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Performance analysis of a cascade high temperature heat pump using R245fa and BY-3 as working fluid

Xuelian Ma[a], Yufeng Zhang[a], Lei Fang[b],

Xiaohui Yu[a], Xiaoqiong Li[a], Ying Sheng[a,*], Yan Zhang[c],

[a] School of Environmental Science and Engineering, Tianjin University, Tianjin, 300072, PR China

[b] International Centre for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark, Lyngby 2800, Denmark

[c] School of Architecture, Tianjin University, Tianjin, 300072, PR China.

ABSTRACT: A high temperature cascade heat pump (HTCHP) using a near-zeotropic mixture named BY-3 as the working fluid in the low-stage refrigerant cycle and R245fa as working fluid in the high-stage refrigerant cycle was proposed in this study. Several experiments were carried out to investigate the performance of the HTCHP at the evaporating temperature from 40℃ to 60℃ and the water outlet temperature on the condensing unit of the high-stage cycle can reach 142℃ with the coefficient of performance (COP) of 1.72. The results showed that BY-3 was feasible to be used in the low-stage cycle. A numerical model of the HTCHP was proposed and validated in this study to evaluate its performance. The comparison between the experimental results and the simulated results showed that the HTCHP system using BY-3 and R245fa can product hot water at 142℃ with good performance and the

* Corresponding author: Ying Sheng

E-mail: ysheng@tju.edu.cn

Address: No. 135 Yaguan Road, Jinnan District, Tianjin, China.

temperature lift of the HTCHP can reach 100℃.

Keywords: High-temperature Cascade Heat Pump; Modeling Analysis; Experimental Verification; Performance Evaluation

Nomenclature	
COP	Coefficient of performance
EEV	Electronic expansion valve
HX	Heat exchanger
Cp	Specific heat(kJ/kg·K)
m	Mass flow rate(kg/h)
P	Power consumption(kW)
h	Enthalpy(kJ/kg)
T	Temperature(℃)
Q	Capacity(kW)
W	Power input(kW)
HTCHP	High temperature cascade heat pump
P_c	critical pressure(bar)
P_{theo}	Theoretical power(kW)
P_{in}	Input power (kW)
GWP	Global Warming Potential
ODP	Ozone Depleting Potential
r	Latent heat(kW/kg)
Ø	EEV opening degree
Subscripts	
w	water
v	vapor
l	liquid
L	Low –stage refrigerant cycle
H	High-stage refrigerant cycle
cond	condenser
evap	evaporator
ref	refrigerant
t	total
out	outlet
in	inlet
Greek symbols	
ρ	density(kg/m^3)
	acentric factor
η_s	volumetric efficiency

η_m	friction efficiency
η_e	motor efficiency
1,2,3,4,5,6,7,8,9,10	state point

1. Introduction

Environmental concerns such as global warming, ozone depletion, global climate change and air population have become worldwide issues in the past several decades mainly due to anthropogenic activities such as vehicle emission, pollutions from power plants, steel mills, and many other industrial sectors[1][2]. The use of substantial fossil fuels contributes this the most, especially the fuel consumption on the industrial aspect. It was reported that the energy was accounted for more than 70% of China's total energy consumption in 2010. In the U.S., the industrial sector accounted for about one third of the total energy consumed and it was estimated that 20 to 50% of industrial energy input was lost as waste heat in different forms [3]. Consequently, new technologies that can improve energy efficiency and recover waste heat losses provide an attractive challenge for an emission-free and less-costly energy resource.

The cascade refrigeration system composed of two independent refrigerant cycles was first introduced in1930s to overcome the disadvantage such as high pressure ratio and low COP of single-stage refrigeration in low-temperature refrigeration systems in the range from -30℃ to -100℃[4][5]. Jung et al. drew the conclusion that the cascade heat pump was much more competitive than single-stage heat pump by studying a cascade multifunctional heat pump using R410A and R134a as working fluids experimentally [6].

Many researches have been done to investigate the performance of the cascade heat pump systems. Wang et al. performed series of experiments to analyze the impact of different operation parameters on the performance of cascade refrigeration and concluded that the NH_3/CO_2 cascade was competitive in low temperature applications [7]. Alberto et al. designed and built a cascade refrigeration system with NH_3 and CO_2 to study the system performance and the results accords with the trends obtained from the theoretical researches published in technical literature [8]. Qu et al. presented a control method of air source heat pump and verified it experimentally and the experimental results showed that the control strategy could realize the control of evaporating temperature and intermediate pressure with high accuracy and sensitivity to adjust the system load variation [9]. Park[10] set up a mathematical model of R134a/R410A cascade heat pump to predict the optimal intermediate temperature to make sure that the system can run with higher COP based on thermodynamic laws and thermodynamic properties of refrigerants. Kim[11] verified the optimal intermediate temperature by experimental investigation conducted on an air to water heat pump, and the numerical model can predict the experimental results with high accuracy. Hansaem [12] et.al studied the performance of a cascade heat pump system by quasi-steady state analysis and Cho [13] et al. investigated a two-stage CO_2 system in cooling mode operation.

As a main contributor to productivity and employment in China, the petroleum refining industry consumed approximately 15% [14] of industrial fuel oil and 10% of industrial coal. During recent years, capacity expansion activities had been carried out for lots of petroleum refining companies, but during the design and construction, the heating capacity between the new and old facilities was not integrated well. Accordingly, lots of low temperature waste heating was produced. Take a a 10 million-ton refinery, the low temperature waste heat was about $1000 \times 10^4 \sim 12000 \times 10^4$kcal / h, the current comprehensive utilization efficiency was only 30% to 50%, through comprehensive optimization of low temperature heat, the comprehensive utilization rate could increase to about 70%.In general, its energy saving potential accounted for 10% to 20% of the total energy saving potential of the refinery.

Although a lot of manpower and material resources were invested to carry on the transformation to the low temperature waste heating(below 50 ℃) use system during China's "Eleventh Five-Year" period, but depending on the energy-intensive characteristic, more cost-effective investments for energy system optimization may be a useful strategy to improve the competitiveness of China's refining industry.

