



Performance Improvement of KCS (Kalina Cycle System) 34 by Replacing Throttle Valve With Single-Screw Expander

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In order to recover the energy loss due to the throttling in the path of ammonia-lean solution in the Kalina Cycle System (KCS) 34, two redesigned cycles, in which single-screw expanders that can perform two-phase expansion are used to replace the throttle valve, are proposed in this paper. The results show that the thermal efficiency and net work of two redesigned cycles are higher than those of the original KCS 34. With the concentration increase of ammonia-water mixture, the work produced by the single-screw expander B in two redesigned cycles gradually decreases, and the difference between the work produced in two redesigned cycles also gradually decreases. The original KCS 34 and two redesigned cycles have high exergy efficiency. The highest cycle exergy efficiency of 56.59% can be obtained in the II-redesigned cycle when the evaporation pressure is 3.0 MPa and ammonia-water concentration is 0.75.

Keywords: kalina cycle system (KCS)34, single-screw expander, throttle valve, thermodynamic performance, ammonia-water concentration

INTRODUCTION

Rapid urbanization and the rapid increase of the world's population have brought huge challenges to the global energy. In order to meet the challenges, the development and application of waste heat recovery technology has become particularly important. Organic Rankine Cycle (ORC) and Kalina Cycle (KC) have attracted wide attention in the field of medium and low-grade waste heat recover (Loni et al., 2020; Gholamian and Zare, 2016; Júnior et al., 2019).

In the 1980s, Alexander I. Kalina proposed a power cycle system using ammonia-water mixture as working fluid to utilize low-grade heat energy, and named it the Kalina cycle (Kalina, 1982; Kalina, 1983; Kalina, 1984). Different kinds of low-grade heat sources can be used in the Kalina cycle (Prananto et al., 2018; Wang J. et al., 2013; Cao, Wang and Dai, 2014; Khankari, Munda and Karmakar, 2016). The Kalina cycle can actually be regarded as an improved Rankine cycle. When the ammonia-water mixture evaporates, the ammonia with the lower boiling point evaporates first, and then the water evaporates. Therefore, the evaporation process of the ammonia-water mixture matches well with the temperature distribution of the heat source. The organic Rankine cycle using pure working fluid is in a state of constant temperature and constant pressure during the evaporation process, which does not match the temperature distribution of the heat source, thus resulting in a large exergy loss.

Many researches have been conducted for the optimization and analysis of the parameters in the Kalina cycle system and its working fluids. Marston (1990) developed an entire cycle model for

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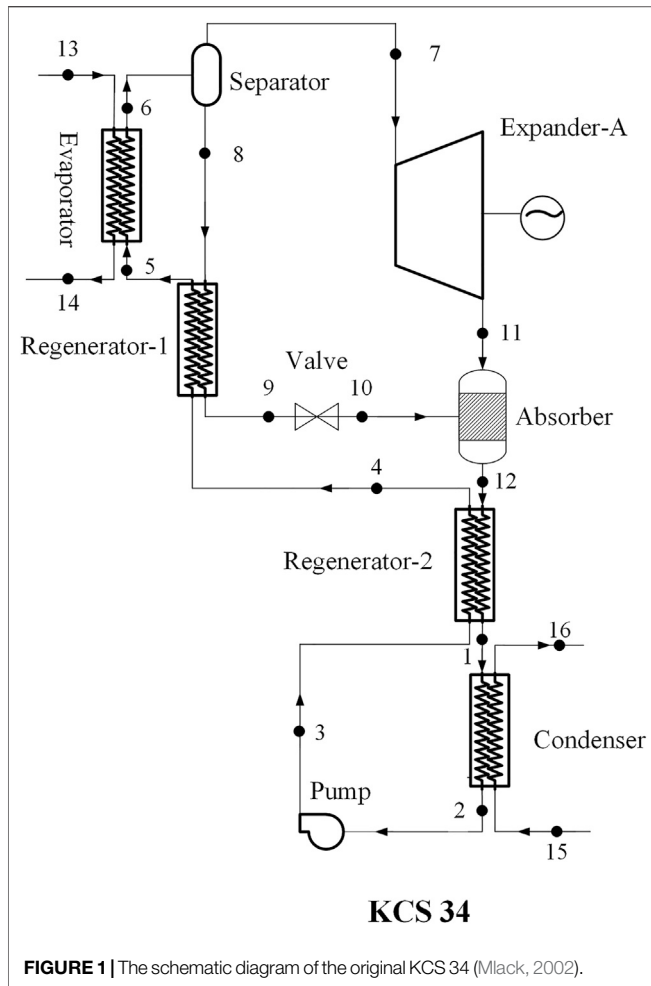
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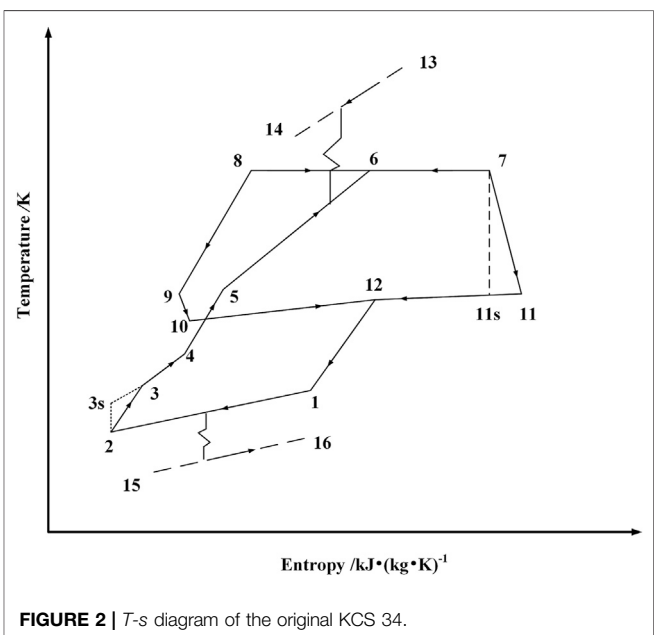
parameter optimization of the Kalina cycle by using the data compiled by predecessors. Wall et al. (1989) used an energy utilization diagram to analyze a 3 MW Kalina bottom cycle, and found that it performed more efficiently than Rankine cycle. Based on Kalina cycle system (KCS) 11, Singh and Kaushik (2013) proposed a combined cycle and found that turbine inlet pressure and ammonia fraction were the key parameters for improving the cycle efficiency. Wang et al. (2017) studied the variation of condensation pressure with ambient temperature under different ammonia-water concentrations and found that a better annual average thermal efficiency can be obtained by using sliding condensation pressure. Eller et al. (2017) investigated alternative working fluids of the Kalina cycle and found that using alcohol/alcohol mixture could improve the second law efficiency of the Kalina cycle.

