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Performance Analysis and Optimization of Heat Pump Steam Generator System Based on Exergy Analysis

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ABSTRACT

It has becoming a developing trend to replace boilers with heat pumps driven by electricity under the global carbon neutrality strategy. The exergy loss and thermal performance of the conventional circulating flash heat pump steam generator (CF-SGHP) system were researched. A new direct evaporative heat pump steam generator (DE-SGHP) system was proposed in order to reduce total exergy loss. Then the exergy loss and thermal performance of the two systems was analyzed comparatively under different discharge steam temperature (T_{out}), condensation temperature (T_{cond}) and heat source water temperature (T_{hsw}). The results are as follows: The exergy efficiency ($\eta_{ex,sys}$) of the CF-SGHP system is 62.98%. In addition to the exergy loss caused by the irreversible process of the compressor, the relative maximum exergy loss occurred in the flash cycle process, whose exergy loss rate is 23.24%; The exergy loss of DE-SGHP system is always lower and the COP is always higher than that of the CF-SGHP system within the range of T_{out}, T_{cond} and T_{hsw}; There existed an optimal T_{cond} of 91 ℃ in the DE-SGHP system to maximum the $\eta_{ex,sys}$ of 67.74% and the COP of 2.71 when T_{out} is 165℃ and T_{hsw} is 60℃, that indicated to have a broader development prospect.

Keywords: Exergy analysis; Performance optimization; CF-SGHP; DE-SGHP

NONMENCLATURE

Symbols	
COP	Coefficient of performance
η_{ex}	Exergy loss rate /%
e	Specific exergy /kW·kg^{-1}
E_x	Exergy /kW
E_{heat}	Heat flow Exergy /kW
E_{power}	Power Exergy /kW
I	Exergy loss /kW
h_{in}	Inflow specific enthalpy of control volume /kW·kg^{-1}
h_{out}	Outflow specific enthalpy of control volume /kW·kg^{-1}
s	Specific entropy /kJ·kg^{-1}·K^{-1}
T	Temperature /℃
T_{cond}	Condensation temperature of refrigerants /℃
T_{dsc}	Discharge supercooling degree of condenser /℃
T_{hsw}	Temperature of heat source water /℃
T_{out}	Temperature of discharge steam /℃
T_{ssh}	Suction superheat degree of the compressor /℃
m_{in}	Inflow mass of control volume / kg/s
m_{out}	Ouflow mass of control volume / kg/s
W_{in}	Work on control volume /kW
W_{out}	Work by control volume /kW
W_{per}	Power consumption of generating a ton steam /kW
η	Efficiency /%
Subscript	
0	Outside environment
ex	Exergy
opt	Optimal value
sys	System
Abbreviations	
CF-SGHP	Circulating flash heat pump steam generator
DE-SGHP	Direct evaporative heat pump steam generator
CIRP	Circulating pump
COND	Condenser
CR	Circulating ratio
ECO	Economizer
EVAP	Evaporator
EV 1	Expansion valve 1
EV 2	Expansion valve 2
EV 3	Expansion valve 3
FLST	Flash tank
GWP	Global Warming and Poverty
NBP	Normal boiling point
ODP	Ozone Depletion Potential
PREH	Preheater
REFC	Refrigerant compressor
REFC	Refrigerant compressor
STEC	Steam compressor

1. INTRODUCTION

In the context of "global carbon neutrality", the electric driven heat pump has become a development trend to replace traditional coal and gas boilers to produce high temperature water or industrial steam with higher energy conversion efficiency and lower CO_2 emissions [1-3]. It is estimated that about 16%（272 PJ）of the industrial heat demand (1,909 PJ in 2012) in Germany could be provided by heat pump technology up to 100℃[4]. The technical heating potential by heat pumps up to 150℃ reaches 626 PJ in European[5]. It is about 22.5%-31.5% of industrial heat demand in China could be provided by heat pump technology up to 170℃,that decreased nearly 200 Mton CO_2 emissions[6]. Therefore, the high temperature heat pump to provide above 100 ℃ heating energy, especially industrial steam, has a great development prospects and application potential.

The World's first Steam Heat Pump Engine named "Steam Grow Heat Pump" is launched by Japanese Kobe Steel in 2011[7], it uses two high temperature twin-screw compression heat pump system to produce steam of 165℃ adopted R245fa/R718 refrigerants, which is described in detail in the following sections. Subsequently, many companies and scientific experts carried out a large number of researches. Ochsner company design a high temperature heat pump uses a screw compressor with a condensation temperature of 95-130°C and a heat capacity of 170-750kW[8]. The Hybrid Energy developed a hybrid heat pump combined absorption heat pump with mechanical compression heat pump, that used a non-azeotropic refrigerant of ammonia and water[9-10]. The vicking heating engines developed a heatbooster S4 uesd a piston company with a heat sink temperature of 150°C and a heat capacity of 28-188kw, that adopted the R1336mzz(Z) refrigerant[11-12].

The steam generator of Kobe Steel used a Circulating flash heat pump generator (CF-SGHP) system, which is considered a key research content. Zhao[13] carried out simulation and experimental research on partial CF-SGHP system，the results shown that the COP is varied of 3.06 to 6.43 at the evaporation temperature increased from 50℃ to 85℃, which could generate saturated steam of 200 kPa. Wang [14] found the COP of system used the mixed refrigerants of R600/R245fa is higher than system used R245fa at the condensation temperature of 100℃. Helminger[11] proposed a single-stage high temperature heat pump circulation system using R1336mzz(Z) as the circulating refrigerant and installed an IHX (intermediate heat exchanger), the results shown that using an IHX could increses the COP of heat pump cycle by a 4% to 47% higher than the simple heat pump cycle system. Wu[15] proposed a water vapor HTHP system using a water injection twin-screw compressor as the driving equipment, the experimental results shown that the COP ranges from 3.64 to 4.87 when the evaporation temperature is 83-87℃ and the condensation temperature among 120-128℃. However, the exergy analysis and exergy loss research of heat pump steam system are rarely reported.

