MODIFICATION OF KALINA CYCLE SYSTEM 34g BY REPLACING THROTTLE VALVE WITH SINGLE-SCREW EXPANDER

by

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Recovery of the energy loss caused by throttling plays an important role in improving the performance of a cycle. Based on the original Kalina cycle system 34g, two redesigned cycles, which have different placement of single-screw expanders used to replace the throttle valve, are proposed in this paper. The thermodynamic performance of two redesigned cycles is analyzed and compared with the original Kalina cycle system 34g. The results show that the thermodynamic performance of each redesigned cycle is better than that of the original Kalina cycle system 34g and the II-redesigned cycle performs best. At a low and moderate evaporation pressure, there is an optimal ammonia-water concentration and it increases with the increase of evaporation pressure. With the concentration increases of ammonia-water, the performance advantage of the redesigned cycle system over the original Kalina cycle system 34g gradually decreases. When the ammonia-water concentration is much lower than the optimal concentration, the single-screw expander produces much work and plays a positive role in net work output of cycle system. The highest cycle exergy efficiency of 54.14% can be obtained in the II-redesigned cycle when the evaporation pressure is 3.0 MPa and ammonia-water concentration is 0.85.

Key words*: Kalina cycle system 34g, single-screw expander, throttle valve, thermodynamic performance, ammonia-water concentration, evaporation pressure.*

Introduction

In order to alleviate global energy crisis and to solve the environmental problems caused by the consumption of traditional energy sources, renewable and sustainable energy must be utilized and low grade waste heat must be recovered for improving overall energy utilization efficiency. A lot of research and development efforts have been made to find a method for renewable and sustainable energy utilization and waste heat recovery. Among all technologies, the organic Rankine cycle (ORC) and the Kalina cycle are two popular ones and extensive research has been conducted on the ORC and Kalina cycle for the past few decades [1-4]. The research hotspot of the ORC mainly includes two aspects: working fluid selection and parameters optimization [5-7]. However, the thermodynamic performance of an ORC with a pure working fluid is low due to its evaporation and condensation at constant temperature. This results in a

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big temperature difference in the evaporator and condenser which in turn increases entropy generation (*i.e.* irreversibility).

In order to overcome this disadvantage, binary mixtures used as working fluid have been under investigation for decades. Cycles using ammonia-water as the working fluid which is a binary mixture offer interesting solutions and a high potential for electricity generation from a low temperature heat source. In early 1980's, Kalina [8, 9] proposed a new series of thermodynamic power cycles in which ammonia-water was used as the working fluid and this series of cycle configurations were named Kalina cycle. The family of Kalina cycle includes Kalina cycle system (KCS) 5 for direct (fuel) fired plants, KCS 6 as the bottoming cycle in combined cycle, KCS 11 for the geothermal energy ranging 121-204 °C (250-400 °F), KCS 34, and KCS 34g for the temperatures below 121 °C (250 °F) [10-13].

Compared with ORC with zeotropic mixture, Kalina cycle can operates in a very wide range of heat source or operation temperatures (90-550 °C). Most of the commonly studied organic compounds would decompose at such high temperatures [14]. After the invention of Kalina cycle, its system performance and optimization have always attracted the attention of researchers. Zhang *et al.* [15] used the Peng-Robinson equation of state to make a thermodynamic analysis on the Kalina cycle based on the First law of thermodynamics and obtained the influence of key parameters on the cycle performance. Saffari *et al.* [16] conducted a thermodynamic analysis on a Kalina cycle used in Husavic geothermal power plant using Artificial Bee Colony algorithm. They found that the optimal thermal and exergy efficiencies were related to the optimal net work output. The pinch point differences in evaporator, recuperators, and condenser were the key constraint parameters. Arslan [17] developed an artificial neural net work model to optimize KCS 34 used for medium temperature geothermal resources and found out the most profitable design.

