Energy Analysis and Exergy Analysis of a New Type Solar Air Conditioning System

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Abstract: This paper presents a novel type of solar air conditioning system. The steady thermodynamic analysis model and exergy analysis model are given to analyse the performance of the novel system. The energy analysis and exergy analysis are carried out through detailed calculation. The exergy system analysis shows that the highest exergy destruction occurs in condensation-generation and condensation 2, which can be reduced with parameter optimization. A range of operating conditions are selected and simulated for the novel solar air conditioning system by changing the corresponding parameters while keeping other parameters consistent with the design conditions. Compared with the original solar air conditioning system, the novel solar air conditioning system possesses the advantages of low requirements for the performance of solar collectors and smaller area of solar collectors. The novel system is also less affected by weather conditions.

1 INTRODUCTION

To reduce environment pollution and energy conservation, solar air conditioning researches are of great significance. Solar energy is a kind of clean and renewable energy. Since the refrigerating demand and the supply of solar radiation are almost in phase with each other, the solar air conditioning system is appealing to many researchers (Hwang, Y., Radermacher, R., Alili, A. and Kubo, I. 2008; Sekret, R. and Turski, M. 2012). The most common kind of solar air conditioning system, which can be seen as the original system, is composed of solar collectors and the single effect LiBr–H_2O absorption chiller. However, the original system is confronted with many problems in practical operation, including crucial problems like intermittency and instability. In order to solve the existing problems, a number of relevant studies are carried out by many researchers.

Xu presented a new solar powered absorption refrigeration system with advanced energy storage technology. The energy collected from the solar radiation was transformed into the chemical potential of the working fluid and stored in order to solve the problem of the unconformity between solar radiation and cooling demand (Xu, S., Huang, X. and Du, R. 2011). Lass-Seyoum tested a similar closed thermo-chemical heat storage system. This method offers several advantages including the possibility of long-term storage with minimal thermal losses and a high-energy storage density compared with sensible and latent thermal storage principles (Lass-Seyoum, A., Blicker, M., Borozdenko, D., Friedrich, T. and Langhof, T. 2012).

Ahachad combined an absorption heat pump system and an absorption refrigeration system to form a two-stage vapor absorption system. And the tests conducted in Rabat (Morocco) showed that the system can be operated at lower heat source temperatures by using flat-plate collectors (Ahachad, M., Charia, M. and Bernatchou, A. 1993). Jain proposed a cascaded vapour compression–absorption system (CVCAS) which consists of a vapour compression refrigeration system (VCRS) coupled with single effect vapour absorption refrigeration system (VARS). Based on first and second laws, a comparative performance analysis of CVCAS and an independent VCRS has been carried out (Jain, V., Kachhwaha, S. and Sachdeva, G. 2013).

Prasartkaew studied the performance of a renewable energy (solar–biomass) based on single effect LiBr–H_2O absorption chiller. The chiller and overall system coefficient of performances were found to be 0.7 and 0.55 respectively and the biomass (charcoal) consumption for 24h operation was 24.44kg/day (Prasartkaew, B. and Kumar, S. 2010). Y.L. Liu presented the performance prediction of a solar/gas driving double effect LiBr-H_2O absorption

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system. Its high-pressure generator is driven by natural gas. Simulation results illustrated that such kind of system is feasible and economical (Liu, Y. and Wang, R. 2004). Apart from gas boiler, there also exist electric heaters and oil boilers used as auxiliary equipment. For a number of absorption systems that employed heat recovery in the industrial sector or in large residential buildings have the auxiliary of gas or oil, as described in the experimental studies (Pérez de Viñaspre, M., Bourouis, M., Coronas, A., García, A., Soto, V. and Pinazo, J. 2004; Sumathy, K., Huang, Z. and Li, Z. 2002; Ali, A., Noeres, P. and Pollerberg, C. 2008).

In order to reduce the influences of the weather conditions on the solar air conditioning system, a novel solar air conditioning system with heat pump as auxiliary equipment is proposed in this paper. Compared with the auxiliary of gas/oil, the novel system needs fewer additional equipment like boilers and gas/oil tanks. The energy analysis and exergy analysis of the novel system are carried out through detailed calculation to get a better view of the system. A comparison between the novel system and the original system is also considered in this study.

