Exergetic Performance Analysis of an Absorption Power Generation Cycle

Kyoung Hoon Kim[1] and Hyung Jong Ko[2]

***Abstract*—**The exergy method of analysis overcomes the limitations of the first law of thermodynamics and can clearly indicate the locations of energy degradation in a process that may lead to improved operation or technology. In this paper, an exergetical performance analysis is carried out for an absorption power generation system of Kalina using ammonia-water mixture as the working fluid for efficient conversion of low-temperature heat source. Effects of the ammonia mass fraction and the turbine inlet pressure are parametrically investigated on the exergy destruction (anergy) ratio of each component of the system and the exergy efficiency of the system. Results show that the exergetical performance is greatly affected by the ammonia mass fraction and the turbine inlet pressure and the component where the maximum exergy destruction occurs varies with the ammonia mass fraction.

***Keywords*—** ammonia-water power cycle, low-temperature source, exergy, exergy destruction (anergy).

I. Introduction

EXERGY is a measure of the departure of the state of a system from that of the environment. The traditional method of assessing the energy disposition of an operation with accompanying transfer and/or transformation of energy is by the completion of an energy balance based on the first law of thermodynamics. However, from such a balance no information is available on the degradation of energy, occurring in the process and to quantify the usefulness or quality of the heat content in various streams leaving the process as products, wastes, or coolants. The exergy method of analysis overcomes the limitations of the first law of thermodynamics. The concept of exergy is based on both first and second laws of thermodynamics and the main aim of exergy analysis is to identify the causes and to calculate the true magnitude of exergy losses [1]. The method of exergy analysis is well suited for furthering the goal of more energy resource use, for it enables the location, cause, and true magnitude of waste and loss to be determined [2]-[6].

Using ammonia-water mixture as the working fluid in the power generation systems instead of pure working fluids has many advantages for the efficient conversion from low-grade heat source to useful work. This is because the zeotropic mixture of ammonia and water evaporates over a wide range of temperature and its boiling temperature varies during the constant pressure phase change process. The variable temperature heat transfer process alleviates the temperature mismatch between hot and cold streams in the heat exchanging components of the system, which results in a reduction of exergy destruction (anergy) in the power cycles [7]-[13].

Ibrahim [14] studied an ammonia–water Rankine cycle and found that the design of heat exchanger networks can have a significant impact on the performance of power cycles. Ibrahim and Klein [15] developed a methodology based on heat-exchanger network syntheses to study and optimize the performance of an ammonia-water Rankine cycle. They showed that the performance of power cycles can be significantly influenced by the design of heat exchanger networks. Wager et al. [16] analyzed the ammonia-water Rankine cycle using scroll expander.

Kalina cycle was basically designed to replace the commonly used Rankine cycle as a bottoming cycle for a combined-cycle energy system as well as for generating electricity using low-temperature heat sources. Kalina cycle can be used in a variety of applications ranging from bottoming cycles for gas turbines to providing power from lower temperature waste heat sources [17]-[18]. Ogriseck [19] investigated the integration of the Kalina cycle process in a combined heat and power generation. Lolos and Rogdakis [20] studied Kalina cycle using low-temperature heat sources. Bombarda et al. [21] performed a comparative analysis on thermodynamic performance of Kalina and organic Rankine cycles. Arslan [22]-[23] presented an exergo-economic evaluation and optimization study of Kalina cycle using geothermal resources. Ganesh and Srinivas [24] examined a low-temperature Kalina cycle to optimize the heat recovery from solar collectors. Sun et al. [25] studied a solar boosted Kalina cycle with an auxiliary superheater.

In this study, exergetical performance analysis based on the first and second laws is carried out for an ammonia-water based Kalina cycle under the conditions of maximum mass flow rate of the working fluid. A particular attention is paid to an extensive investigation of the effects of ammonia mass fraction of the working fluid of binary mixture and turbine inlet pressure on the exergy destruction (anergy) at each component of the system and the exergy efficiency of the system.