As stated above, the exergy analysis and performance optimization were proposed for the CF-SGHP system in this paper. A mathematical model has been developed using the engineering equation solver (EES). The exergy loss and thermal performance of the equipments and cycles were obtained. According to the results, the performance optimization of the system could be carried out. The effects of the discharge steam temperature (T_{out}), the condesation temperature (T_{cond}), and the heat source water temperature (T_{hsw}), on coefficient of performance (COP) , power consumption of generate a ton steam (W_{per}), exergy loss (I) and exergy efficiency (η_{ex}) are studied in detail.

2. SYSTEM DESCRIPTION OF CF-SGHP SYSTEM

The flowsheet of the CF-SGHP system is shown in Fig.1, which is mainly divided into refrigeration cycle and water cycle, that including preheater, evaporator, economizer, refrigerant compressor, condenser, flash tank, steam compressor, feed pump, circulating pump and expansion valves, etc. The soften waters at normal temperature enter the system from the water inlet (state point 1). Then they are divided into two parts after heated by the preheater and booster by the water pump, the majority of them (state point 3) enter into the flash tank used to replenish the flash evaporation water. The others enter into steam compressor from the middle nozzle (state point 9), which can reduce the superheat of the steam compression process and improve the compression efficiency. The saturated liquid waters (state point 4) at the bottom of flash tank enter into the condenser of heat pump system by circulating pump. They are heated to high temperature and booster to high pressure water (state point 6) and return to the flash tank passing through the expansion valve 3. The flash steam in flash tank (state point 8) enters into the steam compressor, at last discharges from the system in the form of high temperature and high pressure steam (state point 10). Medium temperature heat source water (state point 11) enters the evaporator and the preheater of CF-SGHP system to release heat successively. Finally, it is discharged from the system as lower temperature water (state point 13). In the refrigerant cycle, the low temperature and low pressure gaseous refrigerants (state point 21) enter the refrigerant compressor from the suction inlet, and be promoted as the gaseous refrigerants of high-temperature and high-pressure (state point 23). Then the refrigerants enter the condenser to release heat through condensation phase transformation, and turns into the high-temperature and high-pressure liquid refrigerants (state point 24). One branch enters the refrigerants compressor through the gas injection inlet (state point 22) after vaporization in the economizer. The main refrigerants also enter the economizer to continue to release heat for further cooling (state point 27), and then enters the evaporator (state point 28) through expansion valves 2. After completing the endothermic gasification process, the refrigerants enter the refrigeant compressor to continue the cycle.

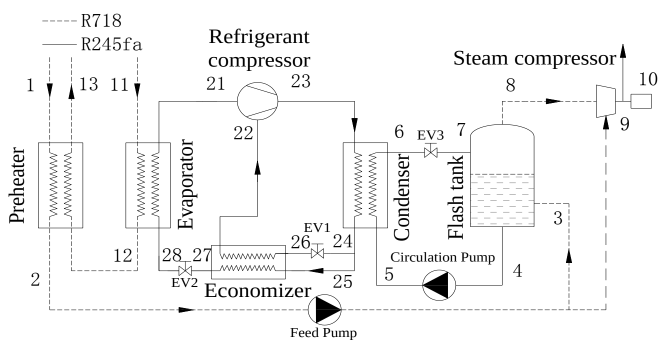


Fig.1. Flowsheet of the circulating flash heat pump steam generator system

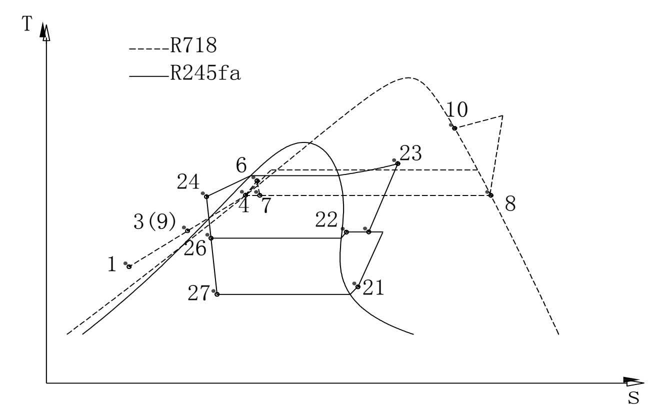


Fig.2. T–S diagram of the refrigerants in the CF-SGHP system

3. MATHEMATICAL MODEL

3.1 Model assumption

The following assumptions are made to simplify the analysis [16-18]:

（1）The cycle is assumed to be stable state and the thermal parameter of each state point is not change with time.

（2）Heat loss and pressure loss of components such as heat exchangers and pipes are ignored；

（3）It is an adiabatic flash process in the flash tank, and the separated steam is saturated dry steam;

（4）It is an adiabatic compression process of the compressor, and the discharge steam of the system is saturated.

3.2 Refrigerant property

In this paper, R245fa and R718 are used as the circulating refrigerant of high temperature heat pump. During the modeling process, thier physical characteristics refer to the physical properties query tool of NIST Refprop. The basic characteristics are shown in the following table.