Based on the current situation of the petroleum refining industry, in this study, a water source cascade heat pump system using a new binary near-zeotropic refrigerant named BY-3[15] and R245fa as working fluid in the low-stage and high –stage refrigeration cycle, respectively, was studied numerically and experimentally. The system was simulated at different water inlet temperature and the simulated results were validated by the experimental results with high accuracy.

2. Methodology

2.1 Working principle of HTCHP system

The working principle for the cascade heat pump system is:

a. For the low-stage refrigerant cycle, the new near-zeotropic mixture named BY-3 absorbs heat in the evaporator while evaporating. The low pressure and temperature BY-3 vapor comes out from the evaporator is then compressed in the hermetic scroll compressor made by Danfoss into high pressure and temperature vapor. The BY-3 vapor discharges from the low-stage cycle compressor then releases heat in the low-stage cycle condenser and turns into liquid with high pressure and temperature. Then the liquid goes into the EEV of the low-stage cycle and changes to liquid with low pressure and temperature, and then the liquid goes into the low-stage cycle evaporator to finish the low-stage refrigerant cycle.

b. For the high-stage refrigerant cycle, the working fluid R245fa absorbs heat in the evaporator while evaporating. The low pressure and temperature R245fa vapor comes out from the evaporator is then compressed in the hermetic scroll compressor made and modified by Danfoss into high pressure and temperature vapor. The R245fa vapor discharges from the high-stage cycle compressor then releases heat in the high-stage cycle condenser and turns into liquid with high pressure and temperature. Then the liquid goes into the EEV of the high-stage cycle and changes to liquid with low pressure and temperature, and then the liquid goes into the high-stage cycle evaporator to finish the high-stage refrigerant cycle.

The P-h and T-s diagram of the cascade system are shown in **Fig.1** and the heat flow of a cascade heat pump system is shown in **Fig. 2.**

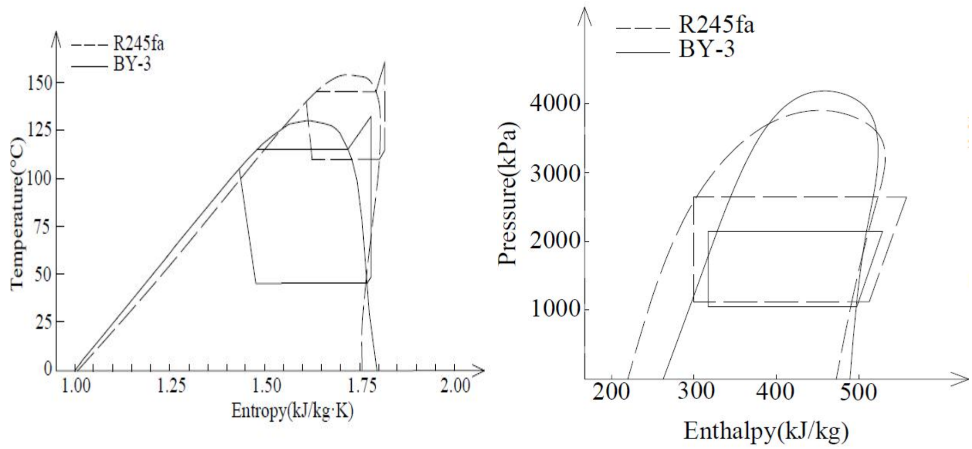


Fig.1. P-h and T-s diagram of cascade cycle

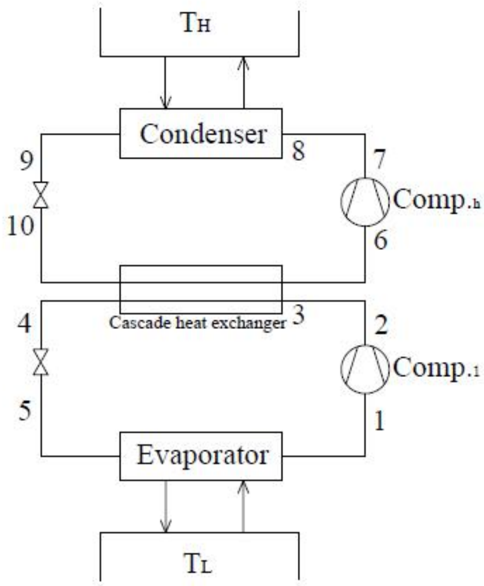


Fig.2 Principle of Cascade heat pump system

2.2 Modeling

A new mathematical model is proposed to evaluate the system performance of the cascade high temperature heat pump. The model is based on the seven main parts of the cascade system: evaporator, cascade heat exchanger, condenser, high-stage and low-stage compressor, high-stage and low-stage EEV.

Some hypotheses are necessary to setup the system model:

(1) The refrigerants in the cascade heat pump are in steady state.

(2) Refrigerants are saturated at the outlet of the evaporator, the cascade heat exchanger and the condenser.

(3) The heat dissipation from the cascade heat exchangers and pipes to environment is negligible, i.e. the heat released from the low-temperature water is totally transferred to the refrigerant in the low-stage refrigerant cycle.

(4) The pressure loss in heat exchangers and the pipes is negligible.

(5) The compression in the compressors is isentropic.