2 SYSTEM DESCRIPTION

Fig. 1 illustrates the main components of the novel solar air conditioning system. The system is mainly composed of three subsystems: solar collecting (SC) subsystem, heat pump (HP) subsystem and absorption refrigeration (AR) subsystem. The SC subsystem, using water as working medium, consists of flat-plate solar collectors and a water pump. The HP subsystem , using R134a as working medium, mainly consists of evaporator 1, condensation 1, expansion valve 1 and compressor. The AR subsystem mainly consists of generator, absorber, evaporator 2, condensation 2, expansion valve 2, a heat exchanger and a solution pump，using lithium bromide solution as working fluid pair. For reducing the component number of the system, condensation 1 and generation can be integrated as one compact component, condensation-generation, which can also help the heat source keep stable.

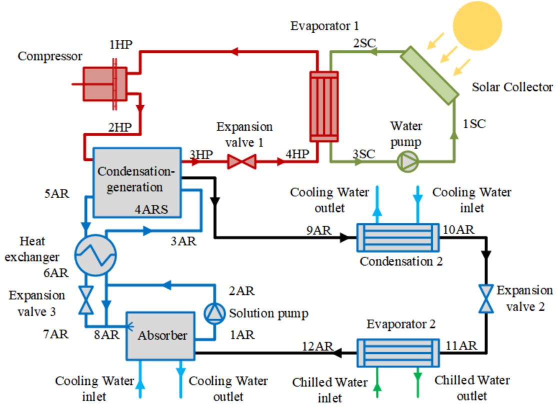

Figure 1: The schematic diagram of a new type of solar air conditioning system.

The working flow of the system is shown as follows.

(1)The SC subsystem receives low-grade energy through gathering solar radiation to warm the working medium water. Then, the low-grade energy was transferred to HP subsystem in evaporator 1.

(2)In the HP subsystem, the working medium R134a is cycled among the components, transforming the energy from evaporator 1 to condensation-generation, which also means that the low-grade energy is transformed to high-grade energy to ensure the normal operation of AR subsystem.

(3)In the AR subsystem, the high-grade energy from the HP subsystem is used to produce chilled water in evaporation 2 for cooling application.

3 MODELS OF THERMODYNAMIC ANALYSIS

In order to simplify the calculation of the performance analysis, a set of assumptions are made as follows.

(1) The analysis is made under steady conditions.

(2) The pressure losses in the pipelines and all components are negligible.

(3) The heat transfer between all components and surroundings is ignored.

(4) The energy consumed by the solution pump and the water pump is negligible.

(5) The lithium bromide aqueous solution at the outlet of absorber and generator is assumed to be in equilibrium at their respective temperatures and pressures.

(6) The refrigerant (water) at the outlet of evaporator 1, condenser 1, evaporator 2 and condenser 2 is saturated.

The efficiency expression of the solar collectors is determined as indicated in Eq. (1). The optical efficiency is 0.67 and the linear coefficient of thermal losses is 5.52 W/°C/m²

$$\eta_{SC} = 0.67 - 5.52\frac{(T_{SC} - T_{am})}{I_{SC}} \tag{1}$$

where I_{SC} is the solar intensity. The terms T_{am} and T_{SC} are the environment temperature and the average temperature of working medium in collectors separately.

The governing equations of mass conservation and species conservation can be expressed as

$$\sum m_i - \sum m_o = 0 \tag{2}$$

$$\sum (mx)_i \text{-} \sum (mx)_o = 0 \tag{3}$$

where m is the mass flow rate and x is mass concentration of LiBr in the solution.

The energy balance of each component based on the first thermodynamic for a steady state can be expressed as

$$\sum (mh)_i \text{-} \sum (mh)_o + \left(\sum Q_i - \sum Q_o\right) + W = 0 \tag{4}$$

The exergy with reference to the environment can be expressed as

$$e = (h - h_0) - T_0 (s - s_0) \tag{5}$$

where h_0 and s_0 are evaluated at the reference environment.

Exergy destruction of a control volume based on the second thermodynamic can be calculated as follow,

$$\begin{aligned} \Delta E_x &= \sum m_i e_i - \sum m_o e_o + \sum W \\ &\text{-}\left[\sum Q\left(1 - \frac{T_0}{T}\right)_i - \sum Q\left(1 - \frac{T_0}{T}\right)_o\right] \end{aligned} \tag{6}$$

The cooling COP of the novel solar air conditioning system can be defined as follow,

$$COP_{SYS} = \frac{m_{AR}(h_{12AR} - h_{10AR})}{W_{HP}} \tag{7}$$

where W_{HP} is the electric energy consumption of the HP subsystem; m_{AR} is the mass flow rate of refrigerant vapour (water) of the AR subsystem.