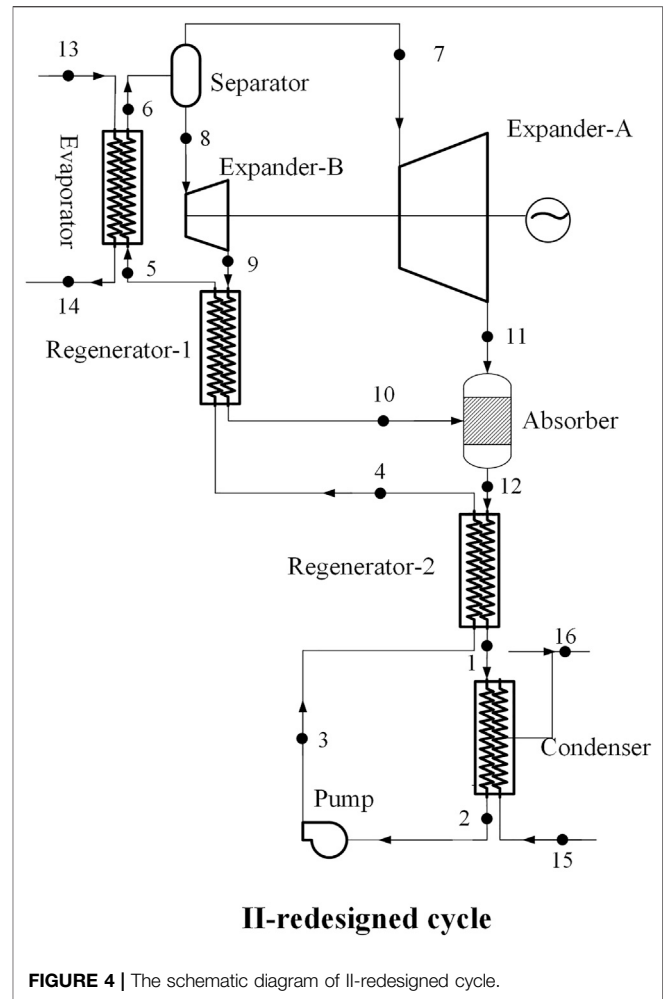
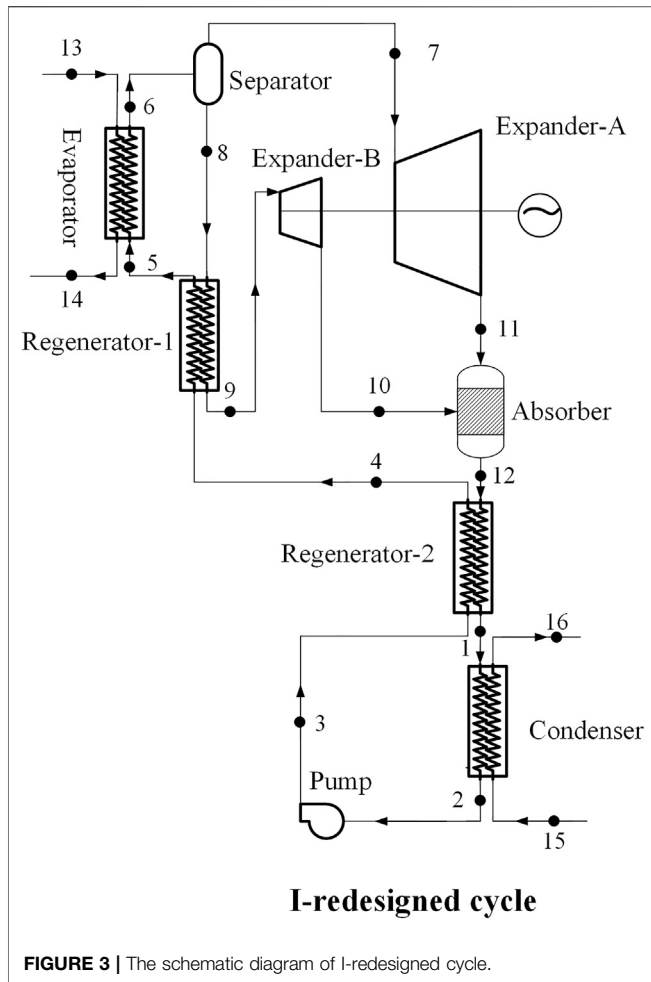
Many researchers have also studied the combination of the Kalina cycle and other thermodynamic cycles. A combined cycle proposed by Zheng et al. (2006) has an overall thermal efficiency of 24.2% and an exergy efficiency of 37.3%. He et al. (2011) combined the organic Rankine cycle (ORC) with the Kalina cycle for recovering the waste heat of internal combustion engine. Murugan and Subbarao (2008) conducted a thermodynamic

analysis on a Rankine-Kalina combined cycle (RKC) and found that the cycle has higher output and higher thermal efficiency than a steam Rankine cycle. Modification of Kalina cycle system is also an important way to improve its performance. Sadeghi et al. (2015) proposed a modified Kalina cycle and optimized its thermal efficiency.

All of the above studies were aimed at improving the thermodynamic performance of the Kalina cycle. In the Kalina cycle, the energy loss due to the throttling in the path of ammonia-lean solution is often ignored. Therefore, it is of great significance to recover this energy loss for improving the performance of the Kalina cycle. Li et al. (2013) used an ejector to replace the throttle valve and the absorber in KCS 11. The results showed that the performance of the modified cycle were better than that of the KCS 11.