According to the hypotheses mentioned above, the conservation of the mass and energy formula of the refrigerant can be acquired as shown in **Eq.(1).**and **Eq.(2).**

$$\sum_{in} m = \sum_{out} m \tag{1}$$

$$Q - W + \sum_{in} mh - \sum_{out} mh = 0 \tag{2}$$

2.2.1 Evaporator in the low-stage cycle

According to the principle described above, the heat transfer in the low-stage cycle can be described in:

Low temperature heat source side:

$$Q_l = c_w m_{evap,w}\left(t_{evap,w,in} - t_{evap,w,out}\right) \tag{3}$$

Refrigerant side:

$$Q_l = (h_5 - h_1) m_{l,ref}$$

(4)

According to **Eq**. **(3)** and **(4)**, the mass flow rate of refrigerant in the low-stage cycle can be described as:

$$m_{l,ref} = \frac{Q_l}{h_5 - h_1} \tag{5}$$

The specific heat of water (C_w) in **Eq.3** is shown in **Eq.6,** and based on a regression analysis, the correlation coefficient can be 100% in high temperature zone of 60-145°C [16].

$$C_w(t) = 4.20734 - 1.02 \times 10^{-3}t + 1.10155 \times 10^{-5}t^2 + 5.53737{\times}10^{-20}t^3 \tag{6}$$

To simply the calculation, the expression of C_w can be described as shown in **Eq.7**:

$$C_w = \frac{C_{,w}(t_{w,in}) + C_w(t_{w,out})}{2} \tag{7}$$

2.2.2 Hermetic scroll compressor

Hermetic scroll compressors were chosen in this paper due to the excellent properties such as simple structure, high reliability, low vibration, high efficiency, and low noise [17].

The input power of the compressor P_{in} in low-stage and high –stage refrigerant cycle is described in Eq.8 and 9:

$$P_{in,l} = \frac{p_{theo,l}}{\eta_s \eta_e \eta_m} = \frac{m_{ref,l}(h_2 - h_1)}{\eta_s \eta_e \eta_m} \tag{8}$$

$$P_{in,h} = \frac{p_{theo,h}}{\eta_s \eta_e \eta_m} = \frac{m_{ref,h}(h_7 - h_6)}{\eta_s \eta_e \eta_m} \tag{9}$$

Where h_1 and h_2 refers to the enthalpy of the refrigerant at the inlet and outlet of the low-stage cycle compressor, h_6 and h_7 refers to the enthalpy of the refrigerant at the inlet and outlet of the high-stage cycle compressor and η_s, η_m and η_e refer to the isentropic efficiency (refers to AHRI540-2004 standard rating condition), the volumetric efficiency and the motor efficiency, and normally the values for fixed speed scroll compressor are 0.7, 0.8 and 0.85, respectively.

2.2.3 Cascade heat exchanger

Condenser in the low-stage refrigerant cycle:

$$Q_{cond,l} = m_{l,ref}(h_3 - h_4) \quad (10)$$

Where h_3 and h_4 refers to the enthalpy at the inlet and outlet of the condenser in the low-stage refrigerant cycle.

Evaporator in the high-stage refrigerant cycle:

$$Q_{evap,h} = m_{ref,h}(h_{10} - h_6) \quad (11)$$

2.2.4 Condenser

Water side:

$$Q_h = C_{cond,w,h}(t_{cond,w,out,h} - t_{cond,w,in,h}) \quad (12)$$

Where Q_h , $C_{cond,w,h}$, $t_{cond,w,out,h}$ and $t_{cond,w,in,h}$ refers to the heating capacity in the high-stage refrigerant cycle, the specific heat of the water in the condenser in the high-stage refrigerant cycle, the water temperature at the outlet and inlet of the condenser in the high-stage refrigerant cycle, respectively.

Refrigerant side:

$$Q_h = m_{ref,h}(h_8 - h_9) \quad (13)$$

2.2.5 Electronic expansion valve

For the low-stage cycle, the refrigerant mass and enthalpy can be described in Eq.13 and14.

$$m_4 = m_5 = m_{l,ref} \tag{14}$$

$$h_4 = h_5 \tag{15}$$

For the high-stage cycle, the refrigerant mass and enthalpy can be described in Eq.13 and 14.

$$m_9 = m_{10} = m_{h,ref} \tag{16}$$

$$h_9 = h_{10} \tag{17}$$

According to the precious analysis, COP can be obtained as below.

$$COP = \frac{Q_h}{P} = \frac{Q_h}{P_{in,l} + P_{in,h}} \tag{18}$$

3. Test rig setup

3.1 Refrigerant and test rig

Refrigerant is very important in the HTCHP system which can affect the performance and the safe-operation, etc., and during the selection of the refrigerant some rules need to be considered [18-20], such as eco-friendly, good at thermodynamics, flammability and toxicity. According to these concerns, a new near-zeotropic mixture named BY-3 was proposed as the working fluid in the low-stage refrigerant cycle, and R245fa was proposed to use in the high-stage refrigerant cycle. The basic properties for BY-3 and R245fa are shown in Table 1.The properties of the refrigerant in this paper are using the data from the software NISTREFPROP9.1.

Table 1The properties of BY-3 and R245fa

Substance	T_c(°C)	P_c(MP_a)	ODP	GWP(100yr)	Security Level
BY-3	126.2	4.664	0.04	2100	A^a1^b
R245fa	154.01	3.65	0	950	B^a1^b

Note: a, According to TLV-TWA [21], while the refrigerant concentration is equal to or less than $400*10^{-4}$%, it is non-toxic; b, There is no flame spreading in the air if the refrigerant is tested under the condition of 102.3kPa and 21℃.

Based on the physical parameters of BY-3, a regression analysis was done and shown in Eq.19-20. The analysis results suggested the correlation coefficients are 99. 87% and 99.89% for liquid and vapor, respectively.

$$h_{l,BY3} = 9 \times 10^{-5} t^3 - 1.05 \times 10^{-2} \times t^2 + 2.552 \times t + 191.72 \quad (19)$$

$$h_{v,BY3} = -1 \times 10^{-4} t^3 + 1.74 \times 10^{-2} t^2 - 2.068 \times 10^{-1} t + 515.07 \quad (20)$$

A regression analysis was also done based on the physical parameters of R245fa as shown in Eq.21-22.The analysis results suggest that the correlation coefficients are 99.88% and 99.92% for liquid and vapor, respectively.

$$h_{l,R245fa} = 5 \times 10^{-5} t^3 - 1.17 \times 10^{-2} \times t^2 + 2.3858 \times t + 167.74 \quad (21)$$

$$h_{v,R245fa} = -8 \times 10^{-5} t^3 + 2.14 \times 10^{-2} t^2 - 1.1363 \times 10^{-1} t + 451.47 \quad (22)$$