The exergy efficiency of the novel solar air conditioning system can be defined as

$$ECOP_{SYS} = \frac{m_{CW}\left(e_{x,CW2} - e_{x,CW1}\right)}{m_{SC}\left(e_{x,2SC} - e_{x,1SC}\right) + W_{HP}} \tag{8}$$

where m_{SC} is the mass flow rate of SC subsystem; m_{CW} is the mass flow rate of the chilled water in evaporator 2.

Based on the properties of the water, R134a and lithium bromide aqueous solution, the equations above are determined to describe the mass transfer, heat transfer and exergy destruction in every component.

4 RESULTS ANALYSIS UNDER DESIGN CONDITIONS

To analyse the performance of system quantitatively, the design operating conditions are selected as follows.

(1) Flat-plate collectors are selected, with an area of 100 m². The inlet temperature of the collectors is maintained at T_{1SC}=70 ℃ by using variable speed pump to regulate water mass flow rate.

(2) The condensation temperature of HP subsystem is maintained at $T_{C,HP}$=90 ℃, which means the heat source temperature of AR subsystem $T_{HS,AR}$= $T_{C,HP}$=90 ℃.

(3) The efficiency of the compressor is 0.63.

(4) The condensation temperature of AR subsystem is maintained at $T_{C,AR}$=40 ℃, the evaporation temperature of AR subsystem is maintained at $T_{E,AR}$ = 10 ℃.

(5) The inlet temperature of cooling water is assumed at T_{W0}=30 ℃. The outlet temperature of cooling water to condensation 2 and absorber is assumed at T_{WC}=T_{WA}=T_{W0}+3 ℃.

(6) The inlet temperature and outlet temperature of the chilled water in evaporator 2 are respectively maintained at T_{CW1}=17 ℃, T_{CW2}=13 ℃.

(7) The heat exchanger effectiveness is 0.7.

Table 1: Thermodynamic properties of each point in the system.

Point	temperature	pressure	concentration	specific enthalpy	specific entropy	mass flow rate
	℃	kPa	wt%	kJ/kg	kJ/(kg·K)	kg/s
1SC	70.00	-	-	293.02	0.96	0.6
2SC	85.58	-	-	358.38	1.14	0.6
3SC	85.58	-	-	358.38	1.14	0.6
1HP	72.79	2252.2	-	428.93	1.69	0.456
2HP	95.56	3244.2	-	438.97	1.70	0.456
3HP	90.00	3244.2	-	342.93	1.44	0.456
4HP	72.79	2252.2	-	342.93	1.45	0.456
1AR	36.00	-	52.62	79.28	0.24	0.459
2AR	36.00	-	52.62	79.28	0.24	0.167
3AR	63.34	-	52.62	137.19	0.42	0.167
4AR	69.86	-	52.62	151.23	0.46	-
5AR	80.00	-	57.68	184.89	0.47	0.152
6AR	48.13	-	57.68	121.42	0.28	0.152
7AR	48.13	-	57.68	121.42	0.28	0.152
8AR	40.67	-	54.35	93.71	0.25	0.444
9AR	74.93	7.38	-	2640.13	8.46	0.015
10AR	40.00	7.38	-	167.54	0.57	0.015
11AR	10.00	1.23	-	167.54	0.59	0.015
12AR	10.00	1.23	-	2519.23	8.90	0.015

Table 2: Exergy destruction of system components.

	Input exergy	Output exergy	Exergy destruction	Exergy destruction ratio
	kJ	kJ	kJ	%
Evaporator 1	-33.39	-34.02	0.63	7.17
Compressor	-36.68	-38.20	1.52	17.21
Expansion valve 1	-48.26	-49.06	0.80	9.12
Condensation-generation	-34.30	-36.35	2.05	23.28
Absorber	10.69	9.57	1.12	12.76
Condensation 2	2.58	1.29	1.28	14.59
Evaporator 2	0.84	0.04	0.80	9.09
Expansion valve 2	0.03	-0.10	0.12	1.39
Heat exchanger	12.55	12.07	0.48	5.40
total	-	-	8.80	-

By using the aforementioned equations, the thermodynamic parameters of the system under the design operating conditions can be calculated.