In order to recover the energy loss due to the throttling in the path of ammonia-lean solution, it is necessary to select an expander that can perform two-phase expansion. Single-screw expander (SSE) can be a good choice. The single screw structure was invented in 1960. Compared with the traditional steam turbine and gas turbine, the SSE can operate under low power conditions. High-pressure gas, superheated steam, saturated steam, gas-liquid two-phase fluid, and heat fluid can all be used as the working fluid in SSE (Wang et al., 2011). The key laboratory where the authors work has conducted a lot of theoretical and experimental researches on the structure of single screw, including the influence of intake pressure (He et al., 2013), gap adjustment (Wang W. et al., 2013), and rotational speed (Li et al., 2018) on the performance of SSE, working fluid selection for ORC using SSE (Zhang et al., 2019), and the performance analysis of an SSE integrated into an ORC (Lei et al., 2016; Wu et al., 2019). In this paper, based on the feasibility of SSE technology, two redesigned KCS 34 systems in which SSE are used to replace the throttle valves, are proposed for





recovering the energy loss due to throttling in the path of ammonia-lean solution. The performance of two proposed redesigned cycles is compared with the original KCS 34.

THERMODYNAMIC MODEL AND SYSTEM ANALYSIS

System Description

Two redesigned cycles with different placement of SSE, namely the I-redesigned cycle and the II-redesigned cycle, are proposed in this paper. Three cycles studied in this paper are illustrated as follows.

Original KCS 34 with a throttle valve.

I-redesigned cycle: The redesigned KCS 34 with a SSE. The SSE replaces the throttle valve and is placed between the absorber and the regenerator -1.

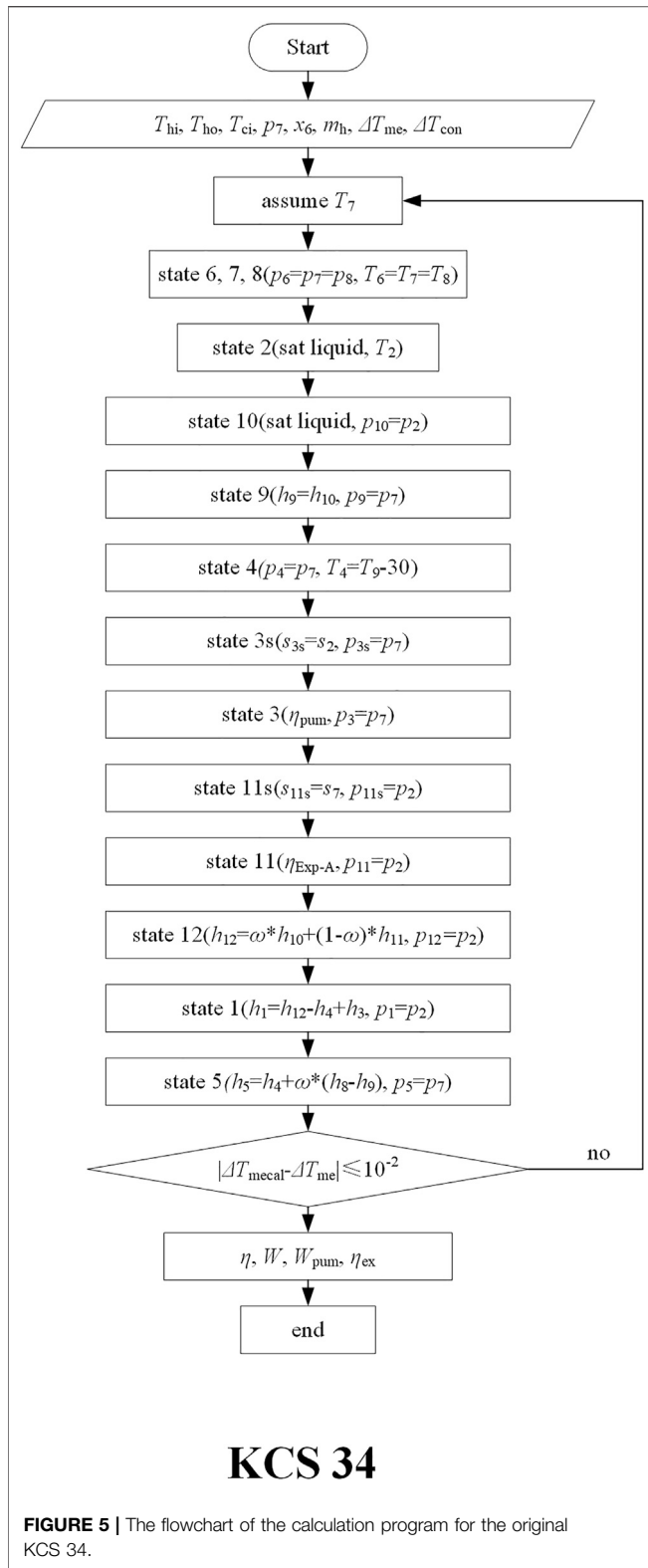
II-redesigned cycle: The redesigned KCS 34 with an SSE. The SSE is placed between the regenerator -1 and the gas-liquid separator. The throttle valve in the original KCS 34 is removed.

The schematic diagram of the original KCS 34 with a throttle valve is depicted in **Figure 1**, and its *T-s* diagram is shown in **Figure 2**. The ammonia-lean solution at point 8 is throttled to the

TABLE 1 | Initial conditions used for calculation.

