTHPs.

2. The compressor discharge temperature and compressor discharge pressure of HTHP using R600/R245fa will be higher than that of R245fa. However, R600/R245fa has advantages in terms of compression ratio and heating capacity, in which the compression ratio decreases 1.2 on average and the heating capacity increases 14.7% on average.

3. Compared with the R245fa, the mixture refrigerant R600/R245fa is more suitable for HTHPs with higher heating capacity demand and higher temperature working conditions.

4. The mixture refrigerant R600/R245fa has lower GWP than the R245fa, which has certain advantages in environmental protection.

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