The effect of the chemical exergy of solution is neglected in most relevant studies, which is also neglected in this paper. Reference temperature and pressure here is set to be T_0= 25 ℃ and p_0=101.325 kPa for the exergy analysis. The calculation results of the exergy analysis of various components are presented in Table 2. As can be deduced from Table 2, among all the components, condensation-generation has the highest exergy destruction，which accounts for 23.28 % of the total amount of exergy destruction. The condensation-generation requires a great amount of heat to produce refrigerant vapour (water) from lithium bromide aqueous solution. In addition, the refrigerant vapour (water) leaves the condensation-generation for condensation 2 overheated, which constitutes a thermodynamic loss in the condensation-generation and leads to extra cooling requirement in condensation 2. Therefore, the condensation 2 has the second highest exergy destruction. The exergy efficiency can be improved with parameter optimization.

5 PERFORMANCE ANALYSIS

5.1 The influence of SC subsystem on system

The function of SC subsystem is to provide energy to drive the solar air conditioning system. In this section, the changes in the energy and exergy provided by SC subsystem are investigated with increasing the water mass flow rate of the solar collectors (m_{SC}) from with 0.2 kg/s to 1.0 kg/s at different inlet temperatures (T_{1SC}).

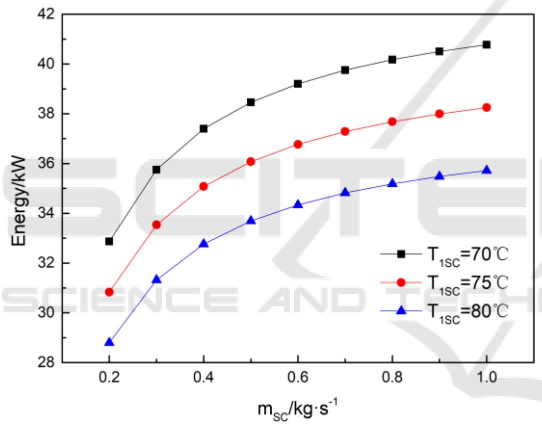

Figure 2: The curves between energy provided by SC subsystem and its mass flow mate.

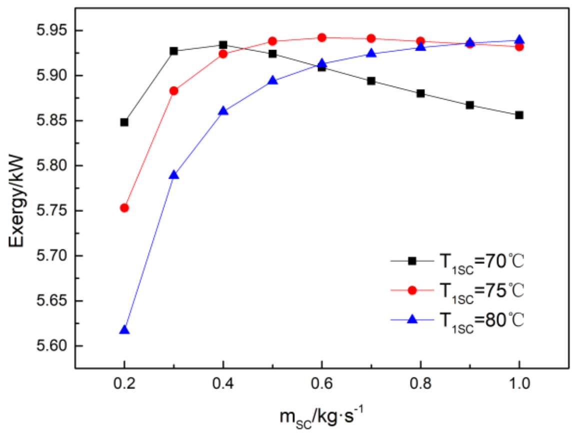

Figure 3: The curves between exergy provided by SC subsystem and its mass flow mate.

Fig. 2 shows the combined effect of m_{SC} and T_{1SC} on the energy output of the SC subsystem. The energy provided by the SC subsystem increases with the increasing of water mass flow rate, and then converges to a constant value. In order to get more energy from the solar intensity, it is better to select a lager mass flow rate. Considering that the water pump will consume more power with the mass flow rate increasing in practical work, an overall consideration should be given to select a medium value of m_{SC}. It is also can be seen that the SC subsystem provide less energy with the increase of T_{1SC}, which means the efficiency of solar collectors decreases with the increase of T_{1SC}.

It is observed in Fig. 3， the effect of m_{SC} on the exergy output of SC subsystem does not produce the same behaviour as it does on the energy output of the SC subsystem. And there exists a value of m_{SC} which makes the exergy provided by the SC subsystem maximum. It also differs from the trends in Fig. 2 that the maximum exergy provided by the SC subsystem under different values of T_{1SC} differs slightly.

5.2 The influence of HP subsystem on system

Working as auxiliary heating equipment, the function of the HP subsystem is to guarantee the temperature of the heat source of the AR subsystem. In this section, the influence of the condensation temperature of HP subsystem ($T_{C,HP}$) on the system performance is investigated, at different condensation temperatures of AR subsystem ($T_{C,AR}$). $T_{C,HP}$ is considered to vary from 83 ℃ to 95 ℃, while $T_{C,AR}$ is set as 38 ℃, 40 ℃ and 42 ℃ respectively.