Initial condition	Value
T_{hi}/K	400
T_{ho}/K	353
$m_h/kg \cdot s^{-1}$	1
$m_c/kg \cdot s^{-1}$	1
T_c/K	290
T_o/K	300
$\Delta T_{me}/K$	20
$\Delta T_{con}/K$	2
$\eta_{EXP-A}/\%$	80
$\eta_{EXP-B}/\%$	65
$\eta_{pump}/\%$	80

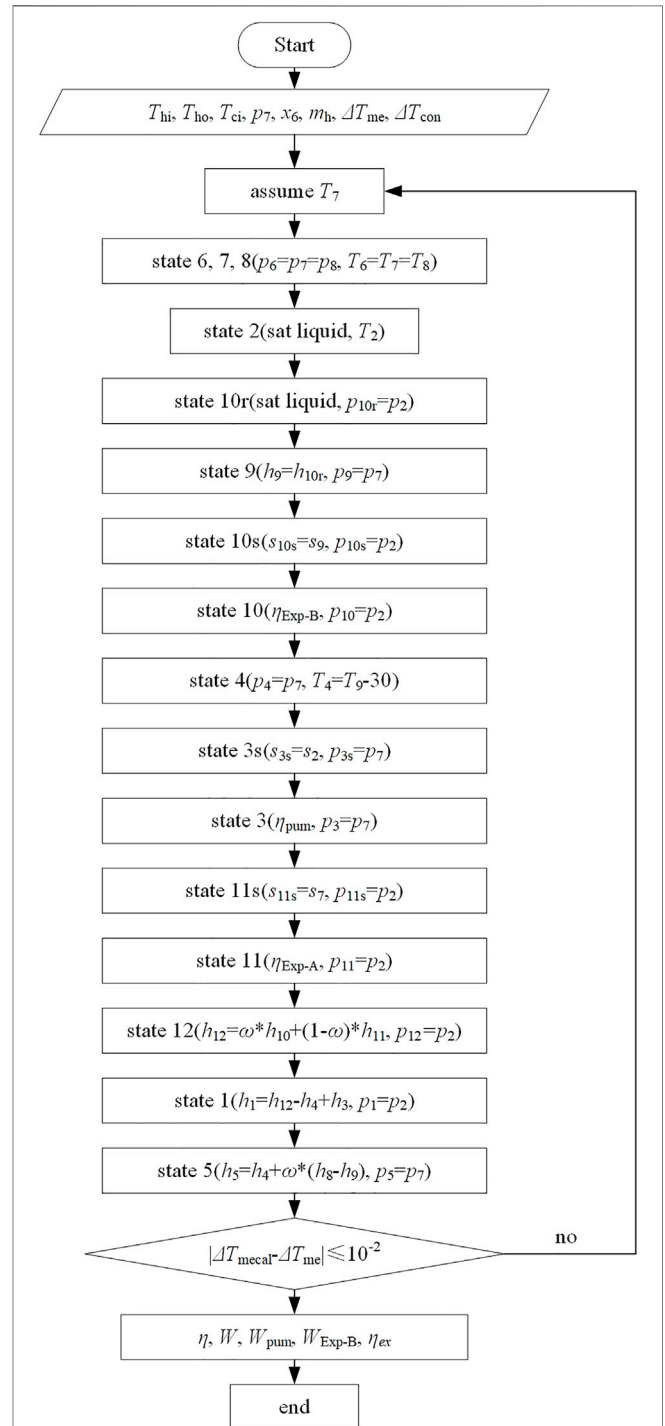
condensation pressure at state point 10 after flowing through the regenerator-1, and then mixed with the ammonia-rich vapor in the absorber to form a working solution with initial ammonia fraction. The working solution successively flows through regenerator-2 and condenser to reach state point 2. After being pressurized by pump, the working solution flows through the regenerator-2 and the regenerator-1, and then flows into the evaporator. After being heated, the working

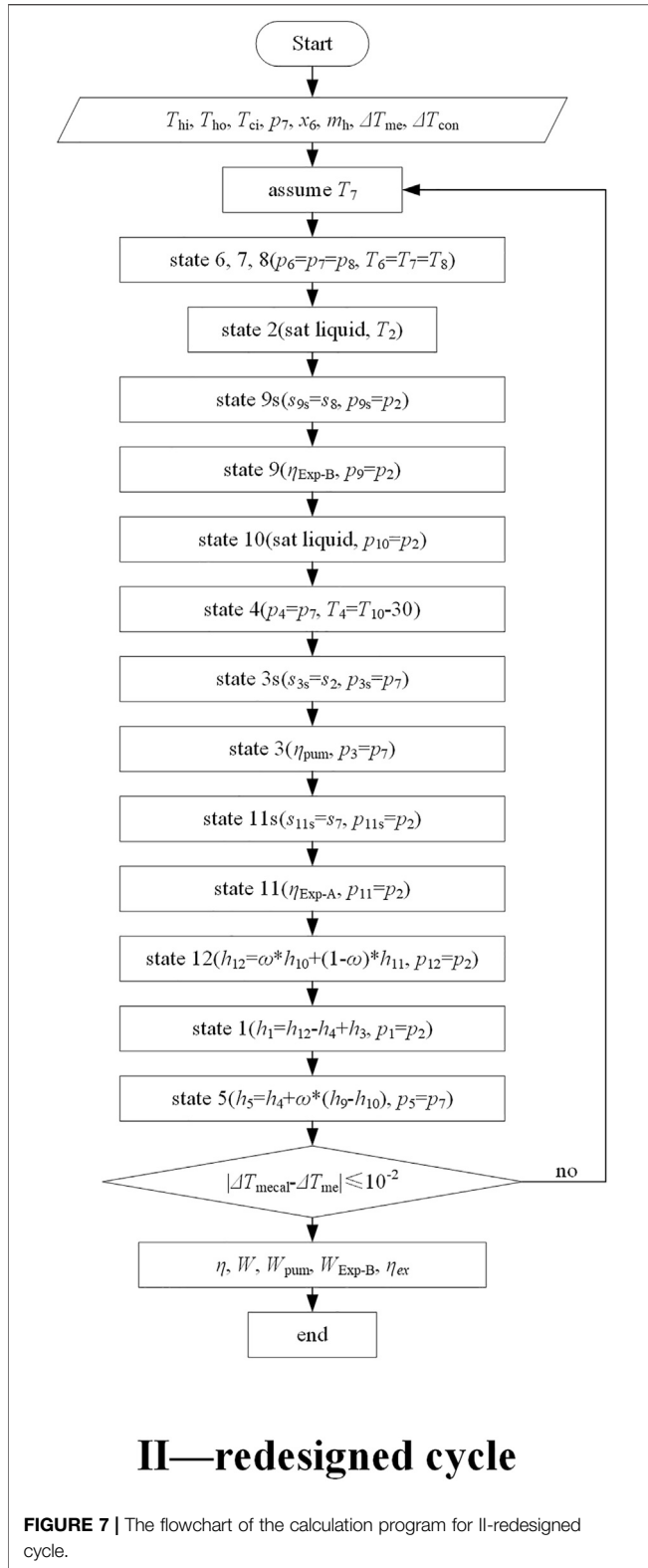


solution at point 6 is sent to separator, in which it is split into ammonia-rich vapor at point 7 and ammonia-lean solution at point 8. Two redesigned cycles based on the original KCS 34 are depicted in Figure 3 and Figure 4.