Table 1 Basic property of R245fa and R718 refrigerant

Refrigerant	Molecular mass	NBP	Critical temperature	Critical pressure	ODP	GWP
R245fa	134	15.3 ℃	153.9℃	3.65 Mpa	0	1030
R718	18	100℃	373.95℃	22.06 Mpa	0	0.2

3.3 Mass and energy balance

$$\sum m_{in} = \sum m_{out} \quad (1)$$

$$\sum m_{in} \bullet h_{in} + \sum W_{in} + \sum Q_{in} = \sum m_{out} \bullet h_{out} + \sum W_{out} + \sum Q_{out} \quad (2)$$

Where the m_{in}, h_{in}, W_{in}, Q_{in} are the inflow mass, inflow specific enthalpy of each control volume, work done on the control volume, and heat absorption of the control volume, respectively; The m_{out}, h_{out}, W_{out}, Q_{out} are the outflow mass and outflow specific enthalpy of each control volume, work done by the control volume and heat release by the control volume, respectively.

3.4 Exergy balance

The part of energy that can be converted into "fully convertible energy" to the maximum extent is called exergy, when the system reversibly transforms from any state to equilibrium with the environmental state. The pressure and temperature of the environmental state (Point 0) in this paper is 0.1013MPa and 293.15K, respectively. The exergy balance equations are as follow [19-20]:

$$\sum m_{in} \bullet e_{in} + \sum E_{heat,in} + \sum E_{power,in} = \sum m_{in} \bullet e_{in} + \sum E_{heat,in} + \sum E_{power,in} + I \quad (3)$$

$$\sum E_{heat} = \sum (1 - T_0 / T) \bullet Q \quad (4)$$

$$e = h - h_0 - T_0 \bullet (s - s_0) \quad (5)$$

Where the e_{in}, $E_{heat,in}$, $E_{power,in}$ are the specific exergy, heat flow exergy and power flow exergy of the inflow control volume, respectively; The e_{out}, $E_{heat,out}$, $E_{power,out}$, I are the specific exergy, heat flow exergy , power flow exergy of the outflow control volume and the exergy loss, respectively; The T_0, h_0, s_0 are the temperature, specific enthalpy and specific entropy of the environmental state.

3.5 Exergy analysis of the equipments and systems

Table 2 Energy equation and exergy analysis equation of the equipments and cycles

Component/system	Energy equation	Exergy loss	Exergy efficiency	Exergy loss rate
Preheater	m_2h_2-m_1h_1=$m_{12}h_{12}$-$m_{13}h_{13}$	(E_{12}-E_{13})-(E_2-E_1)	(E_2-E_1)/(E_{12}-E_{13})	I_{prep}/I_{sys}
Evaporator	$m_{11}h_{11}$-$m_{12}h_{12}$=$m_{21}h_{21}$-$m_{28}h_{28}$	(E_{11}-E_{12})-(E_{21}-E_{28})	(E_{21}-E_{28})/ (E_{11}-E_{12})	I_{prep}/I_{sys}
Economizer	$m_{22}h_{22}$-$m_{26}h_{26}$=$m_{25}h_{25}$-$m_{27}h_{27}$	(E_{25}-E_{27})-(E_{22}-E_{26})	(E_{22}-E_{26})/(E_{25}-E_{27})	I_{econ}/I_{sys}
Refrigerant compressor	$m_{21}h_{21}$+$m_{22}h_{22}$+W_{refc}=$m_{23}h_{23}$	(E_{21}+E_{22}-E_{23})+$E_{power,r}$	(E_{23}-E_{21}-E_{22})/$E_{power,r}$	I_{refc}/I_{sys}
Condenser	$m_{23}h_{23}$-$m_{24}h_{24}$=m_6h_6-m_5h_5	(E_{23}-E_{24})-(E_6-E_5)	(E_6-E_5) /(E_{23}-E_{24})	I_{cond}/I_{sys}
Flash tank	m_3h_3+m_7h_7=m_4h_4+m_8h_8	(E_7+E_3)-(E_4+E_8)	(E_8-E_3)/(E_7-E_4)	I_{flash}/I_{sys}
Circulating pump	W_{pump}=m_4*ΔP/(ρ*η_{pump})	(E_4-$E5$)+W_{pump}	($E5$-E_4)/W_{pump}	I_{pump}/I_{sys}
Steam compressor	m_8h_8+m_9h_9+W_{vapc}=$m_{23}h_{23}$	(E_8+E_9-E_{10})+$E_{power,v}$	(E_{10}-E_8-E_9)/$E_{power,v}$	I_{stec}/I_{sys}
Expansion valve 1	/,	E_{24}-E_{25}-E_{26}	(E_{25}+E_{26})/E_{24}	I_{evI}/I_{sys}
Expansion valve 2	/	E_{27}-E_{28}	E_{27}/E_{27}	I_{evII}/I_{sys}
Expansion valve 3	/	E_6-E_7	E_7/E_6	I_{evIII}/I_{sys}
Refrigerant cycle	$m_{11}h_{11}$+m_5h_5+W_{vapc}=m_6h_6+ $m_{12}h_{12}$	I_{econ}+I_{evap}+I_{cond}+I_{refc}	(E_6-E_5)/($E_{power,r}$+ E_{21}-E_{27})	I_{hp}/I_{sys}
Flash steam cycle	m_3h_3+$m_{23}h_{23}$+$W_{pump*}\eta_p$ =m_8h_8+ $m_{24}h_{24}$	I_{cond}+I_{plat}+I_{pump}	(E_8-E_3)/($E_{power,p}$+ E_6-E_5)	I_{hp}/I_{sys}
Heat pump steam system	m_1h_1+$m_{11}h_{11}$+W_{vapc}+W_{vapr} + W_{pump} =$m_{10}h_{10}$+ $m_{13}h_{13}$	I_{prep}+I_{econ}+I_{evap}+I_{cond} +I_{refc}+I_{flash}+I_{vapc}+I_{pump}	(E_{10}-E_1)/($E_{power,r}$+ $E_{power,p}$+$E_{pover,v}$+E_{11}-E_{13})	1

Exergy analysis was carried out for the equipments and cycles of the system, according to the above calculation equations in chapter 3.3-3.4. The results are shown in Table 2.