Fig. 4 shows the variation of the refrigerating capacity of the novel solar conditioning system with the condensation temperature of SC subsystem ($T_{C,HP}$). As can be seen from Fig. 4, the refrigerating capacity of the novel solar conditioning system increases when the condensation temperature of the HP subsystem ($T_{C,HP}$) increases. This is mainly caused by the reasons below. (1) The condensation temperature of the HP subsystem ($T_{C,HP}$) affects the AR subsystem as the temperature of heat source. As we know the refrigerating capacity of the absorption refrigeration system increases with the increase of its heat source temperature. (2) The amount of energy from the SC subsystem is also amplified by the HP subsystem.

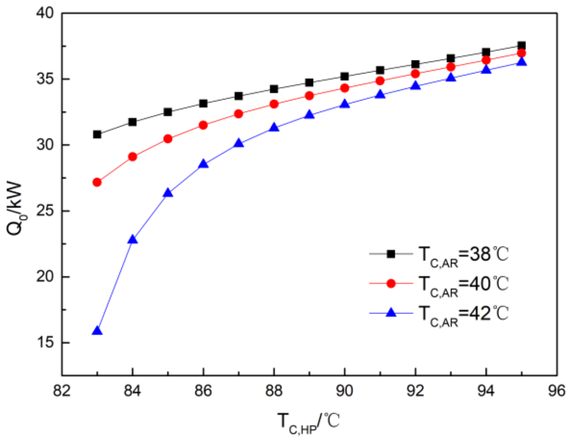

Figure 4: The curves between refrigerating capacity and $T_{C,HP}$.

According to the definition of in Eq. (7), COP_{YSY} can intuitively reflect the electric power consumption of the novel solar air conditioning system, instead of reflecting the solar energy consumption. Fig. 5 shows the variation of COP_{SYS} with the condensation temperature of SC subsystem ($T_{C,HP}$). As can be seen from Fig. 4, with $T_{C,HP}$ increasing, the electric power consumption of the system also increases to transform the low-level energy to a higher level, which leads to a decline in COP_{SYS}. Although high COP_{SYS} provides a better operating condition for the AR subsystem, the electric power consumption caused by the HP subsystem increases in a faster speed, which ultimately leads to the decline of COP_{SYS}. When $T_{C,HP}$ is set as 90 ℃, COP_{SYS} is more than 7.5, which is much higher than the normal vapour compression refrigeration systems.

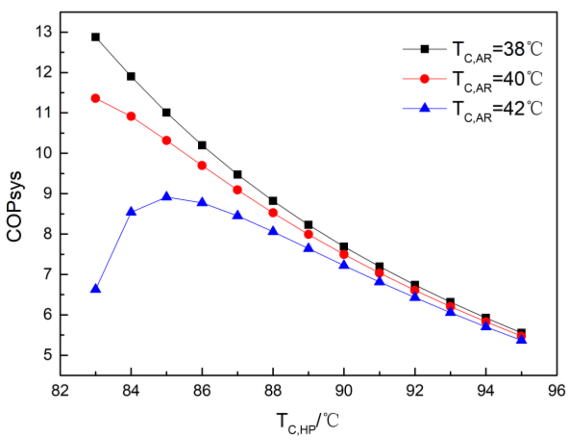

Figure 5: The curves between COP_{SYS} and $T_{C,HP}$.

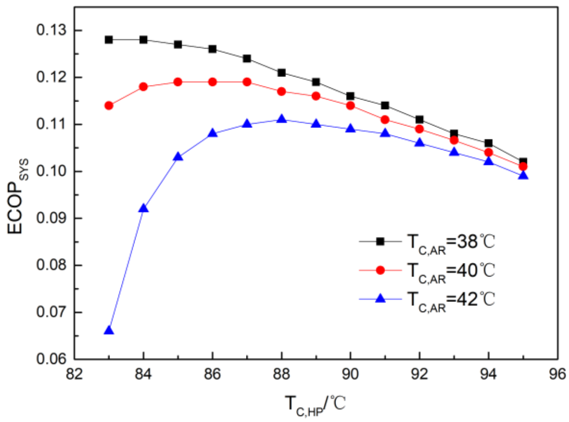

Figure 6: The curves between $ECOP_{SYS}$ and $T_{C,HP}$.

