COMPARATIVE EXERGY ANALYSIS OF ORGANIC AND AMMONIA-WATER RANKINE CYCLES

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Abstract - This paper presents a comparative exergy analysis for organic Rankine cycle (ORC) and ammonia-water based Rankine cycle (AWRC). Effects of the system parameters such as working fluid, turbine inlet pressure, and mass fraction of ammonia on the exergetical performance are systemically investigated. The results show that as the turbine inlet pressure increases, the exergy destruction ratios of source exhaust, turbine and pump increase, but those of the heat exchanger, condenser, and the coolant exhaust decrease. The component where the maximum exergy destruction occurs is the condenser in ORC but the coolant exhaust in AWRC.

Index Terms - Organic Rankine Cycle, Ammonia-Water Rankine Cycle, Exergy, Exergy Destruction.

I. INTRODUCTION

In recent years the ORC and AWRC have become a field of intense research and appeared as promising technologies for conversion of heat into useful energy of electricity [1]-[4]. The ORC has many advantages such as adaptability to various heat sources, lesser complexity and lesser maintenance. [5]. In ORC the saturation vapor curve is a crucial characteristics of a working fluid and affects the fluid applicability, cycle efficiency, and arrangement of associated equipment in a power generation system. Therefore, the selection of working fluid matching with the available heat source is essential to its successful conversion into useful energy [6]-[9].

Drescher and Bruggemann [10] performed an investigation of the ORC in solid biomass power and heat plants, and Schuster et al. [11] reported numerous running applications, such as geothermal power plant, biomass fired combined heat and power plants, solar desalination plants, waste heat recovery or micro CHP. Tchanche et al. [12] performed a comparative performance analysis of solar organic Rankine cycle using various working fluids. Kim and Perez-Blanco [13] investigated a cogeneration of power and refrigeration using ORC for conversion of low-temperature heat source.

The power generation systems using ammonia-water mixture as a working fluid have been proven to be a feasible method for conversion of low-grade finite heat source. A major advantage for using zeotropic mixtures is that heat can be supplied or rejected at variable temperature for a constant pressure, which alleviates the temperature mismatch between hot and cold streams in heat exchanging components of the system [14]-[15]. Ogriseck [16] carried out an analysis of the integration of the Kalina cycle process in a combined heat and power generation, while Roy et al. [17] investigated an ammonia-water Rankine cycle with finite size thermodynamics in the context of reasonable temperature differences in the heat exchangers. Kim and Kim [18] conducted a thermodynamic analysis of a combined cycle consisting of an ammonia-water Rankine cycle and a LNG Rankine cycle using low-grade heat source.

Exergy is a measure of the departure of the state of a system from that of the environment, and the exergy analysis based on the thermodynamic second law is suitable for more effective energy resources use, since it enables the location, cause, and true magnitude of waste and lost to be determined [19]-[21]. In this work a comparative exergetical performance analysis is carried out for an organic Rankine cycle and an ammonia-water Rankine cycle for recovery of low-grade heat source. Special attention is focused on the effects of variation of working fluid, turbine inlet pressure, and ammonia mass fraction on the exergetical system performance.

II. SYSTEM ANALYSIS

A schematic diagram of the ORC or AWR combined cycle, consisting of a turbine, pump, heat exchanger and condenser is shown in Fig. 1.

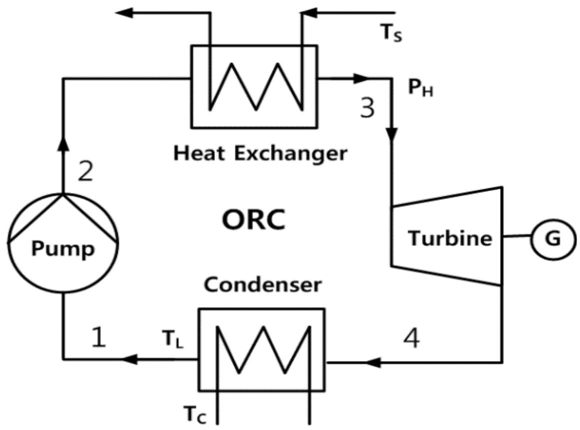

Fig. 1. Schematic diagram of the system.