3.6 Assessment of SGHP system

The exergy efficiency (η_{ex}), exergy loss rate (η_{el}) and exergy loss coefficient (d_{el}) were evaluated for each equipment of the heat pump steam generator system in this paper. The coefficient of performance (COP) and power consumption of generating a ton steam (W_{per}) were evaluated for the system. The calculation expressions are as follow[21-22]:

$$\eta_{ex} = E_{out} / E_{in} \quad (6)$$

$$I_{sys} = \sum I_i \quad (7)$$

$$\eta_{el} = I_i / I_{sys} \quad (8)$$

$$d_{el} = I_i / E_{in,sys} \quad (9)$$

Where the E_{in}, E_{out} are the inflow exergy and outflow exergy of each control volume respectively, The I_i, I_{sys} are exergy loss of each control volume and total exergy loss of the system respectively. The $E_{in,sys}$ is the input exergy of the system.

$$COP = (m_{10}h_{10} - m_1h_1) / \left(W_{refc} + W_{vapc} + W_{pump}\right) \quad (10)$$

$$W_{per} = (W_{refc} + W_{stec} + W_{pump}) / m_{10} \quad (11)$$

Where the W_{refc}, W_{stec}, W_{pump} are the power consumption of refrigerant compressor, steam compressor and circulating pump of the system, respectively.

4. Exergy analysis and performance optimization

4.1 Model calculation and typical parameters

According to the above results, the Engineering Equation Solver (EES) software was used to program the mathematical model, and the convergence tolerance was set as 10^{-6} to carry out exergy analysis and thermal performance calculation of the SGHP system. The known parameters set during the calculation are shown in Table 3, and the calculation flow chart is shown in Fig.3.

Table 3 Input values of the typical parameters

Typical parameter	Value
Mass flow of rerigerants (m_{21})	1kg·s^{-1}
Condensation temperature of refrigerants (T_{cond})	110℃
Suction superheat degree of the refrigerant compressor (T_{ssh})	5K
Discharge supercooling degree of condenser (T_{dsc})	10K
Flash vaporization pressure (P_{flash})	101.3Kpa
Circulating ratio (CR)	75
Inlet temperature of feed water (T_1)	25℃
Inlet temperature of heat source water (T_{hsw})	50-70℃
Outlet steam temperature (T_{out})	150-165℃
Efficiency of refrigerant compressor (η_{refc})	0.7
Efficiency of steam compressor (η_{stec})	0.65
Efficiency of circulating pump (η_{pump})	0.6

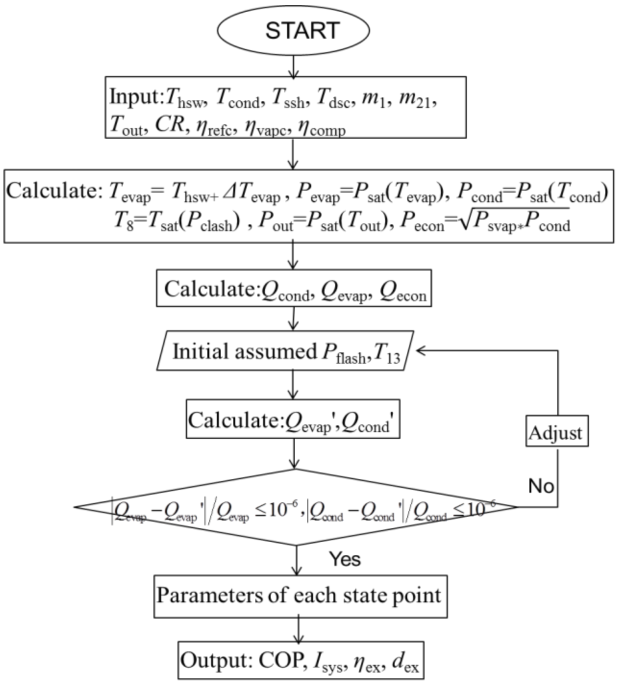


Fig.3. Calculation flow chart of the mathematical model

4.2 Model validation

The comparison and verification results are shown in Fig.4. When the heat source temperature is 50℃, the COP of the system is 2.153. As the heat source temperature rises to 70℃, the COP of the system increases linearly to 2.561. Compared with the data provided by the published literature[23], the maximum error rate is 7.65%, and the average error rate is 4.46%. It can be considered that the mathematical model and calculation method are basically reliable.

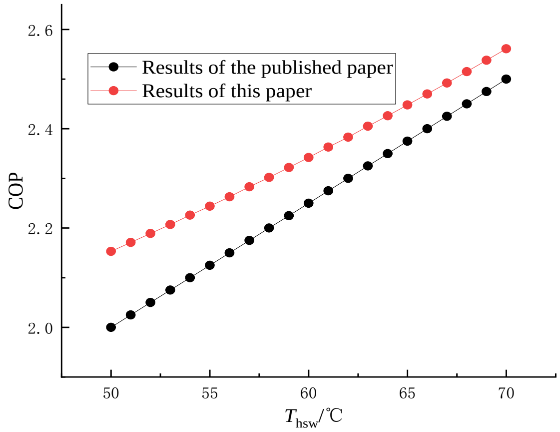


Fig.4 Result of the COP vs. T_{hsw} calculated by paper's model compared with the published literature