Literature survey shows that few studies have focused on the energy loss due to the throttling in the path of ammonia-lean solution in the Kalina cycle. Li *et al.* [18] used an ejector to replace the throttle valve and the absorber in KCS 11. The results showed that the performance of the modified cycle was better than that of the KCS 11. In KCS 34g, the ammonia-lean solution with high pressure is throttled to the outlet pressure of expander and mixed with the ammonia-rich vapor in the absorber. Throttling leads to the exergy loss in the path of ammonia-lean solution, resulting in a decrease in system efficiency. Therefore, it is significant to recover the energy loss in the throttling process. Expander is a key device in a thermodynamic cycle. Sometimes the selection of expander will directly determine whether a thermodynamic cycle or process can be successfully realized [19]. Considering that ammonia-water is a binary mixture and the ammonia-rich vapor and ammonia-lean solution in the cycle are both at gas-liquid two-phase state, it is necessary to select an expander that can perform two-phase expansion replace the throttle valve for recovering energy loss due to the throttling in the path of ammonia-lean solution. In all types of expanders, single-screw expander (SSE) can meet all the requirements.

Single-screw structure was first invented in the 1950 and it was initially mainly used in compressors [20]. At present, this structure has been widely used in the design and manufacture of expander. Compared with other expanders, SSE has many unique advantages, such as balanced load of the screw, high volumetric efficiency, good performances in partial load, long service life, low leakage, low vibration, low noise, and simple configuration [21]. The fluid in various states, such as steam at superheated state or at saturated state, gas at high pressure, fluid at gas-liquid two-phase, and heat fluid, can all be used in SSE as its working fluid [22]. The key laboratory where the authors work has rich experience in manufacturing and researching SSE [21-26]. Therefore, in this paper based on the feasibility of SSE technology, two redesigned

KCS 34g 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 34g.

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 34g with a throttle valve.

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

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

The schematic diagram of the original KCS 34g with a throttle valve is presented in fig. 1. The ammonia-lean solution at Point 9 is throttled to the condensation pressure at state Point 13 after flowing through the regenerator, and then mixed with the ammonia-rich vapor in the absorber to form a working solution with initial ammonia fraction. The working solution flows through condenser to reach state Point 2. After being pressurized by pump, the working solution splits into two fluids which flow through evaporator and regenerator, respectively. After being heated, the aforementioned two fluids combine into one at Point 8 and then it is sent to separator, in which it is split into ammonia-rich vapor at Point 10 and ammonia-lean solution at Point 9. Two redesigned cycles based on the original KCS 34g are presented in fig. 2.

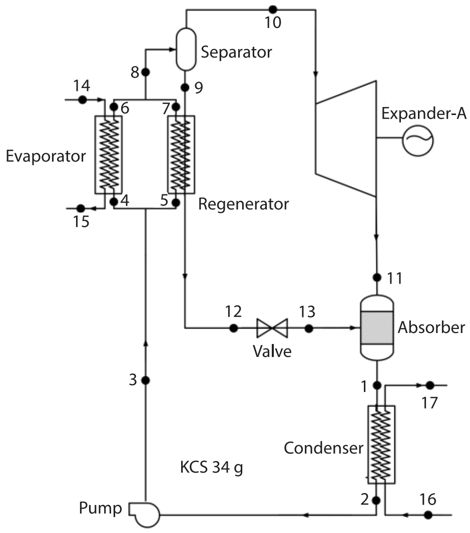

Figure 1. The schematic diagrams of the original KCS 34g [12]

