EXERGETICAL PERFORMANCE ANALYSIS OF KALINA AND ORGANIC RANKINE CYCLES

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Abstract - The organic Rankine cycle (ORC) and the Kalina cycle system (KCS) are being considered as the most feasible and promising ways for recovery of low-grade finite heat sources. This paper presents a comparative exergetical performance analysis for ORC and Kalina cycle using ammonia-water mixture as the working fluid for recovery of low-grade heat. Effects of the system parameters such as working fluid, turbine inlet pressure, and mass fraction of ammonia on the exergetical performance are parametrically investigated. Results show that KCS does not show higher exergy efficiency, however, shows lower exergy destruction ratio of source heat exchanger and higher second-law efficiency than ORC.

Index Terms - Organic Rankine cycle, Kalina cycle, ammonia-water mixture, low-grade heat source, exergy.

I. INTRODUCTION

Renewables deployment has been a component of national planning agenda for many developed countries over the past few decades. Many countries and international organizations now view renewables as important elements of energy security, dynamic economic development, and environmental protection and greenhouse gas (GHG) emissions reduction efforts [1]. For efficient recovery of low-grade heat sources, power generation cycles such as Organic Rankine Cycle (ORC) and ammonia-water based cycles have been recognized as feasible and competitive and intensively investigated [2-4].

The ORC is a Clausius–Rankine Cycle in which an organic working fluid is used instead of water–steam. In the last years it became quite popular in energy production processes, due to the fact that it gives the possibility to use exhaust heat of low energy and temperature level [5]. The ORC has many advantages such as adaptability to various heat sources, lesser complexity and lesser maintenance [6]-[8]. The economics of a Rankine system is closely linked to the thermodynamic properties of the working fluid and a bad choice could lead to a low efficient and expensive plant [9]-[11].

If a zeotropic mixture like ammonia-water mixture is used as a working fluid in a power generation systems, the phase changes with varying temperature at constant pressure. Thus, the process of heat transfer with changing temperature reduces the imbalance of temperature difference between hot and cold streams in a heat exchanger, which is particularly effective for power cycle using a low-grade heat source in the form of sensible energy [12]-[13]. The Kalina cycle process is a modified Clausius Rankine process and the basic solution in a Kalina plant is a binary fluid with a certain ratio of water and ammonia. For the design of a Kalina cycle, an optimum between heat exchanger surface and the generated electricity has to be found. This optimum is influenced by different parameters such as the temperature level at the condenser, ammonia mass fraction in the binary fluid, turbine inlet pressure, and the temperature of heat source [14]-[15]. KCS-11 is a well-known Kalina cycle for low-temperature driven power generation and using ammonia-water as its working fluid for this purpose. Many researchers tried their best to clear the characteristics of the KCS-11 for using various forms of low-temperature heat sources [16].

Exergy is a measure of the departure of the state of a system from that of the environment and has been viewed as the maximum theoretical work obtainable from an overall system of system plus environment as the system passes from a given state to the dead state. 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 [17]-[18]. This paper presents a comparative analysis of exergetical performances for a Kalina cycle of KCS-11 and an ORC for recovery of low-grade finite heat source. The effects of working fluid, turbine inlet pressure, and ammonia mass fraction on the exergetical system performance are parametrically investigated.

II. SYSTEM ANALYSIS

The schematic diagram of the ORC (a) and (b) KCS are illustrated in Fig. 1. The ORC system considered consists of a heat exchanger, turbine, condenser and pump, while the KCS considered consists of an evaporator, separator, turbine, throttle valve, condenser, pump, and regenerator.

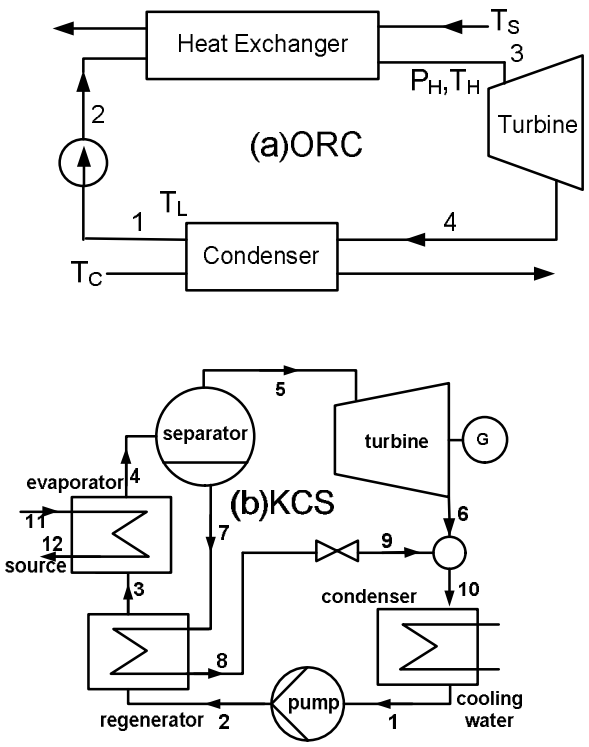


Fig. 1. Schematic diagram of the (a) ORC and (b) KCS.