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II. System Analysis

This study presents an exergetical performance analysis for a Kalina cycle using ammonia-water mixture as the working fluid for the purpose of efficiently converting a low-temperature heat source in the form of sensible heat to useful work. The schematic diagram of the system is shown in Figure 1. In the cycle, the working fluid which comes out of the condenser as a saturated liquid at temperature of T_L of state 1 is compressed through a pumping process in a pump to pressure of P_H of state 2. It is preheated in a regenerator to state 3, and then further heated in the source heat exchanger to the turbine inlet temperature of T_H of state 4. Then, it is separated in the separator into a saturated vapor of state 5 and a saturated liquid of state 7. The vapor of state 5 from the separator enters the turbine and produces useful work during the expansion in the turbine to the condensing pressure of state 6. On the other hand, the liquid of state 7 from the separator enters the regenerator to heat the mixture of state 2 exiting the pump to state 6. As a result of heat exchange it is cooled down to state 8, and then it is expanded in a throttle valve to state 9. The fluids of state 6 and state 9 are mixed in a mixer and enter the condenser with state 10.

In this study, it is assumed that the heat source fluid is a standard air at temperature of T_s, and the heat losses except at the heat exchangers and the pressure changes except at the turbine and pump are ignored. The isentropic efficiencies of the pump and turbine are assumed to be constant at η_P and η_T, respectively. In addition, it is assumed that the heat exchangers are operated with the pinch point condition, that is, the temperature difference between hot and cold fluids in the heat exchangers becomes a prescribed pinch point temperature difference of ΔT_{pp}. This is because it is important to produce the maximum power from the supplied heat source when producing the power from the low-temperature heat source in the form of sensible heat [11]-[12].

When the temperatures of the source and the coolant are T_s and T_c, respectively, and the base value of ammonia mass fraction at pump or condenser is x_b, the thermodynamic properties of ammonia-water mixture from state 1 to state 10 can be determined from the thermodynamic laws.

The rate of exergy flow, E, are calculated as

$$E = m[h - h_0 - T_0(s - s_0)] \tag{1}$$

where m is the mass flow rate, s is the specific entropy and the subscript 0 refers to the dead state. The rate of exergy input by source fluid, E_s, and the rate of net power production, W_{net}, can be obtained as follows

$$E_s = m_s c_{ps}[T_s - T_0 - T_0 \ln(T_s / T_0)] \tag{2}$$

$$W_{net} = m_5(h_5 - h_6) - m_1(h_2 - h_1) \tag{3}$$

Here, c_{ps} is the isobaric specific heat of source fluid. The second law efficiency η_2 and exergy efficiency of the system η_{ex} which is defined as the ratio of net power to exergy input can be written as follows

$$\eta_2 = W_{net} / (E_s - E_{s,out}) \tag{4}$$

$$\eta_{ex} = W_{net} / E_s \tag{5}$$

where $E_{s,out}$ denotes the rate of exergy flow of the exhausted source fluid. Anergy (exergy destruction) ratio of a component is defined as the ratio of the exergy destruction rate at the component to the exergy input rate by the source fluid to the system.

In this paper, the thermodynamic properties of liquid and vapor phase of the ammonia-water mixture are evaluated by

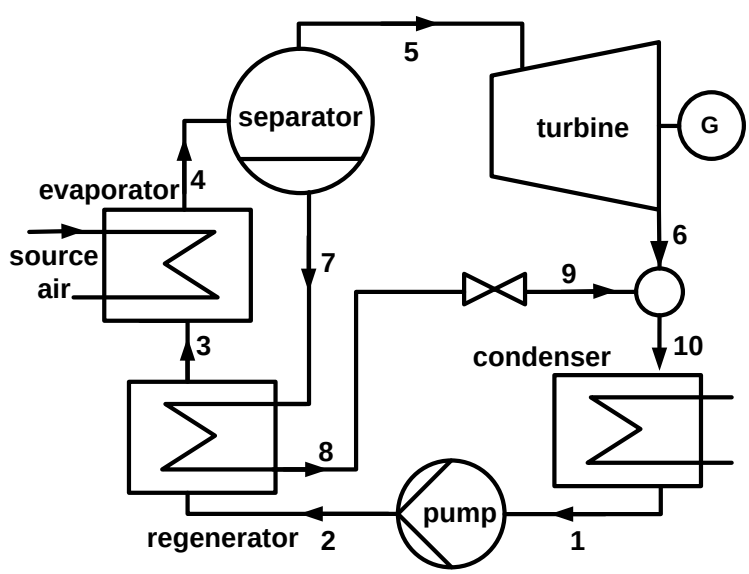

Figure 1: Schematic diagram of the system.