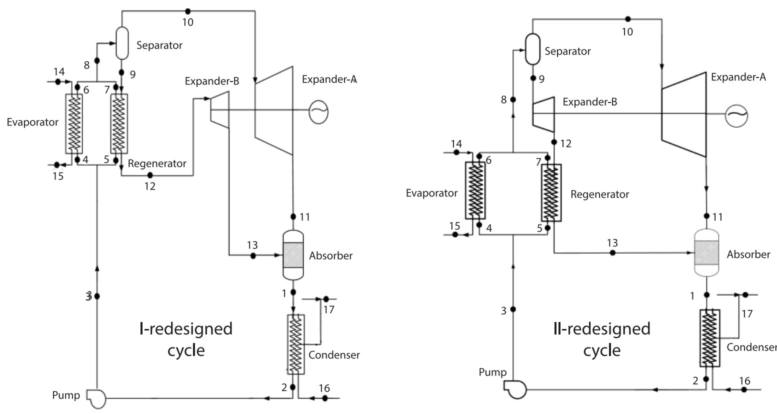

Figure 2. Two redesigned cycles based on the original KCS 34g

Table 1. Initial conditions used for calculation

Initial condition	Value
T_{14} [K]	400
T_{15} [K]	353
m_h [kgs^{-1}]	1
m_c [kgs^{-1}]	1
T_{16} [K]	290
T_0 [K]	300
ΔT_{me} [K]	25
ΔT_{con} [K]	2
η_{Exp-A} [–]	0.8
η_{Exp-B} [–]	0.65
η_{pum} [–]	0.8
n [–]	12:1

General assumptions

In this paper, the 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 3 shows the flowcharts of the calculation programs for the original KCS 34g and its two redesigned cycles. In order to simplify the calculation, the following assumptions are used:

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

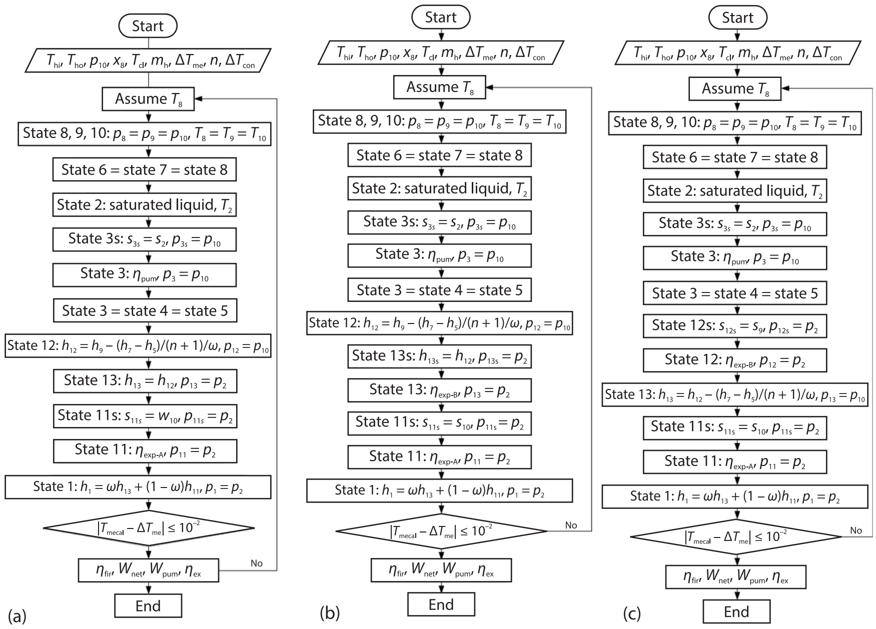

Figure 3. The flowcharts of the calculation programs for the optimization and analysis of the original KCS 34g and its two redesigned cycles; (a) the KCS 34g, (b) I-redesigned cycles, and (c) II-redesigned cycles

Thermodynamic analysis

In the cycle, flue gas is used as the heat source, and its inlet and outlet temperatures are T_{14} and T_{15}, respectively. The heat input of the system:

$$Q_{eva} = \dot{m}_h c_h \left(T_{14} - T_{15}\right) \tag{1}$$

The mass-flow ratio of evaporator to regenerator:

$$n = \dot{m}_{4\text{-}6} : \dot{m}_{5\text{-}7} \tag{2}$$

Heat exchanged in condenser and regenerators:

$$Q_{\text{con}} = \dot{m}_{\text{wf}} \left(h_1 - h_2 \right) \tag{3}$$

$$Q_{\text{reg}} = \dot{m}_{5-7} \left(h_7 - h_5 \right) \tag{4}$$

Work produced by expander-A:

$$W_{\text{Exp-A}} = \dot{m}_{10} \left(h_{10} - h_{11} \right) \tag{5}$$

Work produced by SSE-B:

