Exergy Analysis of Organic Rankine Cycle with Ejector Using Dry Fluids

Hyung Jong Ko and Kyoung Hoon Kim

***Abstract*—**Exergy performance of an organic Rankine cycle combined with an ejector refrigeration cycle is thermodynamically investigated for the utilization of low temperature heat sources. The cycle is driven by the sensible heat of water and the five kinds of dry fluids are considered as working fluid. The exergy consumption at each cycle elements and the exergy and second-law efficiencies of the cycle are simulated for varying turbine inlet pressure (TIP). Results show that the exergy loss of source exhaust and exergy destructions of boiler, ejector, condenser, and evaporator are dominant over those of coolant exhaust, turbine, pump, and expansion valve. For each working fluid the exergy efficiency has a peak value with respect to TIP, while the second-law efficiency increases monotonically with TIP. The exergy efficiency is higher for the working fluid having lower critical temperature.

***Keywords*—**ejector, exergy destruction, exergy efficiency, organic Rankine cycle, second-law efficiency

I. Introduction

SINCE it is difficult to efficiently convert low-grade thermal energy into electricity by conventional methods, there has been an increasing interest in the methods to effectively convert the low-grade thermal energy into useful forms of energy. The organic Rankine cycle (ORC) has attracted much attention as it is proven to be the most feasible method to achieve high efficiency in converting the low-grade thermal energy to more useful forms of energy [1-4]. Important reviews on the state of the art of research in energy conversion from low-grade energy sources are among others those of Raj et al. [5], Tchanche et al. [6] and Chen et al. [7]. A review on a state of the art report of ORC applications by Schuster et al. [8] is also worthy to cite, in which geothermal power plants, biomass fired cogeneration plants, and solar desalination plants are included.

The selection of working fluid matching with the available heat source is essential to its successful conversion into useful energy. Velez et al. [9] compared the working fluids used for the ORCs operating at low temperature. Wang et al. [10] proposed a theoretical thermal efficiency model based on an ideal ORC to analyze the influence of working fluid properties. Mago et al. [11] presented an analysis of regenerative ORCs using dry organic fluids. An exergy-based fluid selection method was suggested by Heberle and Brueggemann [12].

Ejectors are fascinating devices which have no moving parts. Because of this feature, ejector refrigeration cycle (ERC) is comparatively simple and can be driven by low-grade energy sources [13]. ERC can be combined with various power cycles for the cogeneration of power and refrigeration. Dai et al. [14] proposed a novel cycle which cogenerates power and refrigeration, by adding a turbine between the boiler and the ejector of ERC and using R123 as working fluid. The same cycle was investigated by Zheng and Weng [15], Ko and Kim [16], and Ko et al. [17] for the working fluid of R245fa, and by Kim et al. [18] for several working fluids. Moreover, its modified version in which only a part of vapor was extracted from the turbine to the ejector was studied by Wang et al. [19] for the same fluid and by Habibzadeh et al. [20] for several working fluids. Li et al. [21] proposed an ORC with ejector for the purpose of increasing the power output capacity and improving its efficiency.

In this study exergy performance of an ORC combined with an ERC is thermodynamically investigated. Exergy analysis is regarded as an appropriate tool for improving the performance of a thermodynamic cycle since it can locate and estimate the loss of useful energy of a cycle [22]. The cycle is driven by the sensible heat of water at 150^{o}C and the five kinds of dry organic fluids are considered as working fluid. The consumption of exergy at the cycle elements and the exergy and second-law efficiencies of the cycle are simulated for varying turbine inlet pressure.

II. System Analysis

The system considered in this study is an organic Rankine cycle combined with an ejector refrigeration cycle and is shown schematically in Fig. 1. The combined cycle is divided into two sub-cycles: a power sub-cycle and a refrigeration sub-cycle. Two sub-cycles share an ejector and a condenser. The primary flow in the ejector comes from the turbine and induces the secondary flow from the evaporator. The entire system is driven by the sensible heat supplied by a low-temperature heat source. Both the source fluid and the coolant for the condenser are considered as water. Five fluids are considered as working fluid of the cycle.

For the simplicity of investigation the following assumptions are used: The system operates in a steady state. Pressure drop and heat loss during the steady process in the system are negligible. The working fluid enters the turbine as superheated vapor, and leaves the condenser and evaporator as saturated

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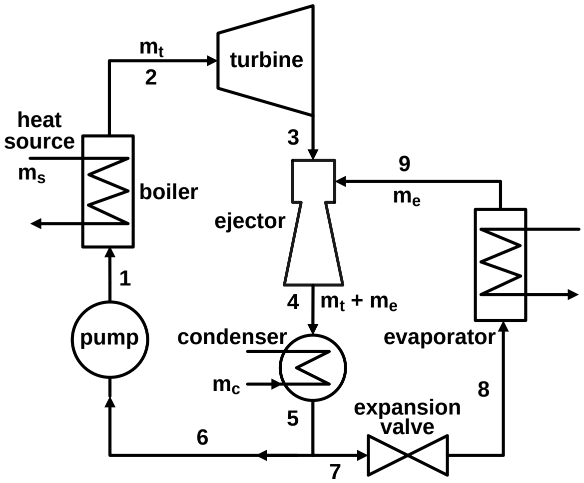

Fig. 1 Schematic diagram of the system.

liquid and saturated vapor, respectively. The minimum temperature difference between the hot and cold streams in the boiler and condenser is equal to a prescribed value of pinch point temperature difference. The isentropic efficiencies of pump and turbine are constant. The flow in the ejector is one-dimensional and the mixing of primary and secondary flows occurs at constant pressure. In addition the irreversible effects of the nozzle, mixing, and diffuser sections of the ejector are taken into account by using their efficiencies [14], [16]. The process in the expansion valve is isenthalpic.