In the system the working fluid is pressurized in the pump from a saturated liquid state (state 1) to a compressed liquid state (state 2). The fluid is then heated in the heat exchanger with the source fluid to a superheated vapor state (state 3). After the mechanical energy is obtained (state 4) owing to the expansion process in the turbine, the flow enters the condenser and is cooled with the coolant and returns to state 1.

In this work it is assumed as follows: The flow is steady and all components are well insulated. The fluid is a pure vapor at the turbine inlet and the isentropic efficiencies of the pump and turbine are constant. The turbine inlet pressure is lower than the critical pressure of the working fluid, so the cycle is a subcritical one. Additionally, each of the heat exchangers is assumed to be operated with a pinch point condition, which means that the minimum temperature difference between the hot and cold streams in the heat exchanger reach the prescribed value of the pinch temperature difference [3].

For a specified mass flow rate of the source fluid ms, the mass flow rates of the working fluid mw and the coolant mc can be determined from the energy balances at the heat exchanger and condenser as follows:

$$m_w = \frac{m_s c_{ps}(T_s - T_{sout})}{h_3 - h_2}, \qquad (1)$$

$$m_c = \frac{m_w(h_4 - h_1)}{c_{pc}(T_{cout} - T_c)}. \qquad (2)$$

Here cp denotes the specific heat, h the specific enthalpy of the working fluid, and the subscripts s and c represent the source fluid and the coolant, respectively. The heat addition rate to the system (Qin), the net power productions of the system (Wnet), can then be evaluated in accordance to the following equations:

$$Q_{in} = m_w(h_3 - h_2), \qquad (3)$$

$$W_{net} = m_w(h_3 - h_4) - m_w(h_2 - h_1). \qquad (4)$$

When a system undergoes a steady state operation, the thermodynamic properties of working fluid can be arbitrarily assigned to be zero as reference values. Therefore the thermo-mechanical enthalpy, entropy, and exergy at the ambient condition or dead state can be neglected regardless of its chemical composition. The specific exergy e and the rate of exergy input to the system by source fluid can be calculated as [21]:

$$e = h - h_0 - T_0(s - s_0), \qquad (5)$$

$$E_{in} = m_s c_{ps}\{T_s - T_0 - T_0 \ln(T_s / T_0)\}. \qquad (6)$$

Here s is the specific entropy and subscript 0 refers the dead state. The exergy efficiency of the system ηex is defined as the ratio of net work to exergy input, and the exergy destruction or anergy of the adiabatic system is calculated as the difference of exergy input and output. The exergy destruction ratio at a system component is defined as the ratio of anergy there to the exergy input by source fluid. Then summation of all anergy ratios of the system and the exergy efficiency becomes unity [21]:

$$\eta_{ex} = W_{net} / E_{in}, \qquad (7)$$

$$\eta_{ex} + D_{sout} + D_h + D_c + D_{cout} + D_w = 1. \qquad (8)$$

where Dsout, Dh, Dc, Dcout, Dw are exergy destruction ratio of the source exhaust, heat exchanger, condenser, coolant exhaust, and net work (turbine and pump), respectively.

In this paper, the thermodynamic properties of ammonia-water mixture in AWRC were evaluated by using the methods of [22]-[23], and the thermodynamic properties of the working fluids in ORC were evaluated by using Patel-Teja equation [24]-[25].

III. RESULTS AND DISCUSSIONS

In this study, it is assumed that the source fluid is water with a mass flow rate of 1 kg/s The basic operation data use for the simulation of this study are as follows; source temperature, TS = 200°C, turbine inlet temperature, TH = 185°C, condensation temperature, TL = 30°C, coolant temperature, TC = 15°C, turbine inlet pressure, PH = 15 bar, pinch temperature difference, ΔTpp = 8°C, isentropic efficiencies of pump and turbine, ηp = 0.80 and ηt = 0.85. The basic ammonia mass fractions are considered as xb = 50%, 60%, 70%, and 80%.