The t-h diagram of BY-3 and R245fa is shown in **Fig 3** and**4.**

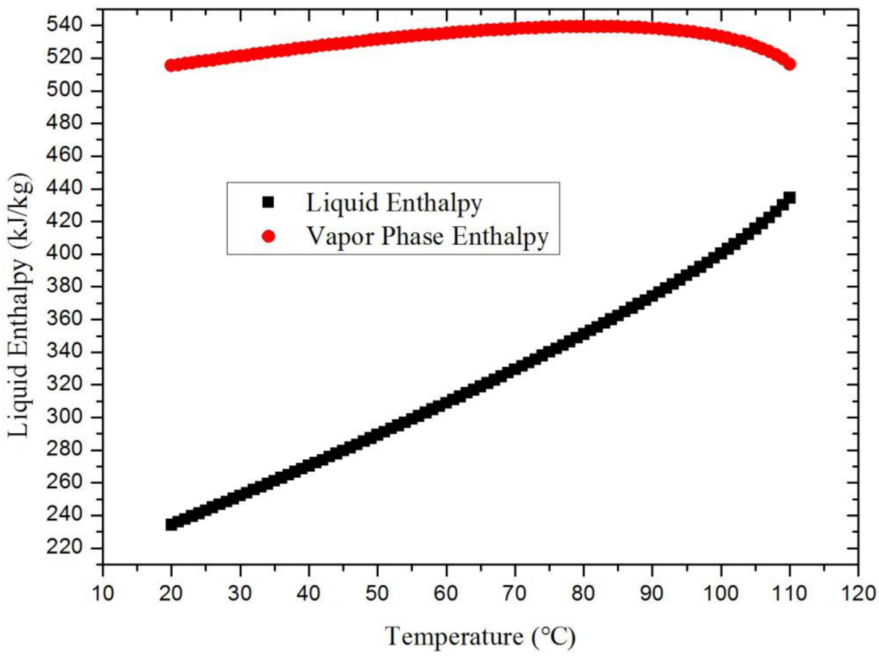


Fig.3. t-h diagram of BY-3

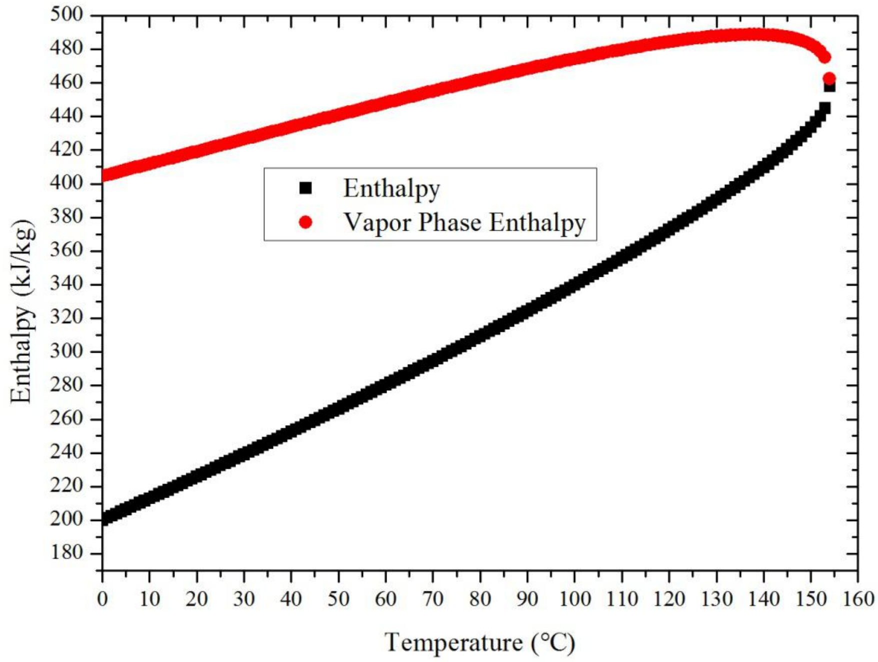


Fig.4. t-h diagram of R245fa

Based on the theoretical analysis above, one test rig was designed in this research to verify the performance of the HTCHP. The test rig was composed of three independent parts: the refrigerant cycle consisted of the low-stage refrigerant and the high-stage refrigerant cycle; a low-temperature water tank with an electric heater which can supply water at the constant temperature so that the evaporating temperature in the evaporator could be controlled and a hermetic high temperature water reservoir was used to support the high-temperature water cycle. The schematic of the HTCHP is shown in **Fig.5.**