In this work it is considered that a low-temperature heat source in the form of sensible energy is supplied to the system and turbine inlet pressure is lower than the critical pressure of the working fluid so the cycle is subcritical. Important assumptions used in this work are as follows: The working fluid leaves the condenser as saturated liquid at temperature of T_L. The minimum temperature difference between the hot and cold streams in the regenerator is operated at a prescribed pinch point, ΔT_{PP}. Pressure drop and heat loss of the systems are negligible [19].

Following equations are for ORC. For a mass flow rate of the source m_s, the mass flow rates of the working fluid m_w and the coolant m_c can be determined from the energy balances at the heat exchanger and condenser [4]:

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

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

Here c_p 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 (Q_{in}), the net power productions of the system (W_{net}), can then be evaluated in accordance to the following equations [19]:

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

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

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 evaluated as [17]-[18]:

$$e = h - h_0 - T_0(s - s_0)$$

(5)

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

(6)

Here s is the specific entropy and subscript 0 refers the dead state.

The exergy and second law efficiencies of the system, η_{ex} and η_{II} are defined as the ratio of net work to exergy input to the system, and the ratio of net work to the supplied exergy to the heat exchanger, respectively [18].

$$\eta_{ex} = \frac{W_{net}}{E_{11}} \quad (7)$$

$$\eta_{II} = \frac{W_{net}}{E_{11} - E_{12}} \quad (8)$$

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 [18]:

$$\eta_{ex} + D_{so} + D_h + D_c + D_{co} + D_w = 1 \quad (9)$$

where D_{so}, D_h, D_c, D_{co}, and D_w 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 KCS 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 air with 1 kg/s. The basic operation data use for the simulation of this study are as follows; source temperature, T_S = 200°C, turbine inlet temperature, T_H = 185°C, condensation temperature, T_L = 30°C, coolant temperature, T_C = 15°C, pinch temperature difference, ΔT_{pp} = 8°C, dead state temperature, T_0 = 15°C, dead state pressure, P_0 = 1 atm, isentropic efficiencies of pump and turbine, η_p = 0.85 and η_t = 0.85. The basic ammonia mass fractions are considered as x_b = 10%, 20%, 30%, 40%, 50% and 60%.

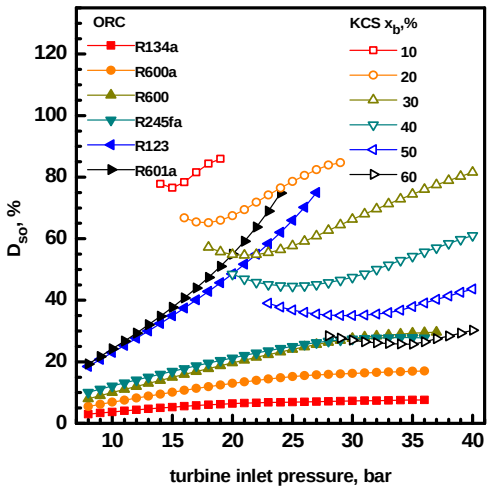


Fig. 2. Effects of turbine inlet pressure on the exergy destruction ratio of source exhaust, D_{so}.

Fig. 2 illustrates the effects of turbine inlet pressure on the exergy destruction ratio of source exhaust, D_{so}. In ORC, the ratio increases with increasing pressure, since the evaporation temperature rises and consequently the exit temperature becomes higher. The ratio increases with increasing critical temperature of the fluid, so the ratio is the highest for R601a and the lowest for R134a. In KCS, as the pressure increases, ratio firstly decreases for low pressures, reaches the minimum value and then increases again. It can be explained as follows. The evaporation temperature of the working fluid increases with increasing pressure, which leads to higher exit temperature of the source fluid. When the pressure becomes higher, however, the effect of decrease in the turbine exit temperature becomes dominant, which reduces the heat transfer with regenerator and consequently the exit temperature of the source fluid [19]. For a given turbine inlet pressure, the ratio decreases with increasing ammonia mass fraction, so the ratio is the highest for 10% and the lowest for 60%.