– For I-redesigned cycle

$$W_{\text{Exp-B}} = \dot{m}_9 \left(h_{12} - h_{13} \right) \tag{6}$$

– For II-redesigned cycle

$$W_{\text{Exp-B}} = \dot{m}_9 \left(h_9 - h_{12} \right) \tag{7}$$

Power consumed by pump:

$$W_{\text{pum}} = \dot{m}_{\text{wf}} \left(h_3 - h_2 \right) \tag{8}$$

Net work output of system:

$$W = W_{\text{Exp-A}} + W_{\text{Exp-B}} - W_{\text{pum}} \tag{9}$$

Thermal efficiency:

$$\eta = \frac{W}{Q_{\text{eva}}} \tag{10}$$

Exergy at the inlet of heat source:

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

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:

$$\eta_{ex} = \frac{W}{E_{\text{in}}} \tag{12}$$

The logarithmic mean temperature difference in evaporator:

$$\Delta T_{\text{mecal}} = \frac{\left(T_{14} - T_6 \right) - \left(T_{15} - T_4 \right)}{\ln \dfrac{T_{14} - T_6}{T_{15} - T_4}} \tag{13}$$

Results and discussion

When neglecting the pressure drop of the working fluid in absorber, heat exchanger, and pipe-line, there are only two pressures in the KCS 34g, namely evaporation pressure and condensation pressure. For KCS 34g, the condensation pressure is determined by the given cooling conditions and ammonia-water concentration. The inlet and outlet temperatures of the separator (T_8, T_9, and T_{10}) are considered equal. The outlet temperatures at the cold side of evaporator and regenerator (T_6, T_7) are also considered equal. Given the heat source temperature and the logarithmic mean temperature difference in evaporator, it can be concluded that the work produced by expander-A and power consumed by pump in three cycle systems are constant under the same pressure and ammonia-water concentration. Therefore, this paper discusses the influences of the ammonia-water concentration, x_8, and evaporation pressure, p_{10}, on the cycle thermal efficiency, net work, work produced by SSE-B, and system exergy efficiency.

Figure 4 depicts the variation of cycle thermal efficiency, exergy effiency, and net work with the evaporation pressure and ammonia-water concentration for the original KCS 34g and its two redesigned cycles. Since the heat input of the system is given in the calculation, net work and thermal efficiency has the same variation trend. 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. It can be seen from fig. 4 that the thermal efficiency and net work of each redesigned cycle is higher than that of the original KCS 34g and the II-redesigned cycle performs best. At a low and moderate evaporation pressure, there is an optimal ammonia-water concentration and it increases with the increase of evaporation pressure. When the ammonia-water concentration is higher than the optimal concentration, the decreasing trend of thermodynamic performance of three cycles gradually slows down with the increase of ammonia-water concentration and evaporation pressure. At a high evaporation pressure, the thermodynamic performance of three cycles increase slowly. When the evaporation pressure is 3.0 MPa, the thermal efficiency and net work of three cycle systems increase with the increase of working fluid concentration and are close to each other. This indicates that the performance advantage of the redesigned cycle systems with single-screw expander gradually decreases at high working fluid concentration. Compared with the low cycle thermal efficiencies of three cycles shown in fig. 4, their cycle exergy efficiencies are higher. The highest cycle exergy efficiency of 54.14% can be obtained in the II-redesigned cycle when the evaporation pressure is 3.0 MPa and ammonia-water concentration is 0.85.