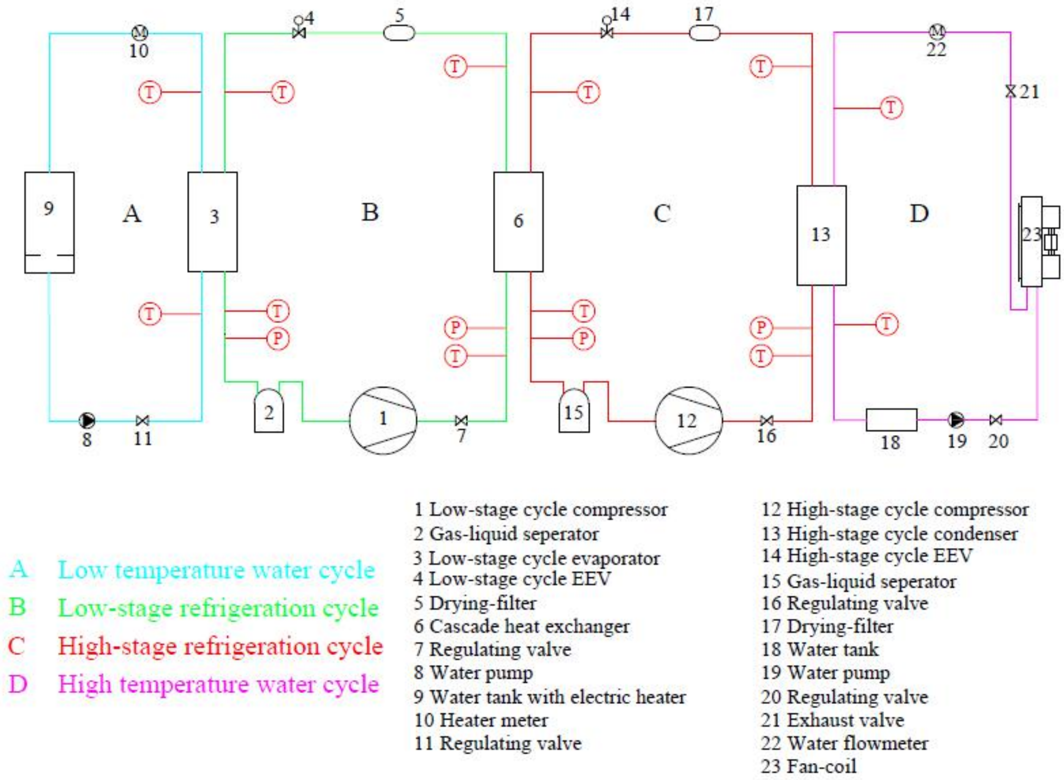


Fig.5. Schematic diagram of the HTCHP test rig

The refrigerant liquid in the low-stage cycle absorbed heat from the low-temperature water and turned into vapor, and then the low temperature and pressure refrigerant vapor was compressed in the low-stage cycle compressor into high temperature and pressure vapor. The cascade heat exchanger of the refrigerant cycle was a double-pipe heat exchanger so that the long flow path can make sure that the refrigerant in the low and high-stage cycle can change heat sufficiently. The low temperature and pressure refrigerant liquid (R245fa) in the high-stage cycle absorbed heat from the high temperature and pressure BY-3 vapor and then was compressed in the high-stage cycle compressor into high temperature and pressure vapor. And then the heat was transferred to the water in the fan coil. In order to reduce the heat emitting to the environment, all the pipes of the cascade system are packaged with thermal insulation material.

The compressors used in this research were made and modified by Danfoss. The specification of the compressors and heat exchangers is shown in **Table** 2 and 3.

Table2: Parameters of the compressors used in the cascade experimental system

Unit	Model	Manufacture	Rated Load Amps(A)	Cooling capacity(w) @ARI 50HZ	Power input(kW) @ARI 50HZ	Swept volume (cm3/rev)	Pressure vessel code
BY-3&R245fa Compressor	SH090A4ALC	Danfoss	14.3	22490	7.215	88.4	PED

Table 3: Parameters of the heat exchangers

Unit	Model	Design pressure(Bar)	Design temperature(℃)	Relative directions of fluids	Heat Exchanged (kW)	Pressure vessel code
evaporator	CB60-26M	40	-196.0/225.0	Countercurrent	20	PED
Cascade heat exchanger	n/a	40	-196.0/225.0	Countercurrent	25	PED
Condens	CB60-32	40	-196.0/225.0	Countercurr	36	PED

er	L	ent

3.2 Compressor setup for high-stage cycle

Two hermetic scroll compressors were used in this research. Some necessary modifications were done in the model shop of Danfoss in order to meet the high discharge temperature in the high-stage cycle.

The scroll compressor is designed to compress the refrigerant gas with low pressure and temperature to high pressure and high temperature state. The orbiting and fixed scrolls are in the upside of the compressor, and the compression process of the refrigerant gas is proceeding in these two scrolls. The working process in the compressor can be described into three steps: suction, compression and discharge as shown in **Fig.6**. The compressor working in the high discharge application will cause the distortion of scrolls and further lead to serious worn and unexpected compressor motor trapped. In order to avoid this kind of issue, the scroll assembly setting was changed. The clearance between the orbiting and fixed scrolls was enlarged 10～20 μm，which could compensate the scroll heat extension，and it was useful to avoid the worn and increase the compressor reliability.

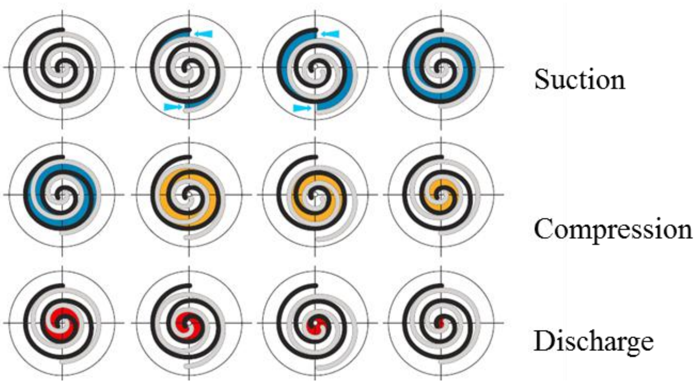


Fig.6. Refrigerant compressing process in the scroll

The protector parameter of the compressor motor in the high-stage cycle need to be adjusted since that the high-stage compressor worked in a high suction temperature which will cause the unexpected compressor motor trapped. Normally, the set for motor protector is 85-115℃,which means if the suction temperature is higher than 115℃,the protector will be trapped and the compressor stops working, and when the motor temperature is lower than 85℃, the circuit will become close again so that the compressor can be restarted. In this paper, the protector parameter of the high-stage cycle was changed to 115-145℃ according to the working condition.

In order to adapt the high discharge temperature in the high-stage cycle, a high-temperature applicative lubricant was used in this research.