4.3 Influence of flash pressure

The flash pressure is the design pressure for the phase transformation of high temperature water in the flash tank, which is drived by reducing pressure of the EV2. The corresponding saturation temperature is the temperature of gas-liquid equilibrium in the tank, and it is an important parameter of the CF-SGHP system. The system performance under different P_{flash} was analyzed in this paper, and it found that there existed an optimal flash pressure ($P_{flash,\ opt}$), which maximizes COP of the system. The $P_{flash,\ opt}$ is 92kPa and the corresponding CR is 75 when T_{hsw} is 60℃ and T_{out} is 165℃. The power consumption of steam compressor (W_{stec}) is decreased with the P_{flash} increasing from 80kPa to 110 kPa, due to the P_{flash} is equal to the steam compressor suction pressure. But the power consumption of circulating pump (W_{pump}) is increased with the variation of $P_{flash.}$ It is because that the circulating mass flow (m_4) and circulating ratio (CR) are increased to generate the same mass flow of steam, due to a higher P_{flash} corresponds to a lower superheating temperature of inflow water. Therefore, the W_{per} decreased first and then increased with the increasing of P_{falsh}, and the COP of the system shown the opposite trend.

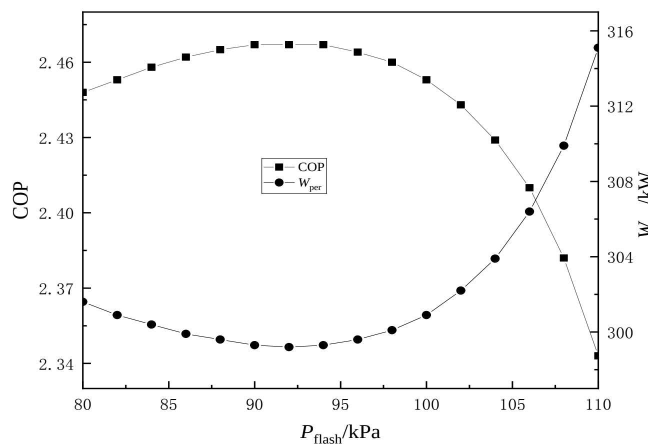


Fig.5 The COP and W_{per} vs. P_{flash} of the CF-SGHP system

4.4 Results of the exergy analysis and thermal performance

According to the mathematical medel established above, the thermodynamic parameters of each state point in the CF-SGHP system are calculated in this paper, the results are shown in Table 4. The exergy analysis results of each device and cycle in the CF-SGHP system are shown in Table 5.

4.5 Results discussed

The above analysis results show that the exergy efficiency of the system ($\eta_{ex,sys}$) is 62.98%. The main exergy losses of CF-SGHP steam system are 11.83kW, 10.38kW and 9.17kW that existed in the refrigerant compressor, steam compressor and flash evaporation cycle. Theie exergy loss rates are 29.97%, 26.30% and 23.24%, respectively. The I_{refc} and I_{stec} are inevitable that caused by the energy loss of electricity to internal energy conversion process and the heat exchange mixing process. And there is no suitable method to reduce exergy loss for high-temperature refrigerant compressors and steam compressors with a large

Table 4 Thermodynamic parameter values of each state point in the CF-SGHP system

State point	T/℃	h/kJ·kg^{-1}	s/kJ·kg^{-1}·K^{-1}	e/kJ·kg^{-1}	E/kW
1	25	104.9	0.3675	0.0794	0.0067
2	51	209.4	0.704	5.9344	0.4983
3	51	210.3	0.7067	6.0429	0.4192
4	99.61	407.8	1.277	36.3594	189.1781
5	99.61	408.2	1.278	36.4663	189.7340
6	106	440.6	1.364	43.6554	227.1389
7	/	440.6	1.365	43.3622	225.6137
8	99.61	2671	7.387	508.4129	35.2686
9	51	209.4	0.704	5.9344	0.0866
10	165	2763	6.707	799.7549	67.1474
11	60	272.2	0.8938	13.0945	39.5454
12	55	230.3	0.7682	8.0141	24.2027
13	54.67	227.4	0.7594	7.6939	23.2355
21	55	446.2	1.779	22.0173	22.0173
22	82.12	463.1	1.781	38.3310	6.1636
23	118.4	485	1.809	52.0228	60.3984
24	100	339.6	1.43	17.7266	20.5806
25	100	339.6	1.43	17.7266	17.7266
26	/	339.6	1.435	16.2609	2.6147
27	87.12	319.8	1.375	14.0499	14.0499
28	/	319.8	1.387	10.5321	10.5321

Table 5 Calculation results of systematic exergy analysis

Component/system	Exergy loss /kW	exergy efficiency /%	exergy loss rate /%	exergy loss coefficient /%	Coefficient of performance
Preheater	0.4391	52.79%	1.11%	0.38%	
Evaporator	3.4566	77.25%	8.76%	3.03%	
Economizer	0.4344	89.09%	1.10%	0.38%	
Refrigerant compressor	10.3788	75.49%	26.30%	9.10%	
Condenser	2.7873	92.95%	7.06%	2.44%	
Flash tank	0.7236	97.97%	1.83%	0.63%	
Circulating pump	11.8252	72.88%	29.97%	10.37%	
Steam compressor	4.0988	9.22%	10.39%	3.59%	
Expansion valve 1	0.2464	98.81%	0.62%	0.22%	
Expansion valve 2	3.5100	74.73%	8.89%	3.08%	
Expansion valve 3	1.5609	99.31%	3.96%	1.37%	
Heat pump refrigerant cycle	9.1706	84.52%	23.24%	8.04%	
Flash steam cycle	21.5371	63.83%	54.58%	18.88%	3.98
Heat pump steam system	39.4611	62.98%	100.00%	34.59%	2.47

pressure ratio. But the exergy loss of the flash evaporation cycle could be reduced by proper design.