using the Gibbs free energy as was introduced by [26]. However, the equilibrium states of liquid and vapor phase are calculated using the method of [11]:

$$\mu_a^L = \left(\frac{\partial G_m^L}{\partial N_a}\right)_{T,P,N_w} = \left(\frac{\partial G_m^g}{\partial N_a}\right)_{T,P,N_w} = \mu_a^g \quad (6)$$

$$\mu_w^L = \left(\frac{\partial G_m^L}{\partial N_w}\right)_{T,P,N_a} = \left(\frac{\partial G_m^g}{\partial N_w}\right)_{T,P,N_a} = \mu_w^g \quad (7)$$

Here, N_a, N_w, and N are numbers of moles of ammonia, water, and the mixture, respectively.

III. Results and Discussions

The source fluid is assumed to be a standard air with the mass flow rate of 1 kg/s. The basic data of the system variables are as follows: source temperature T_s = 120℃, separator temperature T_H = 100℃, condensation temperature T_L = 25℃, coolant temperature T_c = 15℃, pinch point temperature difference ΔT_{pp} = 5℃, isentropic efficiency of pump η_p = 0.85, isentropic efficiency of turbine η_t = 0.90, quality limit at turbine exit y_t = 0.10, respectively. The ammonia mass fraction and the turbine inlet pressure are chosen as the key parameters of parametric investigation and are allowed to vary from 0.4 to 0.99 and from 15 to 40 bar, respectively.

The total exergy destruction is plotted against ammonia mass fraction in Figure 2 for various turbine inlet pressures. It can be seen that there exists a lower limit of ammonia mass fraction for the proper system operation for each turbine inlet pressure. This is because when the ammonia mass fraction is too low for a specified turbine inlet pressure, the working fluid of ammonia-water mixture still remains a compressed liquid and it cannot be separated into liquid and vapor in the separator. When the turbine inlet pressure is low such as 15 bar or 20 bar, the total exergy destruction decreases with increasing ammonia mass fraction and reaches a minimum value and then turns to increase with increasing ammonia mass fraction. If the turbine inlet pressure equals to or is greater than 25 bar, however, the total exergy destruction decreases monotonically with increasing ammonia mass fraction.

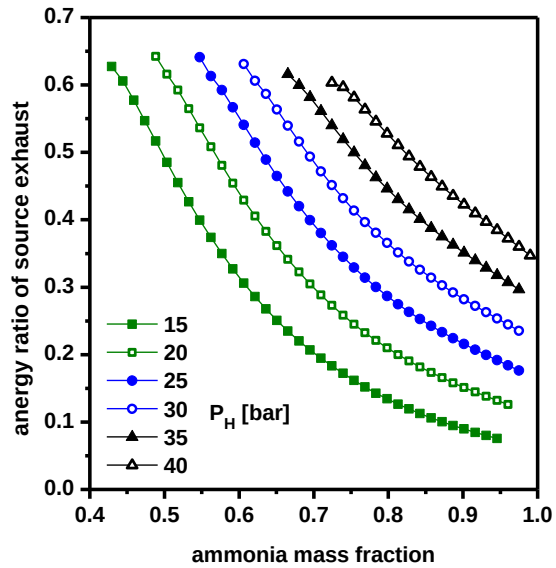

Figure 3. Plot of anergy ratio of source exhaust against ammonia mass fraction for various turbine inlet pressures.

Figure 3 shows the anergy ratio of source exhaust as a function of ammonia mass fraction for various turbine inlet pressures. The anergy ratio of source exhaust decreases with increasing ammonia mass fraction or decreasing turbine inlet pressure. It is because as ammonia mass fraction increases or turbine inlet pressure decreases, the specific heat transfer and the heat transfer rate in the source heat exchanger increases and this leads to lowered temperature of exhausted source fluid.