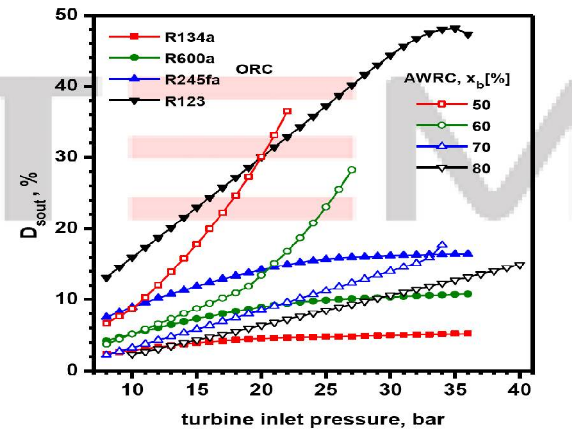

Fig. 2. Effects of turbine inlet pressure on the exergy destruction ratio of source exhaust, Dsout.

Fig. 2 shows the effects of turbine inlet pressure on the exergy destruction ratio of source exhaust, Dsout. In ORC, the ratio increases with increasing turbine inlet pressure, since as the turbine inlet pressure increases, the evaporation temperature of the working fluid increases, which leads to higher exit temperature of the source fluid. For a specified turbine inlet pressure, the ratio increases with increasing critical temperature of the working fluid, so the ratio is the highest for R123 and the lowest for R134a. In AWRC, the ratio increases with increasing turbine inlet pressure or decreasing ammonia mass fraction, since the evaporation temperature increases with increasing turbine inlet pressure or decreasing ammonia mass fraction.

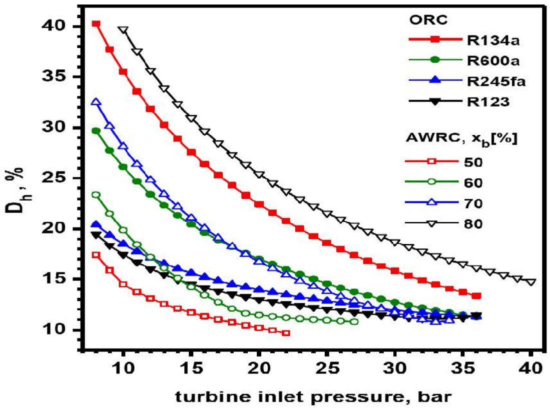

Fig. 3. Effects of turbine inlet pressure on the exergy destruction ratio of heat exchanger, Dh.

Fig. 3 displays the effects of turbine inlet pressure on the exergy destruction ratio of the heat exchanger, Dh. In ORC, the ratio decreases with increasing turbine inlet pressure, since as the turbine inlet pressure increases, the evaporation latent heat decreases, which causes less heat transfer in the heat exchanger. For a given turbine inlet pressure, the ratio decreases with increasing critical temperature of the working fluid, so the ratio is the highest for R134a and the lowest for R123. In AWRC, the ratio decreases with increasing turbine inlet pressure or ammonia mass fraction, since the heat transfer in the heat exchanger decreases with increasing turbine inlet pressure or ammonia mass fraction.

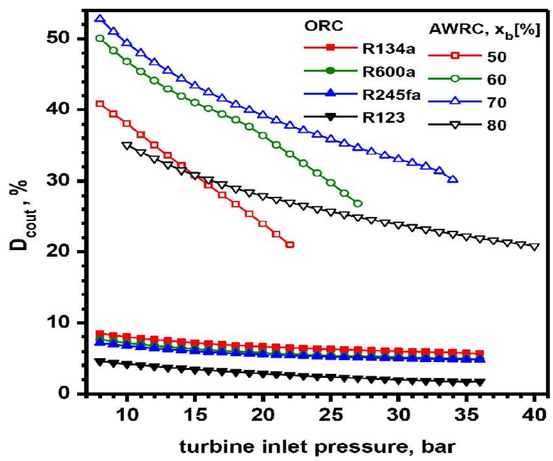

Fig. 4. Effects of turbine inlet pressure on the exergy destruction ratio of coolant exhaust, Dcout.