The exergy loss of flash evaporation cycle is seious of 9.17kW, accounting for 23.24% of the total exergy loss. In order to reduce total exergic loss of the system, a direct evaporation heat pump steam system (DE-SGHP) is proposed in this paper.

5 Exergy analysis and thermal performance of DE-SGHP system

5.1 System description of the DE-SGHP system

The flowsheet of the DE-SGHP system is shown in Fig.6. The main difference is that it used a steam generator to replaces the flash evaporation cycle, which realizes the purpose of bilateral phase transformation and gas-liquid separation at the same time.

Thermodynamic performance and exergy analysis were carried out again according to the above model and method. The results are shown in Table 6.

Table 6 Calculation results of systematic exergy analysis

Component/system	Exergy loss /kW	exergy efficiency /%	exergy loss rate /%	exergy loss coefficient /%	Coefficient of performance
Preheater	0.4208	52.77%	1.55%	0.41%	
Evaporator	3.4566	77.25%	12.75%	3.37%	
Economizer	0.4344	89.09%	1.60%	0.42%	
Refrigerant compressor	10.3788	75.49%	38.29%	10.13%	
Condenser	2.5553	93.53%	9.43%	2.49%	
Expansion valve 1	0.2464	98.81%	0.80%	0.24%	
Expansion valve 2	3.5100	74.73%	11.47%	3.43%	
Steam compressor	9.6102	73.63%	31.39%	9.38%	
Heat pump refrigerant circulation	17.0715	64.24%	55.77%	16.66%	3.98
Heat pump steam system	30.6125	67.74%	100.00%	29.88%	2.71

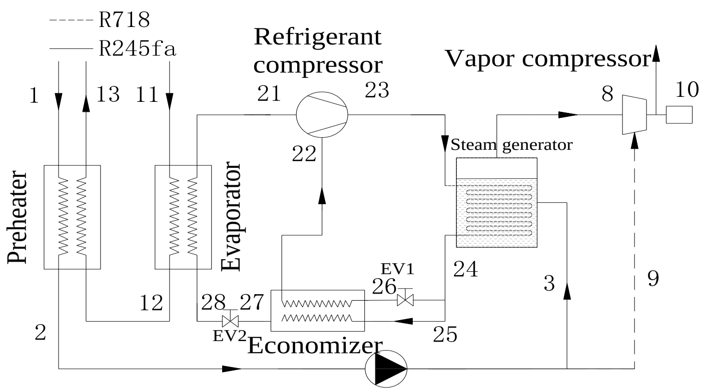
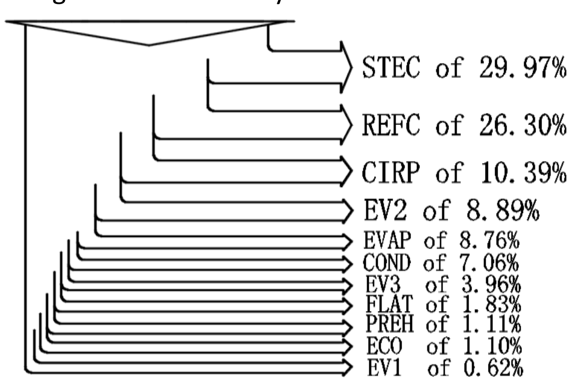


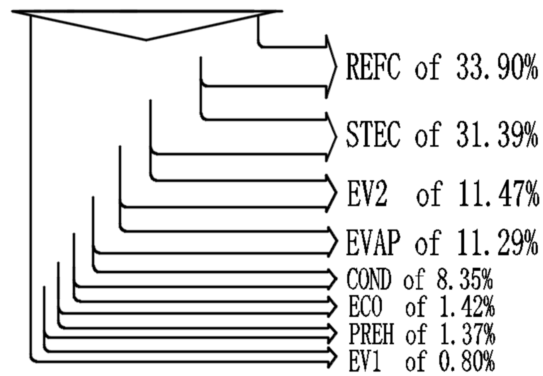
Fig.6. Flowsheet of a direct evaporative heat pump generator steam system

5.2 Exergy flow diagram analysis

Exergy loss of the above two systems was analyzed, and the results were shown in Fig.7. The exergy loss of all heat exchangers is small because the assumption of pressure drop of heat exchanger and pipe is ignored. The majority of exergy losses are occurred in the compressors and expansion valves. It found that the I_{sys} of DE-SGHP decreased from 35.96kW to 27.11kW (24.6%), and the $\eta_{ex,sys}$ of DE-SGHP increased from 62.98% to 67.74% (7.6%), comapared with the exergy flow diagrams of the two systems.



(a) Circulating flash heat pump steam generator system



(b) Direct evaporative heat pump steam generator system

Fig.7. Exergy flow diagram of the two heat pump steam system

5.3 Influence of discharge steam temperature

The variations of I_{sys} and $\eta_{ex,sys}$ of the two systems with different discharge steam temperature are given in Fig.8. The I_{sys} of CF-SGHP system increses from 33.09kW to 35.96kW as the T_{out} increased from 150℃ to 165℃. The $\eta_{ex,sys}$ of DE-SGHP system increses from 62.11% to 62.98% as the T_{out} increased from 150℃ to 165℃. However, the I_{sys} of DE-SGHP system is always lower than that of CF-SGHP syetem with an average decrease of 25.2%, and the $\eta_{ex,sys}$ of DE-SGHP system is always higher than that of the CF-SGHP syetem with an average increase of 18.5%, in the range of T_{out} increasing from 150℃ to 165℃.