Figure 4 shows the anergy ratio of source heat exchanger

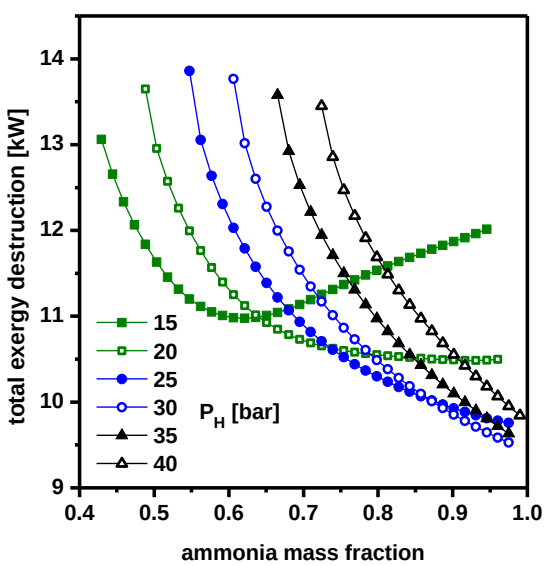

Figure 2. Plot of total exergy destruction against ammonia mass fraction for various turbine inlet pressures.

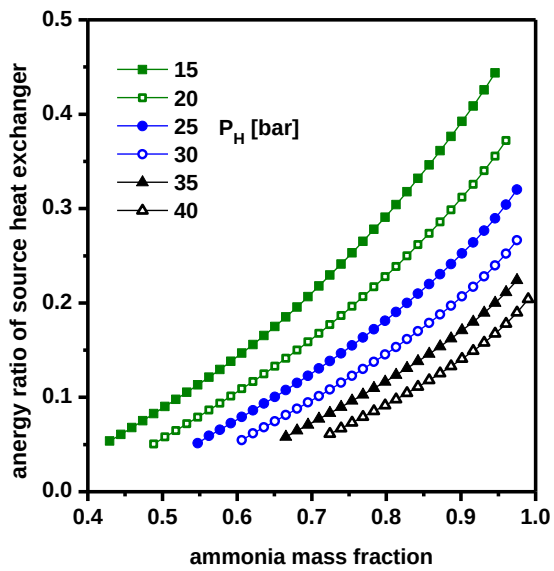

Figure 4. Plot of anergy ratio of source heat exchanger against ammonia mass fraction for various turbine inlet pressures.

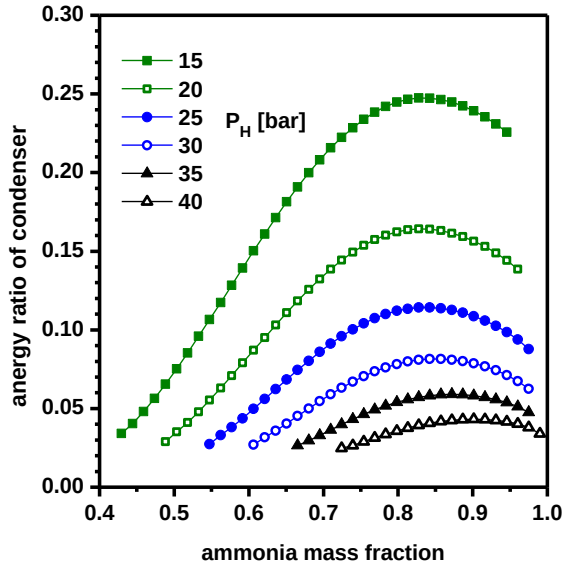

Figure 5. Plot of anergy ratio of condenser against ammonia mass fraction for various turbine inlet pressures.

with respect to ammonia mass fraction for various turbine inlet pressures. The source heat exchanger shows a behavior opposite to the source exhaust. Its anergy ratio increases with increasing ammonia mass fraction, since the temperature mismatch between the hot and cold streams in the heat exchanger becomes worse with increasing ammonia mass fraction. It is worthy to note that for a given turbine inlet pressure, the anergy ratio is the biggest among the components of the entire system.

The anergy ratio of condenser is plotted against ammonia mass fraction in Figure 5 for various turbine inlet pressures. It is higher for lower turbine inlet pressure if the ammonia mass fraction is specified. While it shows a convex upward curve and has a peak value with respect to ammonia mass fraction for each fixed turbine inlet pressure.