General Assumptions

In this paper, the EES (Engineering Equation Solver) software is used to calculate the thermophysical properties of ammonia-water. The logarithmic mean temperature difference in the





evaporator is used as the convergence condition in the calculation. Table 1 lists the initial conditions used for calculation. Figure 5, Figure 6, and Figure 7 shows the flowcharts of the calculation programs for the original KCS 34

and its two redesigned cycles. In order to simplify the calculation, the following assumptions are used.

- 1) The system and its components are at steady states.
- 2) Pipeline pressure loss and the energy loss caused by fluid friction in the system are neglected.
- 3) Heat loss in the system is neglected.
- 4) The isentropic efficiency of the SSE in the two redesigned cycles is equal.
- 5) The exergy loss of cooling water is neglected.
- 6) According to engineering experience, the highest pressure in the system is maintained within 3 MPa.

Thermodynamic Analysis

In the cycle, flue gas is used as the heat source, and its inlet and outlet temperatures are T_{13} and T_{14} , respectively. The heat input of the system is

$$Q_{eva} = m_h \cdot c_h \cdot (T_{13} - T_{14}). \quad (1)$$

Heat exchanged in condenser and regenerators is

$$Q_{con} = m_{wf} \cdot (h_1 - h_2). \quad (2)$$

$$Q_{reg-1} = m_{wf} \cdot (h_5 - h_4). \quad (3)$$

$$Q_{reg-2} = m_{wf} \cdot (h_4 - h_3). \quad (4)$$

Work produced by expander A is

$$W_{Exp-A} = m_7 \cdot (h_7 - h_{11}). \quad (5)$$

Work produced by SSE B is

For I—redesigned cycle

$$W_{Exp-B} = m_8 \cdot (h_9 - h_{10}). \quad (6)$$

For II—redesigned cycle

$$W_{Exp-B} = m_8 \cdot (h_8 - h_9). \quad (7)$$

Power consumed by pump is

$$W_{pum} = m_{wf} \cdot (h_3 - h_2). \quad (8)$$

Net work output of system is

$$W = W_{Exp-A} + W_{Exp-B} - W_{pum}. \quad (9)$$

Thermal efficiency is

$$\eta = \frac{W}{Q_{eva}}. \quad (10)$$

Exergy at the inlet of heat source is

$$E_{in} = Q_{eva} \cdot \left(1 - \frac{T_0}{T_{mh}}\right), \quad (11)$$

Where T_{mh} is the average temperature of heat source.

Exergy efficiency of the system is defined as the ratio of the net work of the system to the exergy at the inlet of the heat source,