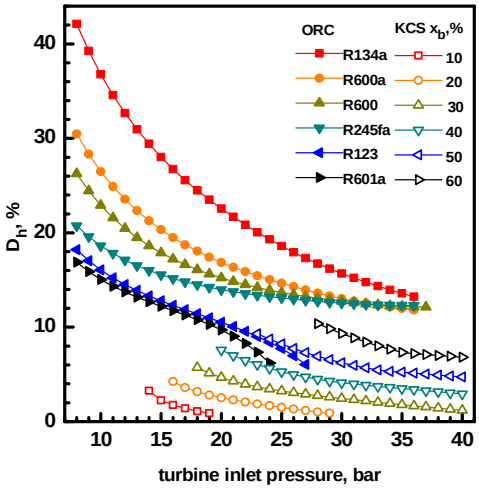


Fig. 3. Effects of turbine inlet pressure on the exergy destruction ratio of heat exchanger, D_h.

Fig. 3 shows the effects of turbine inlet pressure on the exergy destruction ratio of the heat exchanger, D_h. In ORC, the ratio becomes lower for higher pressures. It is because as the pressure increases, the evaporation latent heat decreases, and it leads to less heat transfer in the heat exchanger. The ratio decreases with increasing critical temperature of the working fluid, so the ratio is the highest for R134a and the lowest for R600a. In KCS, the ratio decreases with increase in the pressure, since the heat transfer in the heat exchanger decreases with increasing pressure. The ratio becomes higher when the ammonia mass fraction increases higher, due to higher mismatch between hot and cold streams in the heat exchanger [19]. It is to be noted that the ratios of KCS are much lower than those in ORC, which means that the exergy destruction in the source heat exchanger is greatly reduced due to phase transformation with variable temperature.

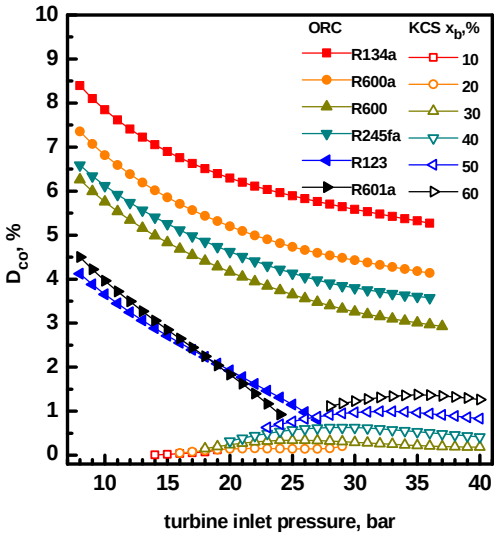


Fig. 4. Effects of turbine inlet pressure on the exergy destruction ratio of coolant exhaust, D_{co}.

Fig. 4 displays how the exergy destruction ratio of the coolant exhaust, D_{co}, is affected by the turbine inlet pressure. In ORC, the ratio drops with increase in the pressure, since as the pressure becomes higher, the expansion ratio of the working fluid through the turbine increases, and consequently the turbine exit temperature of the working fluid drops. The ratio drops with increasing critical temperature of the working fluid. In KCS, the ratio drops as the pressure increases, since higher turbine inlet pressure leads to a lower temperature of working fluid at the turbine exit. It can be observed from the figure that the exergy destruction ratios in KCS are much lower than ORC.

Fig. 5 shows how the exergy destruction ratio of the condenser, D_c, is affected by the turbine inlet pressure. In ORC, the ratio decreases with increasing pressure, since the higher turbine inlet pressure causes lower temperature of the working fluid at the turbine exit

and consequently lower heat removal at the condenser. On the other hand the ratio becomes lower for higher critical temperature of the working fluid, so the ratio is the highest for R134a and the lowest for R600a. In KCS, the exergy destruction ratio becomes lower as the turbine inlet pressure increases, due to lowered temperature of working fluid at the turbine exit. It can be seen from the figure that the destruction ratio becomes higher with increasing ammonia mass fraction, so the destruction ratio is the lowest for ammonia mass fraction of 10% and the highest for 60%.

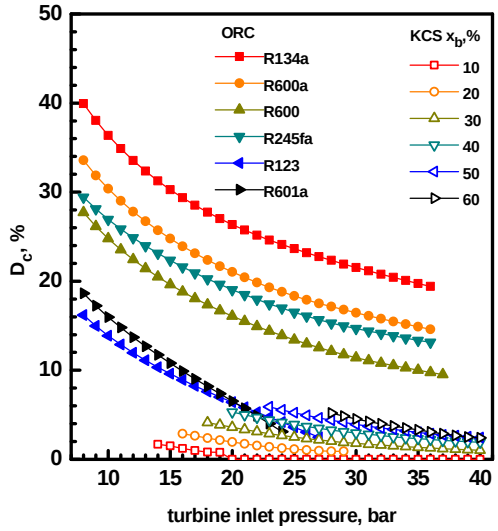


Fig. 5. Effects of turbine inlet pressure on the exergy destruction ratio of condenser, D_c.