Table 4 The properties of lubricant used in high-stage cycle

Item	Kinematic viscosity(40℃)	Pour point	Water content	Density (15℃)	Specific Resistance
unit	mm^2/s	℃	ppm	g/m^3	TΩm
specification	61.2-74.8	-30	200	0.929-0.945	0.01

3.3 Measurement equipment and data acquisition

In this study, the signal of the mass flowmeter, the temperature and the pressure were all sent to Agilent 34972A data acquisition machine for further analysis. The values of power consumption and heat meter were recorded in the same time with the temperatures. The parameters of the measurement equipment are shown in **Table 5** as follows:

Table5: Parameters of the measurement equipment used in the cascade experimental system

Part	model	accuracy	comment
Pt100	n/a	+/-0.3℃	n/a

Pressure transducer(suction)	HSTL-800A,0-1.5MPa	0.2%F.S	Beijing Zhong xing Ltd. corp
Pressure transducer(discharge)	HSTL-800A,0-5.0MPa	0.2%F.S	Beijing Zhong xing Ltd. corp
Heat meter	IC-3	+/-2%	Yidu
Mass flowmeter	HH-LUCB-32	1.0 grade	Huai an
Power meter	DED194E-9SYK02	+/-(0.5%+1)	Shanghai Dingde
Data acquisition	34972A	n/a	Agilent

3.4 Measurement Uncertainty

The uncertainty evaluation is necessary in order to evaluate the test data more effectively. Normally, uncertainty is composed of Uncertainty Type A and Uncertainty Type B [22]. And the uncertainty discussed in this paper is relative overall uncertainty. The basic equation of uncertainty analysis [23] is shown in Eq. (19).

$$\delta R = \left\{ \sum_{i=1}^{N} \left(\frac{\delta R}{\delta X_i} \delta X_i \right)^2 \right\}^{1/2} \tag{19}$$

Where Xi, δXi and δ R refers to the measurement value, the uncertainty of Xi and the overall uncertainty, respectively.

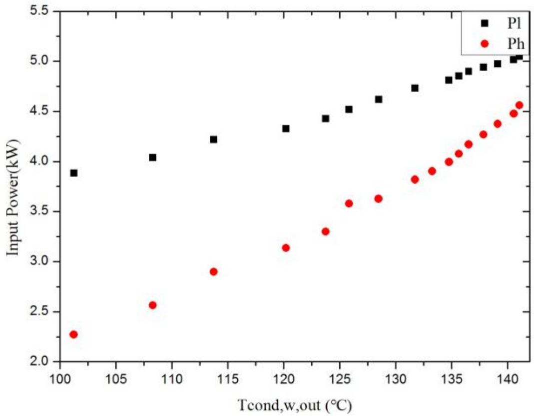
The maximum relative overall uncertainties of COP and heating capacity are 4.15% and 4.30%, respectively.

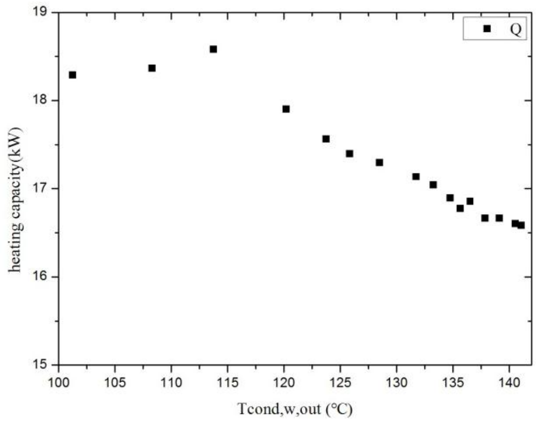
4. Results and discussion

4.1 Experimental results and discussion

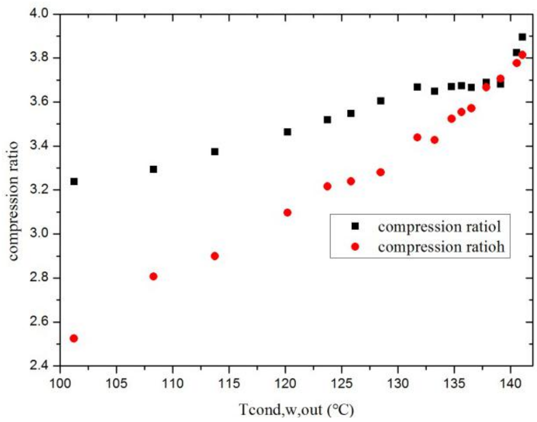
4.1.1 *Performance variation of cascade system due to the water outlet* *temperature in high-stage refrigerant cycle*

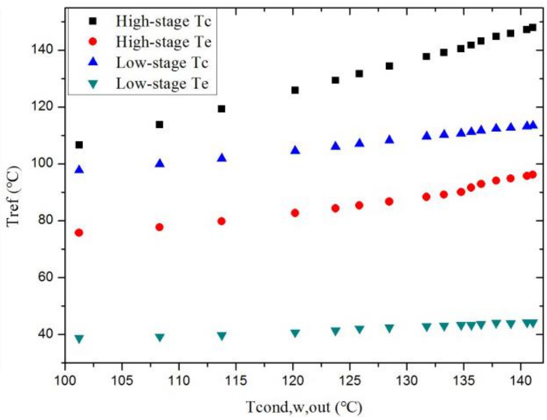
Fig7 shows the experimental results (water inlet temperature at 50 ℃) at the optimum refrigerant charge amount of R245fa equals to 3400g and BY-3 equals to 2000g in the low & high-stage refrigerant cycle, respectively. From Fig 7-a, we can see that with the water outlet temperature increasing, the input power in both low-stage and high-stage refrigerant cycle increased accordingly due to the compression ratio increased in the low and high-stage refrigerant cycle. From Fig7-d, it could be found that the evaporating temperature in high-stage refrigerant cycle kept nearly stable before the water outlet temperature reached 115℃,and then it increased according to the water outlet temperature. Consequently, the heating capacity increased firstly and then when the water outlet temperature reached 115℃, the heating capacity decreased according to the water outlet temperature increasing. The temperature difference between DT in high-stage and water outlet temperature is about 5-6 ℃ , and the value increased a little when the water outlet temperature reached 130℃ due to the decreasing latent heat of the refrigerant. The total COP of the HTCHP decreased according to the increasing of the water outlet temperature in the high-stage cycle condenser due to the input power and heating capacity changing.