$$\eta_{ex} = \frac{W}{E_{in}}. \quad (12)$$

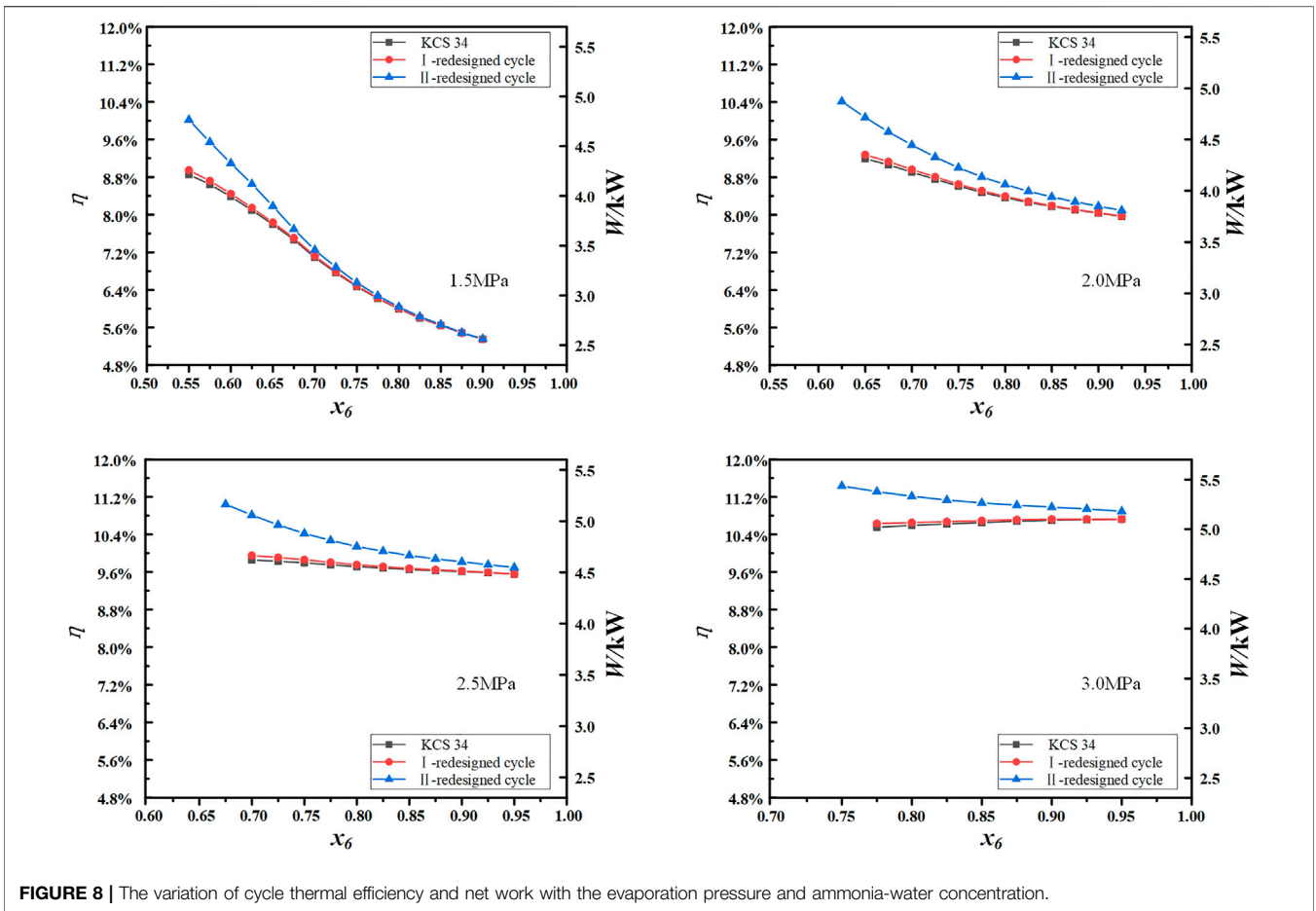


FIGURE 8 | The variation of cycle thermal efficiency and net work with the evaporation pressure and ammonia-water concentration.

The logarithmic mean temperature difference (LMTD) in evaporator is

$$\Delta T_{mecal} = \frac{(T_{13} - T_6) - (T_{14} - T_5)}{\ln T_{13} - T_6 / T_{14} - T_5} \quad (13)$$

RESULTS AND DISCUSSION

When neglecting the pressure drop of the working fluid in absorber, heat exchanger, and pipeline, there are only two pressures in the KCS 34, namely evaporation pressure and condensation pressure. For KCS 34, the condensation pressure is determined by the given cooling conditions and ammonia-water concentration. The inlet and outlet temperatures of the separator (T_6 , T_7 and T_8) are considered equal. Therefore, this paper discusses the influences of the ammonia-water concentration (x_6) and evaporation pressure (p_7) on the cycle thermal efficiency, net work, work produced by SSE B, and system exergy efficiency.

Figure 8 depicts the variation of cycle thermal efficiency and net work with the evaporation pressure and ammonia-water concentration for the original KCS 34 and its two redesigned cycles. It can be seen from **Figure 8** that the thermal efficiency

and net work of the I-redesigned cycle and the II-redesigned cycle are higher than those of the original KCS 34, but the difference between the I-redesigned cycle and the original KCS 34 is small. When the evaporation pressure is 1.5 MPa and the ammonia-water concentration is 0.6, the thermal efficiency of the II-redesigned cycle is 8.5% higher than that of the original KCS 34, while the thermal efficiency of the I-redesigned cycle is only 0.74% higher than that of the original KCS 34. Since the working fluid releases a large amount of energy in the regenerator-1 before entering the SSE B, the cycle thermal efficiency of the I-redesigned cycle is very close to that of the original KCS 34. When the evaporation pressure is low, with the increase of ammonia-water concentration, the cycle thermal efficiency and net work of the three cycles gradually decrease, and the difference among the three cycles also gradually decrease. Therefore, in the case of low pressure, the SSE B does not play a positive role as the ammonia-water concentration increases. When the evaporation pressure is relatively high, the thermal efficiencies of the three cycles are relatively high and the net work large. The decreasing trend of the thermal efficiency and net work of the three cycles gradually slowed down.

Figure 9 depicts the variation of the work produced by SSE B with the evaporation pressure and ammonia-water concentration in the I—and II—redesigned cycles. It can be seen from **Figure 9**