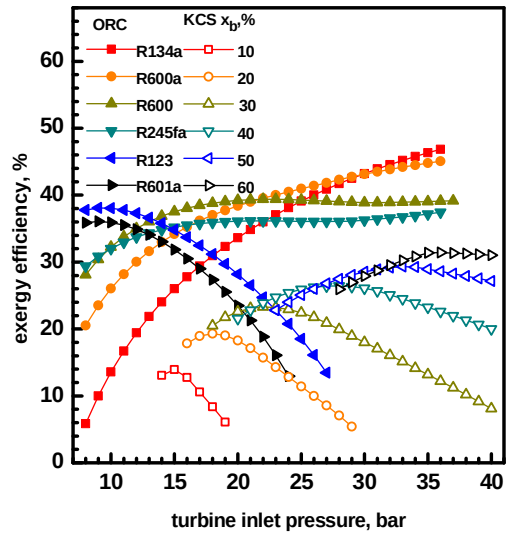


Fig. 6. Effects of turbine inlet pressure on the exergy efficiency.

Fig. 6 illustrates the effects of the turbine inlet pressure on the exergy efficiency. In ORC, the effects of the turbine inlet pressure on the exergy efficiency vary depending on the working fluid. As the turbine inlet pressure increases, the exergy efficiency increases for the working fluids with lower critical temperature such as R134a, R600a, R600, and R245fa, however, decreases for the working fluids with higher critical temperature such as R123 and R601a. It can be seen from the figure that the maximum exergy efficiency is the highest with R123 for low turbine inlet pressures, with R600 for middle turbine inlet pressures, and with R600a or R134a for high turbine inlet pressures. In KCS, the exergy efficiency firstly increases, reaches a peak value, and then decreases again, as the turbine inlet pressure increases. It is to be noted that the maximum exergy efficiency and the corresponding optimum turbine inlet pressure increase with increasing ammonia mass fraction as 13.9% and 15 bar for x_b = 10%, 19.3% and 18 bar for 20%, 23.3% and 22 bar for 30%, 26.4% and 28 bar for 40%, 29.3% and 33 bar for 50%, 30.8% and 34 bar for 60%.

Fig. 7 shows the effects of the turbine inlet pressure on the second law efficiency. In ORC, the second law efficiency increases with increasing turbine inlet pressure for the working fluids with lower critical temperature such as R134a, R600a, R600, and R245fa, however, has a local maximum value with respect to the turbine inlet pressure for the working fluids with higher critical temperature such as R123 and R601a. In KCS, the second law efficiency has a local maximum value with respect to the turbine inlet pressure for low ammonia mass fractions from 10% to 30%, however, increases with increase in the turbine inlet pressure for high turbine inlet pressures from 40% to 60%. As the turbine inlet pressure increases, the maximum efficiency decreases and the corresponding optimum turbine inlet pressure increases. It can be seen from the figure that the highest second law efficiency in KCS is higher than ORC.

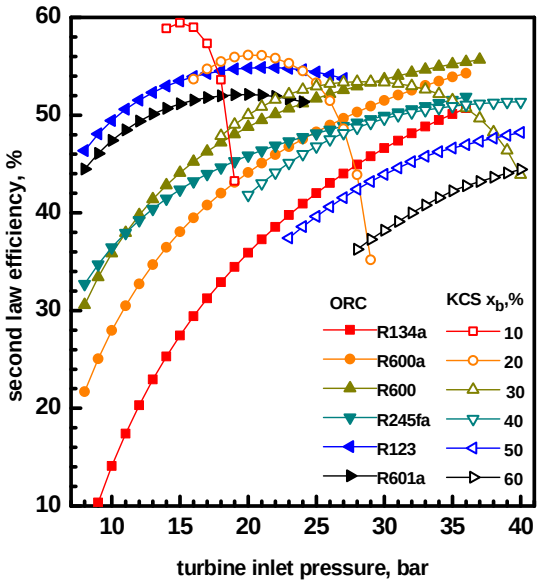


Fig. 7. Effects of turbine inlet pressure on the second law efficiency.

CONCLUSIONS

In this paper a comparative exergetical analysis is carried out for organic Rankine cycle (ORCR) and Kalina Cycle System (KCS) with KCS-11 using ammonia-water mixture as the working fluid for recovery of low grade finite heat source. R134a, R600a, R600, R245fa, R123, and R601a were considered as the working fluids in ORC and ammonia mass fraction from 10% to 60% were considered in KCS. The results are as follows. The exergy destruction ratio of source exhaust increases for higher critical temperature of working fluid in ORC and decreases with increasing ammonia mass fraction in KCS. The exergy destruction ratio of heat exchanger decreases with increasing turbine inlet pressure or critical temperature of working fluid in ORC, and with increasing turbine inlet pressure and with decreasing ammonia mass fraction in KCS. The destruction in KCS is much lower than ORC. KCS does not show higher exergy efficiency but show higher second law efficiency than ORC.

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