Fig. 4 illustrates the effects of turbine inlet pressure on the exergy destruction ratio of the coolant exhaust, Dcout. In ORC, the ratio decreases with increasing turbine inlet pressure, since as the turbine inlet pressure increases, the expansion ratio of the working fluid through the turbine increases, which leads to a lower the turbine exit temperature of the working fluid. For a given turbine inlet pressure, the ratio decreases with increasing critical temperature of the working fluid, so the ratio is the highest for R134a and the lowest for R123. In AWRC, the ratio decreases with increasing turbine inlet pressure, since as the turbine inlet pressure, the pressure ratio increases, which leads to a lower temperature of working fluid at the turbine exit. It can be seen from the figure that the exergy destruction ratios in AWRC are much higher than those in ORC.

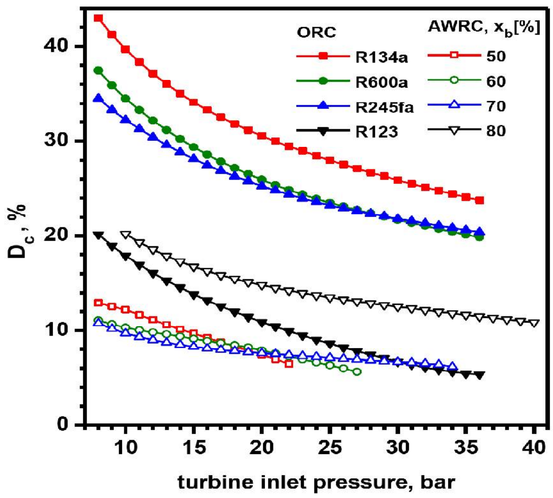

Fig. 5. Effects of turbine inlet pressure on the exergy destruction ratio of condenser, Dc.

Fig. 5 shows the effects of turbine inlet pressure on the exergy destruction ratio of the condenser, Dc. In ORC, the ratio decreases with increasing turbine inlet pressure, since as the turbine inlet pressure increases, the turbine exit temperature of the working becomes lowered and consequently the heat removal at the condenser decreases. For a given turbine inlet pressure, the ratio decreases with increasing critical temperature of the working fluid, so the ratio is the highest for R134a and the lowest for R123. In AWRC, the ratio decreases with increasing turbine inlet pressure, since as the turbine inlet pressure, the transfer at the condenser decreases due to lowered turbine exit temperature of the working fluid.

Fig. 6 shows the effects of the turbine inlet pressure on the exergy efficiency which is defined as the ratio of the net power production to the exergy supply into the system. In ORC, the exergy efficiency increases with increasing turbine inlet pressure for R134a, R600a, and R245fa, but has a peak with respect to the turbine inlet pressure for R123. And the working fluid which makes the exergy efficiency the highest is R123 when the turbine inlet pressure is lower than 20 bar, but R600a when the turbine inlet pressure is higher than 20 bar. In AWRC, the exergy efficiency increases with increasing turbine inlet pressure for 70% and 80% of mass fraction, but has a peak with respect to the turbine inlet pressure for 50 and 60% of mass fraction for R123. It can be seen from the figure that the exergy efficiencies in ORC are higher than those in AWRC.

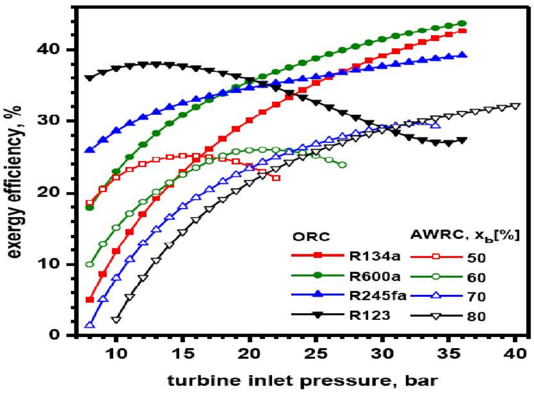

Fig. 6. Effects of turbine inlet pressure on the exergy efficiency.