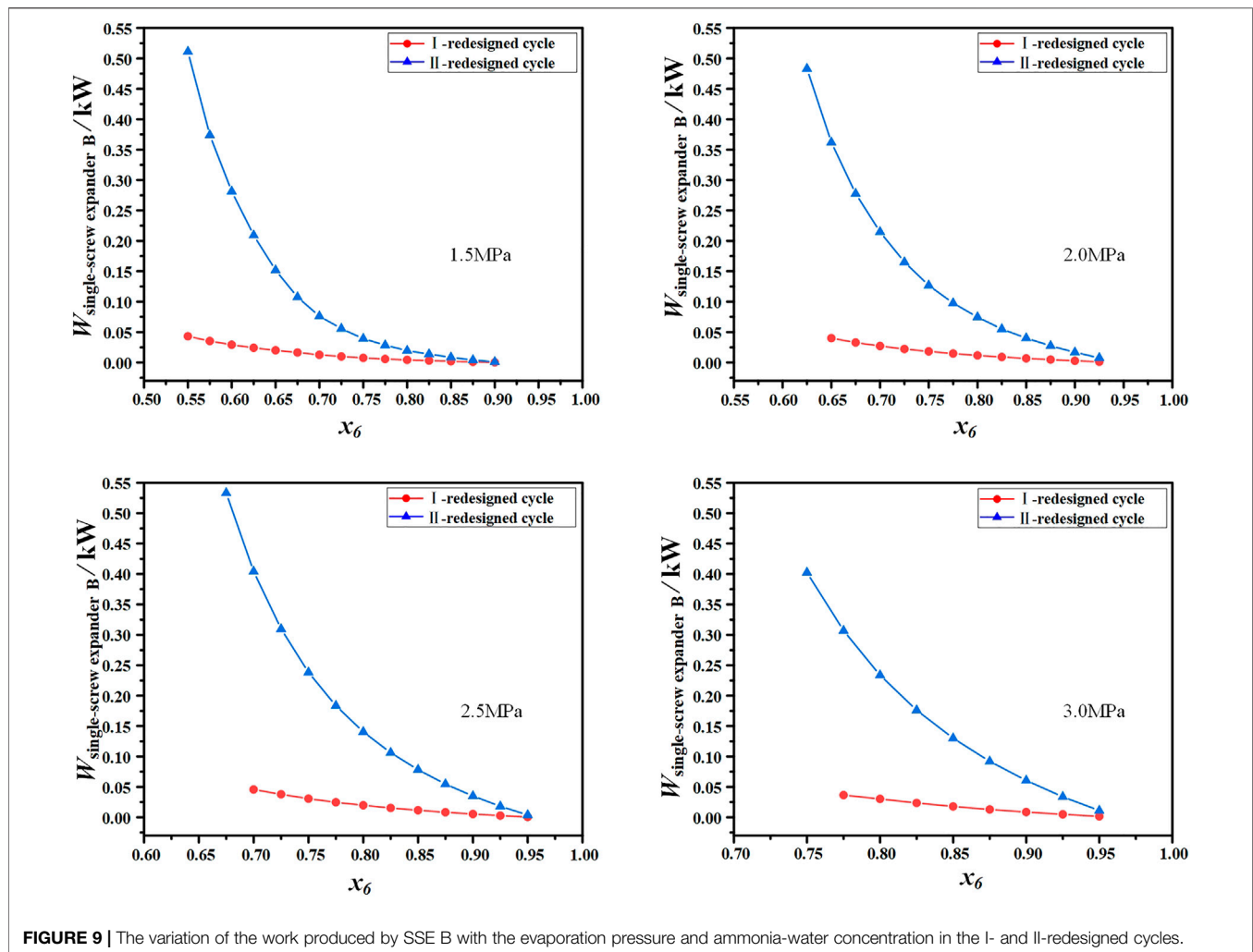


FIGURE 9 | The variation of the work produced by SSE B with the evaporation pressure and ammonia-water concentration in the I- and II-redesigned cycles.

that the work produced by the SSE B in the II-redesigned cycle is higher than that produced by the SSE B in the I-redesigned cycle. When the evaporation pressure is 1.5 MPa and ammonia-water concentration varies from 0.55 to 0.8, the work produced by SSE B in the II-redesigned cycle is 5–12 times that of SSE B in the I-redesigned cycle. With the concentration increase of ammonia-water, the mass flow of working fluid in the path of ammonia-lean solution gradually decreases, so the work produced by the SSE B in the two redesigned cycles gradually decreases, and the difference between the work produced in two redesigned cycles also gradually decreases.

Figure 10 depicts the variation of exergy efficiency of three cycles with the evaporation pressure and ammonia-water concentration. Since the inlet and outlet temperatures and flow rates of the flue gas in three cycles are given and their values are the same, exergy efficiency has a similar variation trend to thermal efficiency and net work. Compared with the low cycle thermal efficiencies of three cycles shown in **Figure 8**, their cycle exergy efficiencies are higher. The highest cycle exergy efficiency of 56.59% can be obtained in the

II-redesigned cycle when the evaporation pressure is 3.0 MPa and ammonia-water concentration is 0.75.

In general, through the above analysis, it can be seen that the performance of the II-redesigned cycle is better than that of the I-redesigned cycle, especially when ammonia-water concentration is moderate. This can be explained by the non-isothermal evaporation characteristics of ammonia-water. **Figure 11** depicts temperature glides of ammonia-water with the ammonia fraction when the evaporation pressures are 2 and 3 MPa, respectively. When the ammonia fraction is moderate, the evaporation temperature glide of ammonia-water is greatest, so it matches the heat source best. While when the ammonia-water concentration is very low or very high, it is closer to pure water or pure ammonia, which leads to a small temperature glide.

CONCLUSION

In order to recover the energy loss due to throttling in the path of the ammonia-lean solution in the Kalina Cycle