The novel solar air conditioning system adds the heat pump as auxiliary equipment, which consumes a certain amount of electric energy to ensure the normal operation of the AR subsystem. Since the solar energy and electric energy differ in quality, for the purpose of getting a better view of the energy consumption of the system, exergy analysis is carried out, which takes the quality of energy into consideration. The exergy efficiency of the novel system is defined as Eq. (8). According to Eq. (8), $ECOP_{SYS}$ can indicate the exergy consumption of the novel system. As can be seen from Fig. 6, with $T_{C,HP}$ increasing, the exergy efficiency of the novel system increases at first and then decreases slightly. In a certain range of $T_{C,HP}$, the consumption of electric power brings a better use of the exergy. When $T_{C,HP}$ is higher than a certain value, more input exergy is supplied to the system and more exergy losses occur.

Figs. 4-6 can be used to select design parameters when the systematic design is carried out. To prevent the risk of crystallization and to get a medium COP_{SYS}, $T_{C,HP}$ should not be too high. However, $T_{C,HP}$ should be as high as possible in order to get a larger refrigerating capacity of the novel system. The selection of parameter needs overall considerations.

5.3 The comparison of novel and original system

In this section, the performance of the novel system is compared with the original system through quantitative calculation. The original system here is composed of solar collectors and a single-effect LiBr absorption refrigeration system, with no auxiliary heating equipment. The calculation conditions include two solar radiation intensities, the low solar radiation $I_{L,SC}$=0.4 kW/m^2 and the high solar radiation $I_{H,SC}$=0.8 kW/m^2.

The refrigerating capacity (Q_0) and the area of solar collectors corresponding to unit refrigerating capacity (A_s) are selected as standards of the comparison between the novel system and the original system. The value of A_s can reflect the initial investment of the system to a certain extent.

The results of comparison between the novel system and the original system under different operating conditions are shown in Table 3. As seen in Table 3, when the solar collectors work at a lower temperature (T_{1SC}=70 ℃) the novel system produces 34.32 kW and 9.00 kW refrigerating capacity respectively in high solar radiation and low solar radiation, while the original system fails to give normal operation. When the solar collectors work at a higher temperature (T_{1SC}=80 ℃)， the refrigerating capacity of the new system is larger than the original system. Through the comparison between two systems, there is a conclusion that the novel system are less affected by the weather condition. The novel system has lower requirements for the performance of solar collectors than the original system, which makes the use of flat-plate collectors reasonable.

It also can be concluded from Table 3 that the novel system needs much smaller area of solar collectors to produce the same refrigerating capacity. With the advantages of low requirements for the performance of solar collectors and smaller area of solar collectors, the initial investment of the novel system is positive compared with the original system.

Table 3: Performance comparison of new solar air conditioning system and original solar air conditioning system.

	T_{1SC}=70 ℃		T_{1SC}=80 ℃	
	$I_{H,SC}$=0.8 kW/m^2	$I_{L,SC}$=0.4 kW/m^2	$I_{H,SC}$=0.8 kW/m^2	$I_{L,SC}$=0.4 kW/m^2
Novel system Q_0 (kW)	34.32	9.00	28.31	4.28
Original system Q_0 (kW)	-	-	22.13	3.22
Novel system A_s (m^2/kW)	2.91	11.11	3.53	23.36
Original system A_s (m^2/kW)	-	-	4.52	31.06

It should be mentioned that the numerical calculating results could be different if the operating conditions change. However, the conclusion of the advantages of the novel system will not be overthrown.

6 CONCLUSIONS

In this study, for the analysis of a novel solar air conditioning system with a heat pump as auxiliary equipment, first law of thermodynamics and second law of thermodynamics are applied in every component. The main results obtained are concluded below:

(1) The exergy analysis of the system shows that the highest exergy destruction occurs in condensation-generation and condensation 2. The exergy destruction can be reduced with parameter optimization.

(2) With the increase of $T_{C,HP}$, the refrigerating capacity of the novel system decreases, while the COP_{SYS} increases roughly. There is a value of $T_{C,HP}$ making the $ECOP_{SYS}$ maximum. When $T_{C,HP}$=90 ℃, COP_{SYS} is more than 7.5.

(3) Compared with the original solar air conditioning system, the novel solar air conditioning system enjoys advantages of low requirements for the performance of solar collectors and smaller area of solar collectors.

Additionally, the results of the exergy analysis presented in this paper can also be used in thermoeconomic optimization of the novel solar air conditioning system in future researches.

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