The variations of COP and W_{per} of the two systems with different discharge steam temperature are given in Fig.9. The COP of CF-SGHP system decreses from 2.64 to 2.47 as the T_{out} increased from 150℃ to 165℃. The W_{per} of CF-SGHP system increses from 277.5kW to 299.3kW as the T_{out} increased from 150℃ to 165℃. However, the COP of DE-SGHP system is always higher than that of the CF-SGHP system with an average increase of 10.6%, and the W_{per} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 9.5%, in the range of the the T_{out} increasing from 150℃ to 165℃.

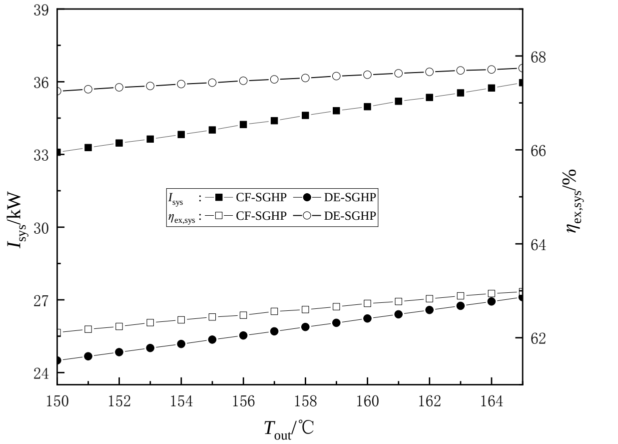


Fig.8. I_{sys} and $\eta_{ex,sys}$ vs. T_{out} of the heat pump steam systems

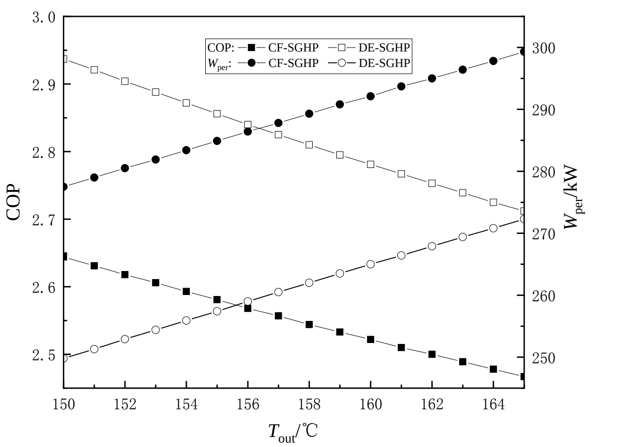


Fig.9. COP and W_{per} vs. T_{out} of the heat pump steam systems

5.4 Influence of condensation temperature

The variations of I_{sys} and $\eta_{ex,sys}$ of the two systems with different condensation temperature are given in Fig.10. Considering the vacuum capacity of the steam compressor, the two systems choose different T_{cond} of 98-110℃ and 85-105℃, respectively. The I_{sys} of CF-SGHP system decreses from 40.63kW to 35.93kW as the T_{cond} increased from 98℃ to 110℃. The $\eta_{ex,sys}$ of CF-SGHP system increses first then decreses and there existed an optimal T_{cond} to maximum the $\eta_{ex,sys}$ value. However, the I_{sys} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 15.9%, and the $\eta_{ex,sys}$ of DE-SGHP system is always higher than that of the CF-SGHP syestem with an average increase of 6.8%.The optimal T_{cond} are 102℃, 91℃ and the maximum $\eta_{ex,sys}$ are 63.6% and 67.6% of the CF-SGHPsystem and DE-SGHP system, respectively.

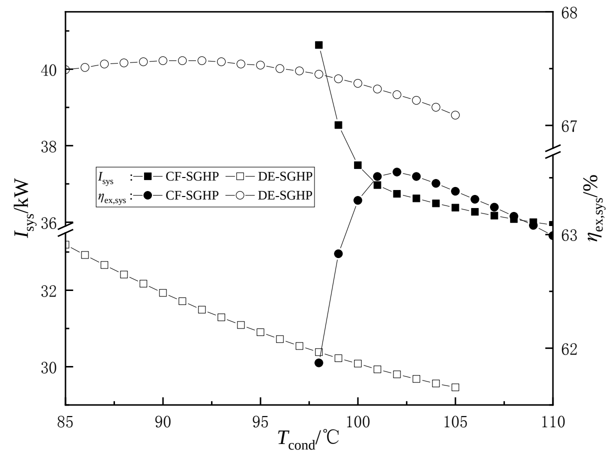


Fig.10. I_{sys} and $\eta_{ex,sys}$ vs. T_{cond} of the heat pump steam systems

The variations of COP and W_{per} of the two systems with different condensation temperature are given in Fig.11. The COP of CF-SGHP system increses first then decreses and there existed an optimal T_{cond} to maximum the COP value. The W_{per} of CF-SGHP system decreses first then increses and there existed an optimal T_{cond} to minimum the W_{per} value. However, the COP of DE-SGHP system is always higher than that of the CF-SGHP system with an average increase of 8.8%, and the W_{per} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 8.1%. The optimal T_{cond} are 102℃, 91℃ and the maximum COP are 2.50 and 2.71 of the CF-SGHP system and DE-SGHP system, respectively.