Fig. 7 shows the effects of the turbine inlet pressure on the exergy destruction ratios in ORC when the working fluid is R600a. It can be seen from the figure that as the turbine inlet pressure increases, the exergy destruction ratios of source exhaust, turbine and pump increase, but those of the heat exchanger, condenser, and the coolant exhaust decrease. Among the exergy destruction ratios of the system, the exergy destruction ratio of condenser is the greatest, and it decreases from 38% to 20% as the turbine inlet pressure increases. On the other hand in AWRC, the exergy destruction ratio of coolant exhaust is the greatest which decreases from 50% to 26% as the turbine inlet pressure increases from 8 bar to 36 bar.

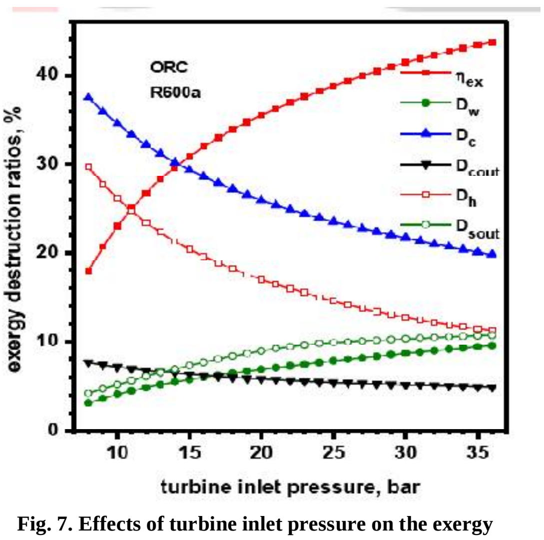

Fig. 7. Effects of turbine inlet pressure on the exergy destruction ratios in ORC for R600a.

Fig. 8 displays the effects of the turbine inlet pressure on the exergy destruction ratios in AWRC for xb = 50%. It can be seen from the figure that as the turbine inlet pressure increases, the exergy destruction ratios of source exhaust, turbine and pump increase, but those of the heat exchanger, condenser, and the coolant exhaust decrease, which are similar to the case of Fig. 7. The exergy destruction ratio of coolant exhaust is the greatest which decreases from 50% to 26% as the turbine inlet pressure increases from 8 bar to 27 bar.

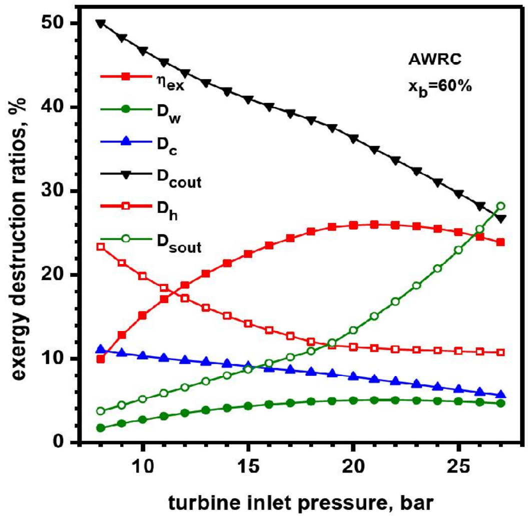

Fig. 8. Effects of turbine inlet pressure on the exergy destruction ratios in AWRC for xb = 60%.

CONCLUSION

This paper presents a comparative exergetical analysis of ORC and AWRC for the recovery of low-grade heat source. Water with 1 kg/s at 200°C is assumed to be the heat source. The parametric study was carried out with respect to the turbine inlet pressure. R134a, R600a, R245fa, and R123 were considered as the working fluids in ORC and ammonia mass fraction from 50% to 80% were considered in AWRC. The results show that as the turbine inlet pressure increases, the exergy destruction ratios of source exhaust, turbine and pump increase, but those of the heat exchanger, condenser, and the coolant exhaust decrease. The component where the maximum exergy destruction occurs is the condenser in ORC but the coolant exhaust in AWRC.

ACKNOWLEDGMENTS

This paper was supported by Kumoh National Institute of Technology.

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