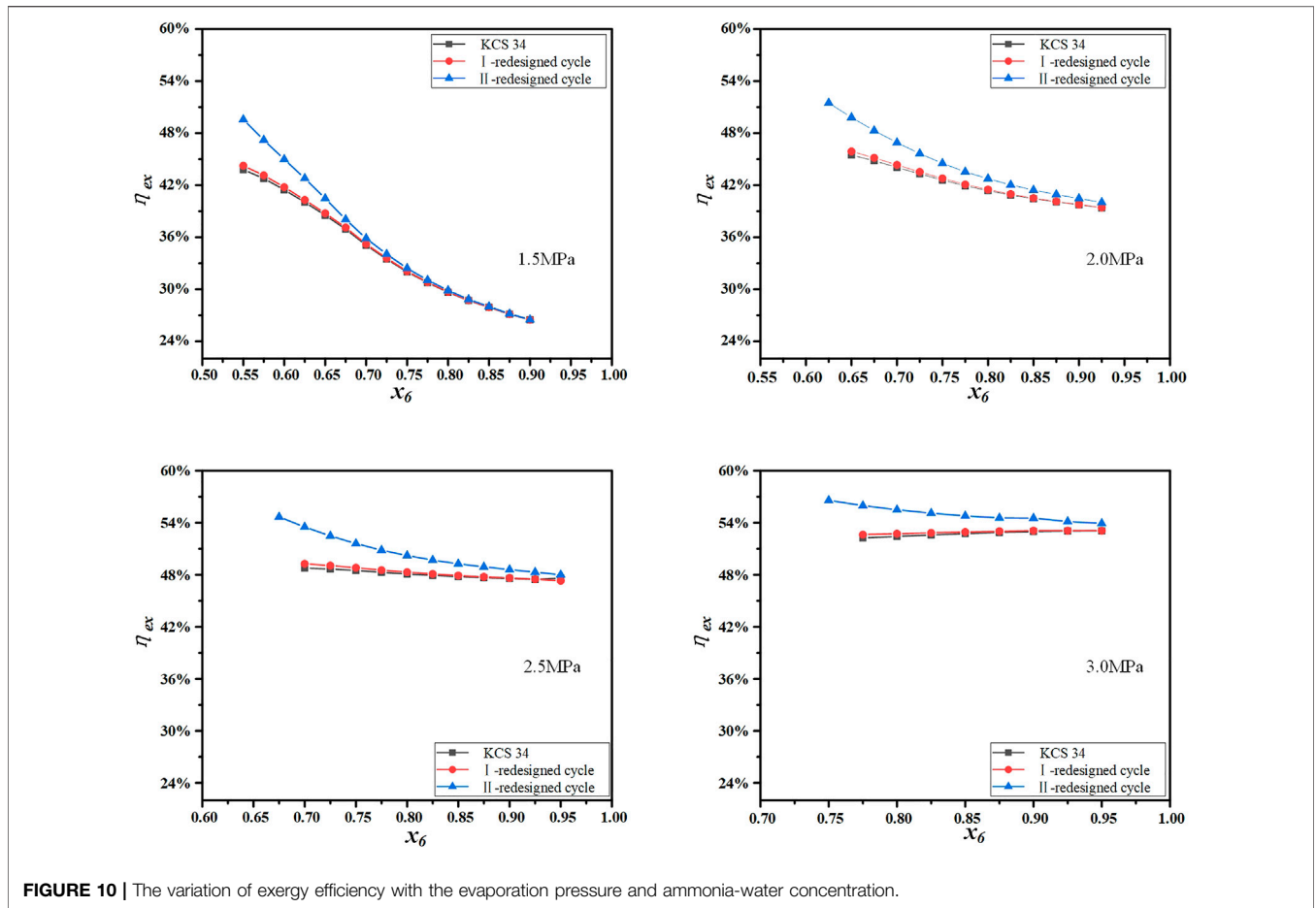


FIGURE 10 | The variation of exergy efficiency with the evaporation pressure and ammonia-water concentration.

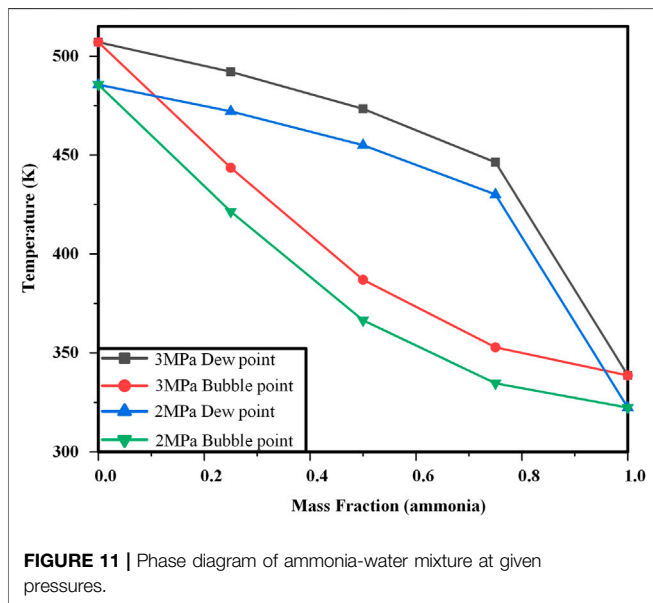


FIGURE 11 | Phase diagram of ammonia-water mixture at given pressures.

System (KCS) 34, two redesigned cycles, namely I-redesigned cycle and II-redesigned cycle, in which SSEs that can perform two-phase expansion are used to replace

the throttle valve, are proposed in this paper. In the I-redesigned cycle, the SSE replaces the throttle valve and is placed between the absorber and the regenerator -1. In the II-redesigned cycle, the SSE is placed between the gas-liquid separator and the regenerator -1. The throttle valve between the absorber and the regenerator-1 in the original KCS 34 is removed.

The thermodynamic performance of two redesigned cycles which have different placement of SSEs is analyzed and compared with the original KCS 34, the following conclusions have been drawn.

- 1) The cycle thermal efficiency and net work of the I-redesigned cycle and the II-redesigned cycle are higher than those of the original KCS 34, but the difference between the I-redesigned cycle and the original KCS 34 is small.
- 2) When the evaporation pressure is small, with the increase of ammonia-water concentration, the cycle thermal efficiency and net work of the three cycles gradually decrease, and the difference among the three cycles also gradually decrease. Therefore, in the case of low pressure, the SSE B does not play a positive role as the ammonia-water concentration increases. When the

evaporation pressure is relatively high, the thermal efficiency of the three cycles are relatively high and the net work large. The decreasing trend of the thermal efficiency and net work of the three cycles gradually slowed down.

- 3) The work produced by the SSE B in the II-redesigned cycle is higher than that produced by the SSE B in the I-redesigned cycle. With the concentration increase of ammonia-water, the mass flow of working fluid in the path of ammonia-lean solution gradually decreases, so the work produced by the SSE B in the two redesigned cycles gradually decreases, and the difference between the work produced in two redesigned cycles also gradually decreases.
- 4) Compared with the low cycle thermal efficiencies of three systems, their cycle exergy efficiencies are higher. The highest cycle exergy efficiency of 56.59% can be obtained in the II-redesigned cycle when the evaporation pressure is 3.0 MPa and ammonia-water concentration is 0.75.

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DATA AVAILABILITY STATEMENT

The original contributions presented in the study are included in the article, further inquiries can be directed to the corresponding author.

AUTHOR CONTRIBUTIONS

XZ and ZL conceptualized the idea. ZL and XZ performed data analysis and formal analysis. XZ and ZL wrote the original draft. YW, CM, and JW obtained resources. XZ and JW reviewed and edited the paper. XZ involved in research project administration and funding acquisition.

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GLOSSARY

Variables

c specific heat, $\text{kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$

E exergy, $\text{kJ}\cdot\text{kg}^{-1}$

h enthalpy, $\text{kJ}\cdot\text{kg}^{-1}$

m mass flow rate, $\text{kg}\cdot\text{s}^{-1}$

Q heat quantity, kW

s entropy, $\text{kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$

T temperature, K

W power, kW

x ammonia-water concentration

ΔT_{con} pinch temperature difference for condenser, K

ΔT_{me} logarithmic mean temperature difference, K

ΔT_{mecal} calculated logarithmic mean temperature difference, K

Abbreviations

KCS Kalina Cycle System

ORC Organic Rankine Cycle

SSE single-screw expander

Greek symbol

η thermal efficiency

η_{ex} exergy efficiency

η_{Exp-A} isentropic efficiency of expander A

η_{Exp-B} isentropic efficiency of expander B

η_{pum} isentropic efficiency of pump

ω concentration of ammonia-water at state point 9

Subscripts

c condenser

ci condenser inlet

con condenser

eva evaporator

h heat source

hi heat source inlet

ho heat source outlet

pum pump

reg-1 regenerator-1

reg-2 regenerator-2

wf working fluid