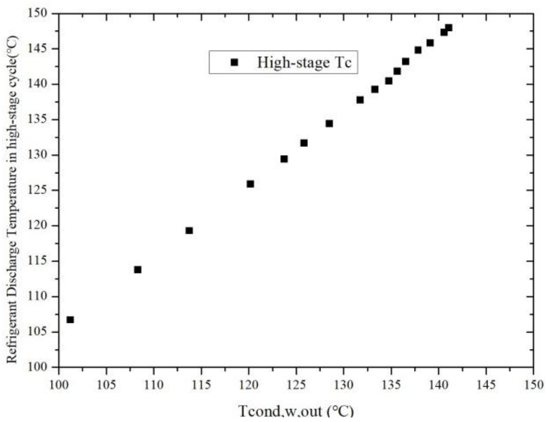


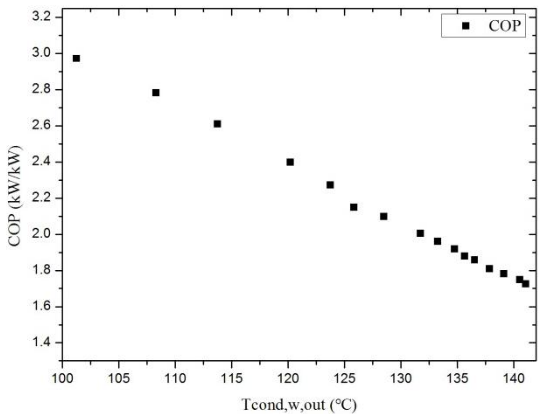
(a) Input Power (b) heating capacity





(c) compression ratio (d) refrigerant temperature



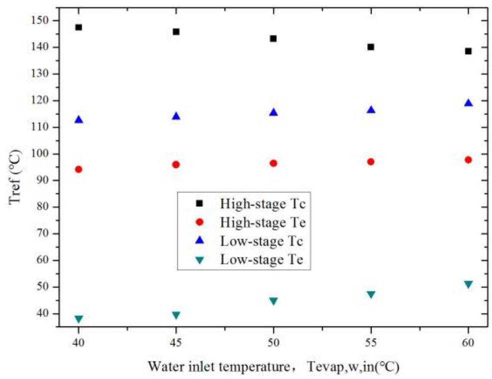


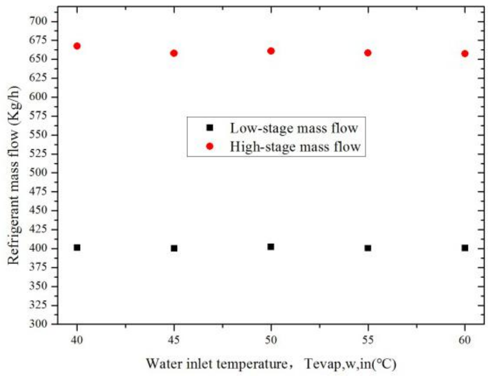
(e) High-stage DT (f) COP

Fig. 7. Variations of (a) Input Power , (b) heating capacity,(c) compression ratio, (d) refrigerant temperature, (e) High-stage DT and (f) COP according to water outlet temperature in high-stage condenser

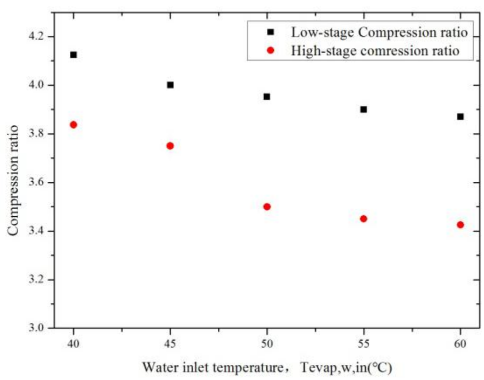
4.1.2 *Performance variation of cascade system due to the water inlet temperature in low-stage refrigerant cycle*

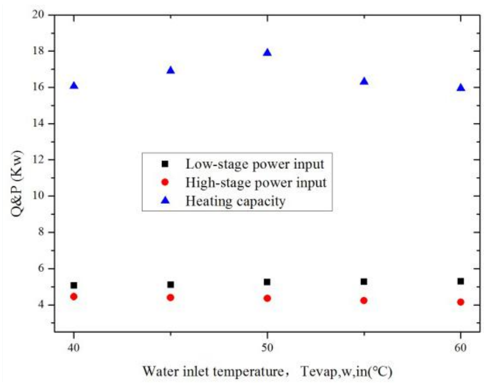
The performance of the cascade system was investigated by changing the water inlet temperature in the evaporator of the low-stage cycle at the optimum refrigerant charge amount of 3400g and 2000g in the low-stage and high-stage refrigerant cycle, respectively. The water flow rate in the low-stage cycle was fixed at 1630kg/h.



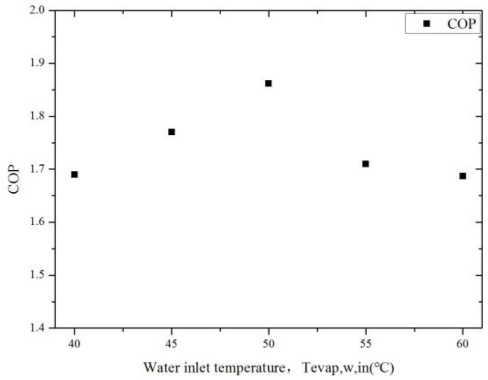


(a) refrigerant temperature (b) mass flow





(c) compression ratio (d) power input and heating capacity



(e) COP

Fig.8. Variations of (a) refrigerant temperature (b) mass flow and(c) compression ratio (d) power input and heating capacity (e) COP according to the water inlet temperature in the low-stage evaporator

As shown in **Fig.8**, the change of the refrigerant temperatures, the refrigerant mass flow, the compression ratio, input power and heating capacity and COP according to the water inlet temperature were studied.