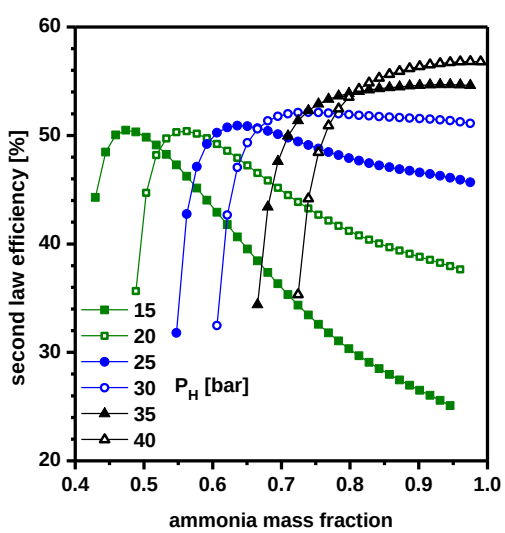

Figure 6. Dependence of the second law efficiency of the system on ammonia mass fraction for various turbine inlet pressures.

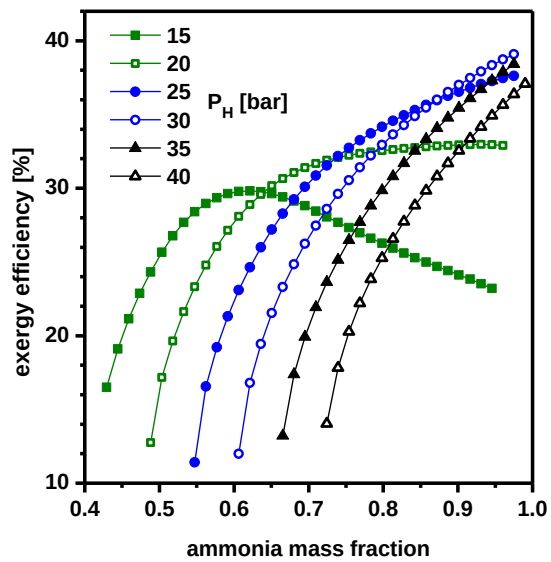

Figure 7. Dependence of the exergy efficiency of the cycle on ammonia mass fraction for various turbine inlet pressures.

Dependence of the second law efficiency of the system on ammonia mass fraction is shown in Figure 6 for various turbine inlet pressures. As ammonia mass fraction increases for a specified turbine inlet pressure, the efficiency η_2 increases initially, but begins to decrease after a peak point. This behavior implies that there exists an optimum value of ammonia mass fraction producing maximum net power for each specified turbine inlet pressure.

The exergy efficiency of the cycle defined as the ratio of net power production to the exergy input by the source fluid is plotted against ammonia mass fraction in Figure 7 for various turbine inlet pressures. For relatively low turbine inlet pressure there exist values of ammonia mass fraction giving the most efficient production of net power from the available heat source. On the contrary, cycles using higher ammonia mass fraction shows a better exergetic performance for relatively higher turbine inlet pressures. It is worthy to note that exergy efficiency is more meaningful than the second law efficiency when there is no additional thermodynamic process utilizing the exhausted source fluid. It is further noted that the exergy efficiency is intrinsically lower than the second law efficiency for any cycle.

IV. Conclusions

In this study, an exergy analysis is performed for a Kalina cycle using ammonia-water mixture as the working fluid for the efficient conversion of the sensible heat of low-temperature heat source to useful work. The main conclusions from the parametric study of the system are summarized as follows.

The heat exchanger and the source exhaust show an opposite anergy behavior with respect to ammonia mass fraction and turbine inlet pressure. The anergy ratio of the former increases, while that of source exhaust decreases as the ammonia mass fraction increases or the turbine inlet pressure decreases. Since these two anergy ratios are biggest among others, it is important

to reduce the exergy destructions by exhausted source fluid and the irreversible heat transfer in the source heat exchanger for the improvement of the system performance. The second law efficiency and the exergy efficiency of the system has an optimum value with respect to ammonia mass fraction. The exergy efficiency rather than the second law efficiency is desirable for the assessment of system performance if the additional thermodynamic process utilizing the exhausted source fluid is absent.

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