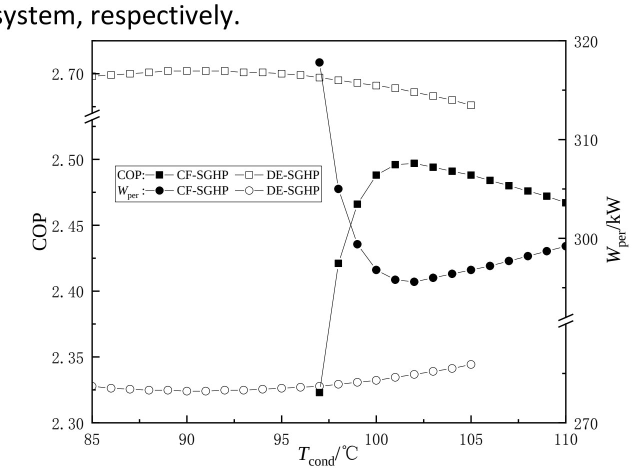


Fig.11. COP and W_{per} vs. T_{cond} of the heat pump steam systems

5.4 Influence of heat source water temperature

The variations of I_{sys} and $\eta_{ex,sys}$ of the two systems with different T_{hsw} are given in Fig.12. The I_{sys} of CF-SGHP system decreses from 39.94kW to 29.87kW as the T_{hsw} increased from 50℃ to 70℃. The $\eta_{ex,sys}$ of CF-SGHP system increses from 61.33% to 66.01% as the T_{hsw} increased from 50℃ to 70℃. However, the I_{sys} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 25.4%, and the $\eta_{ex,sys}$ of DE-SGHP system is always higher than that of the CF-SGHP syestem with an average increase of 7.8%, in the range of T_{hsw} increasing from 50℃ to 70℃.

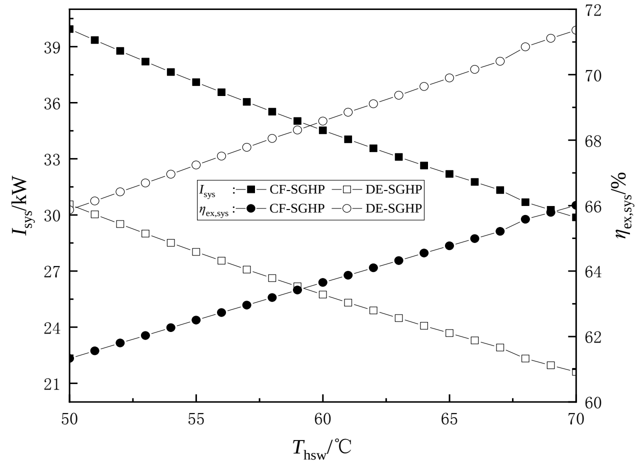


Fig.12. I_{sys} and $\eta_{ex,sys}$ vs. T_{hsw} of the heat pump steam systems

The variations of COP and W_{per} of the two systems with different heat source water temperature are given in Fig.13. The COP of CF-SGHP system increses from 2.25 to 2.69 as the T_{hsw} increased from 50℃ to 70℃. The W_{per} of CF-SGHP system decreses from 328.7kW to 274.1kW as the T_{hsw} increased from 50℃ to 70℃. However, the COP of DE-SGHP system is always higher than that of the CF-SGHP system with an average increase of 9.8%, and the W_{per} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 9.0%, in the range of the the T_{hsw} increasing from 50℃ to 70℃.

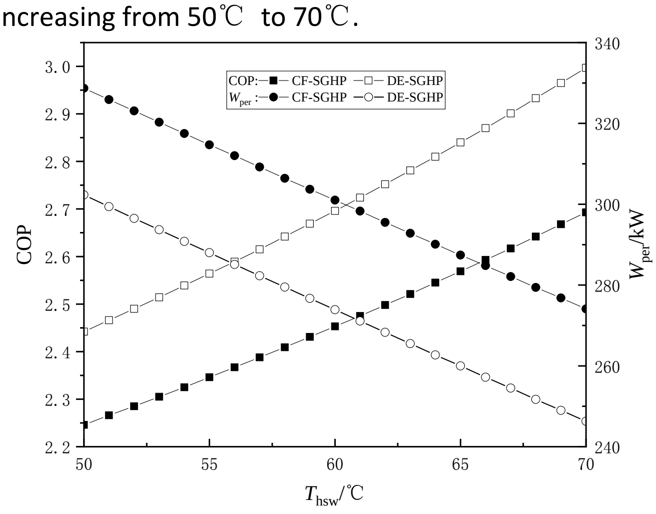


Fig.13. COP and W_{per} vs. T_{hsw} of the heat pump steam systems

6. Conclusion

The exergy loss and thermal performance of the conventional CF-SGHP system was researched in this paper. In order to reduce total exergy loss of the system, a DE-SGHP system was proposed. Then the exergy loss and thermal performance of the two systems was analyzed comparatively under different T_{out}, T_{cond}, T_{hsw}. The conclusions are as follows:

(1) The I_{sys} of the CF-SGHP system is 39.46kW. In addition to the exergy loss caused by the irreversible loss of the compressors, the relative maximum exergy loss occurred in the flash cycle process, whose exergy loss rate is 23.24%.

(2) The $\eta_{ex,sys}$ is 67.74% and the COP is 2.71 of the DE-SGHP system when the T_{out} is 165℃ and T_{hsw} is 60℃.

(3) The I_{sys} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 25.2% and the COP of DE-SGHP is always higher than that of the CF-SGHP with an average increase of 10.6%, as the T_{out} increasing from 150℃ to 165℃.

(4) There existed the optimal T_{cond} to maximum the $\eta_{ex,sys}$ and the COP of the of the CF-SGHP and DE-SGHP systems, the $T_{cond,opt}$ are 102℃ and 91℃, respectively.

(5) The I_{sys} of DE-SGHP system is always lower than that of the CF-SGHP system with an average decrease of 25.4%, and the COP of DE-SGHP is always higher than that of the CF-SGHP with an average increase of 9.8%, as the T_{hsw} increasing from 50℃ to 70℃.

(6) Therefore, it found that the DE-SGHP system has a better exergy efficiency and thermal performance, thus has a broader development prospect.

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