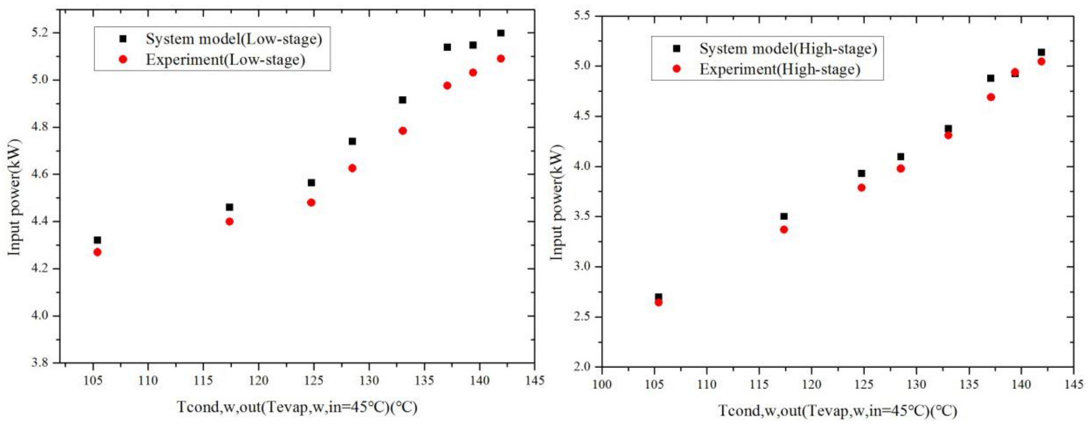
The low-stage cycle evaporating temperature increased due to the increasing water inlet temperature. The condensing temperature in the low-stage cycle increased according to the water inlet temperature and the rate of the condensing temperature is less than the evaporating temperature due to the decreasing compression ratio in the low-stage cycle as shown in Fig.8-c. The high-stage cycle evaporating temperature increased due to the increasing condensing temperature in the low-stage cycle according to the water inlet temperature. The high-stage cycle condensing temperature decreased due the decreasing rate of the compression ratio in the high-stage cycle is larger than the increasing ratio of the evaporating temperature in the high-stage cycle. The input power in the low and high-stage cycle kept nearly the same due to the constant refrigerant mass flow. The heating capacity of the cascade system increased firstly and when the water inlet temperature reached 50℃, it reached the maximum and then it decreased according to the water inlet temperature increasing because of the compressor running map(at the same evaporating temperature, when the condensing temperature increases, power input will increase while the cooling capacity will decrease). Accordingly, the COP of the cascade heat pump system followed the same tendency as the heating capacity.

4.2 Simulated results

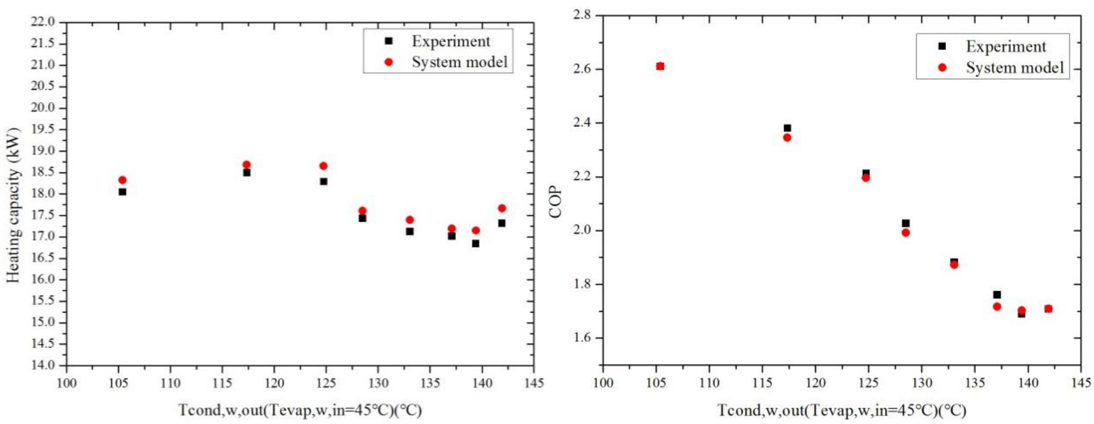
4.2.1 Model validation

The presented model was verified comparing the experimental and simulated results.

The input power (low-stage and high-stage), heating capacity and COP of experiment and system model according to outlet water temperature (with water inlet temperature of 45℃) were shown in Fig. 9.



(a) Input power in low-stage (b) Input power in high-stage



(c) Heating capacity (d) COP

Fig.9 Comparisons of System model and Experimental results of (a) Input power in low-stage (b) Input power in high-stage (c) Heating capacity (d) COP according to the water outlet temperature

In Fig.9, the changing current of the system model results of the input power, heating capacity and COP followed the same tendency with that of the experimental results. In the low-stage refrigeration cycle, the system model of the input power was higher than that of the experimental results with the maximum relative deviation of 3.27%. The system model result of the input power in the high-stage refrigeration cycle was a little higher than that of the experimental result and the maximum deviation is 3.93%.The heating capacity of the system model was higher than that of the experimental result with the maximum deviation of 1.96%. Consequently, the COP of the system model was lower than the experimental result and the maximum deviation was -2.53%. In this case, the system model was validated with the experimental result.

5. Conclusions

The system model was validated via lots of experimental results in this paper, and the following conclusions could be drawn based on the results discussed before:

(1) The experimental results showed that the cascade heat pump system could reach water outlet temperature of 142℃ and the maximum temperature lift could reach 100℃ which was the highest level in the opening literature. BY-3 was proved to have good performance as the working fluid in the low-stage refrigeration cycle. The pressure ratio in the high and low-stage refrigeration cycle was 3.4 and 3.9, respectively with the system COP of 1.72.

(2) The system model was validated with high feasibility via the experimental results. The maximum relative deviation of input power in high & low-stage, the heating capacity and the COP between the system model and the experimental results was 3.93%,3.27%,1.96% and -2.53%, respectively.

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