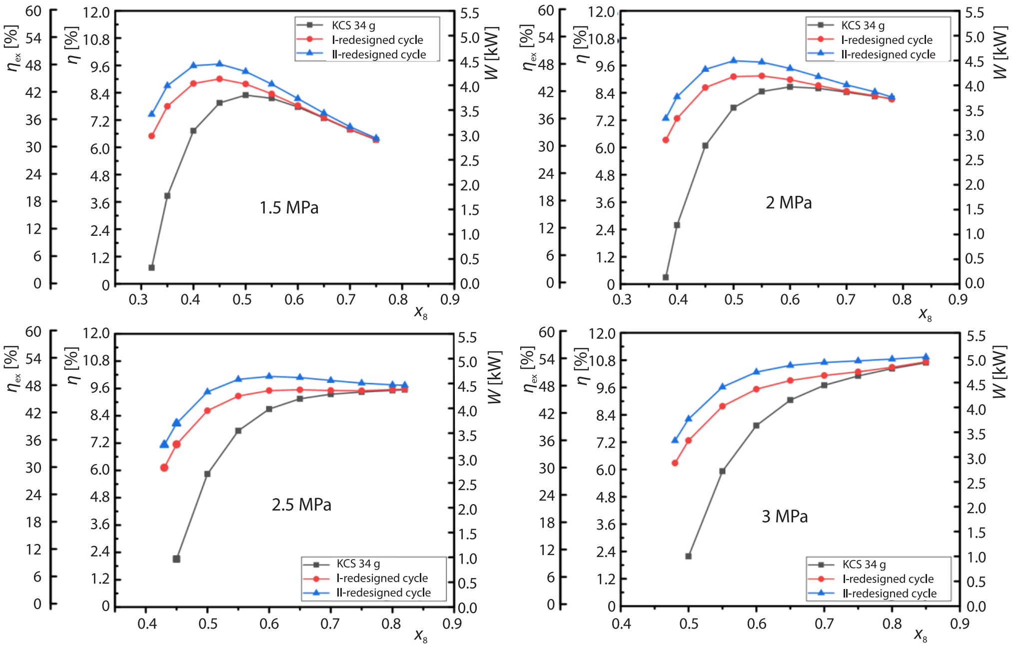

Figure 4. The variation of cycle thermal efficiency, exergy efficiency, and net work with the evaporation pressure and ammonia-water concentration

Figure 5 depicts the variation of the work produced by SSE-B with the evaporation pressure and ammonia-water concentration in the I-redesigned and II-redesigned cycles. It can be seen from fig. 5 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. With the concentration increase of ammonia-water, the work produced by the SSE-B in the two redesigned cycles gradually decreases. When the ammonia-water concentration is much lower than the optimal concentration, the SSE-B produces much work and plays a positive role in net work output of cycle system. This can be explained by the non-isothermal evaporation characteristics of ammonia-water. When the ammonia-water concentration is very low, it is closer to pure water, which leads to a small temperature glide. In this case, SSE can just play its technical advantage of two-phase expansion. 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.

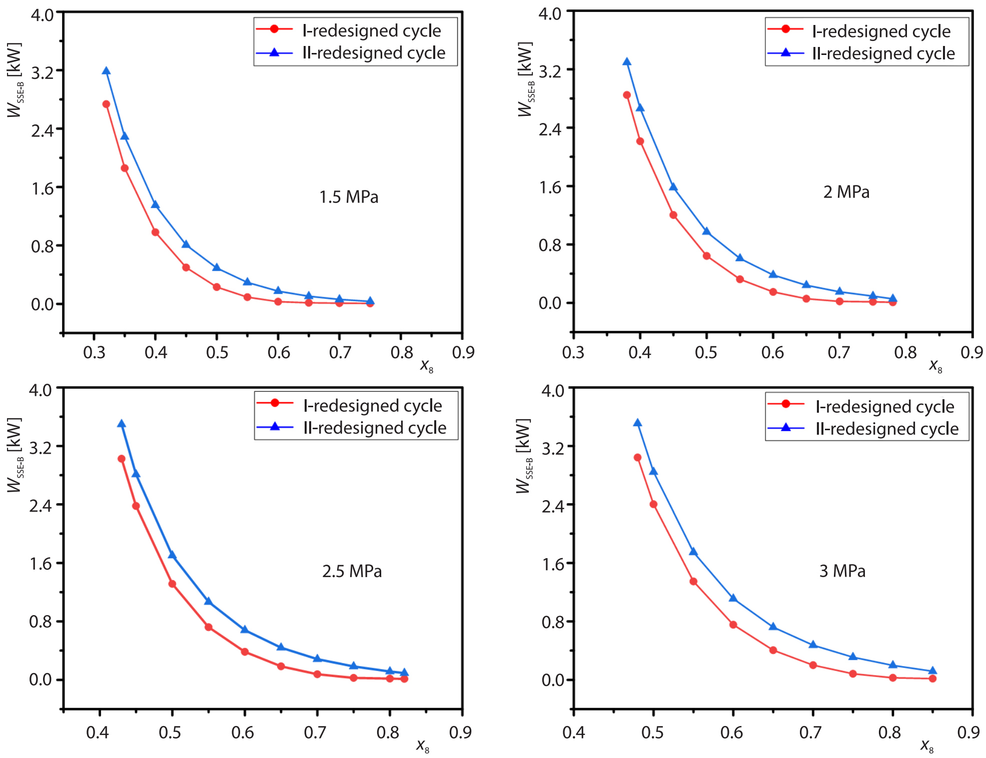

Figure 5. The variation of the work produced by SSE-B with the evaporation pressure and ammonia-water concentration in the I-redesigned and II-redesigned cycles

Conclusions

In order to recover the energy loss due to throttling in the path of the ammonia-lean solution in the KCS 34g, two redesigned cycles, namely I-redesigned cycle and II-redesigned cycle, in which SSE 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. In the II-redesigned cycle, the SSE is placed between the gas-liquid separator and the regenerator. The throttle valve between the absorber and the regenerator in the original KCS 34g is removed.

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

- The thermodynamic performance, including cycle thermal efficiency and net work, of each redesigned cycle is better than that of the original KCS 34g and the II-redesigned cycle performs best. At a low and moderate evaporation pressure, there is an optimal ammonia-water concentration and it increases with the increase of evaporation pressure. The performance advantage of the redesigned cycle systems with SSE gradually decreases at high working fluid concentration.
- 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 work produced by the SSE-B in the two redesigned cycles gradually decreases. When the ammonia-water concentration is much lower than the optimal concentration, the SSE-B produces much work and plays a positive role in net work output of cycle system.
- Compared with the low cycle thermal efficiencies of three systems, their cycle exergy efficiencies are higher. The highest cycle exergy efficiency of 54.14% can be obtained in the II-redesigned cycle when the evaporation pressure is 3.0 MPa and ammonia-water concentration is 0.85.

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Nomenclature

c – specific heat, [kJkg^{-1}K^{-1}]
E – exergy, [kJkg^{-1}]
h – enthalpy, [kJkg^{-1}]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
n – mass-flow ratio, [–]
Q – heat rate, [kW]
s – entropy, [kJkg^{-1}K^{-1}]
T – temperature, [K]
T_{ci} – temperature at condenser inlet, [K]
ΔT_{con} – pinch temperature difference for condenser, [K]
ΔT_{me} – logarithmic mean temperature difference, [K]
ΔT_{mecal} – calculated logarithmic mean temperature difference, [K]
W – power, [kW]
x – ammonia-water concentration, [–]

Greek symbol

η – thermal efficiency, [–]
η_{ex} – exergy efficiency, [–]
$\eta_{Exp\text{-}A}$ – the isentropic efficiency of expander A, [–]
$\eta_{Exp\text{-}B}$ – 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 – regenerator
wf – working fluid

Acronyms

KCS – Kalina cycle system
ORC – organic Rankine cycle
SSE – single-screw expander

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