The thermodynamic states at 1 to 9 of Fig. 1 can be determined if the turbine inlet pressure and temperature, turbine outlet pressure, and condenser and evaporator temperature are specified. In the process the mass flow rate entrained from the evaporator m_e which is the mass flow rate of refrigeration sub-cycle and the exit state 4 of the ejector are determined iteratively from the balance equations of mass, momentum and energy, along with the equation of state.

For a specified mass flow rate of source fluid m_s, the mass flow rates of the working fluid through the turbine and the coolant, denoted by m_t and m_c, respectively, can be determined from the equations of energy balance and the pinch point conditions [18]. Note that the mass flow rate through the condenser is equal to the sum of those through the turbine and the evaporator:

$$m_{cd} = m_t + m_e = m_t(1 + r_e), \tag{1}$$

where r_e is the entrainment ratio of ejector.

The rate of heat input, net power production, and the rate of refrigeration output are obtained as

$$Q_{in} = m_t(h_2 - h_1), \tag{2}$$

$$W_{net} = W_t - W_p = m_t[(h_2 - h_3) - (h_1 - h_6)], \tag{3}$$

$$Q_e = m_e(h_9 - h_8). \tag{4}$$

The rates of exergy flow of the working fluid, exergy input to the system by the source fluid, and exergy output associated with refrigeration can be written as

$$E = m[h - h_0 - T_0(s - s_0)], \tag{5}$$

$$E_{si} = m_s c_{ps}[T_{si} - T_0 - T_0 \ln(T_{si}/T_0)], \tag{6}$$

$$E_e = Q_e(T_0 - T_{cs})/T_{cs}. \tag{7}$$

Here m, h, s, and T denote the mass flow rate, specific enthalpy and entropy, and temperature of the fluid. The subscript 0 refers to the environment state and c_{ps} is the isobaric specific heat of the source fluid. T_{si} and T_{cs} are the temperatures of the source inlet and the space cooled by the evaporator, respectively. The rate of exergy loss due to source exhaust, E_{so}, can be obtained by replacing T_{si} with the source outlet temperature T_{so}. The rate of exergy flow of the coolant can be obtained similarly by using the isobaric specific heat c_{pc} and the temperature T_{ci} or T_{co}.

The exergy destruction of adiabatic system is equal to the difference of the input and output of exergy. The exergy destruction or loss ratio of each component is defined as its value divided by the exergy input by the source fluid. For example, exergy destruction ratio of boiler D_b and exergy loss ratio due to source exhaust D_{so} are calculated as

$$D_b = (E_{si} - E_{so} + E_1 - E_2)/E_{si}, \tag{8}$$

$$D_{so} = E_{so}/E_{si}. \tag{9}$$

The exergy efficiency and the second-law efficiency of the cycle are defined as follows:

$$\eta_{ex} = (W_{net} + E_e)/E_{si}, \tag{10}$$

$$\eta_{II} = (W_{net} + E_e)/(E_{si} - E_{so}). \tag{11}$$

η_{ex} is based on the exergy input and represents an estimation by assuming that the exergy of exhaust source is merely discharged to the environment. While η_{II} is based on the net exergy delivered to the system through the boiler and represents an optimistic estimation with a hope that the exergy of exhaust can be fully recovered. Note that two efficiencies are equal when the source exhaust temperature is equal to that of environment. In addition, if the coolant inlet state is taken as the dead state, the sum of the exergy efficiency and all exergy destruction and loss ratios of the system becomes unity.

III. Results and Discussions

An exergetical performance analysis based on the second law of thermodynamics is carried out for an organic Rankine cycle (ORC) combined with an ejector refrigeration cycle (ERC) driven by the low-temperature heat source. The considered source fluid is water flowing at a rate of m_s = 1 kg/s with inlet temperature of 150°C. Five dry fluids are considered as working fluid of the cycle and can be sequenced by their critical temperature: normal pentane (nC_5H_{12}), isopentane (iC_5H_{12}),

TABLE I
BASIC DATA FOR THE WORKING FLUIDS

Substance	M (kg/kmol)	T_{cr} (K)	P_{cr} (bar)	ω
normal pentane	72.150	469.65	33.69	0.249
isopentane	72.150	462.43	33.81	0.228
R123	136.467	456.90	36.74	0.282
R245fa	134.048	427.20	36.40	0.3724
isobutane	58.123	408.14	36.48	0.177

R123 ($CHCl_2CF_3$), R245fa ($CHF_2CH_2CF_3$), and isobutene (iC_4H_{10}). Their thermodynamic properties are calculated by the Patel-Teja equation of state [3], [23], [24]. The basic data of each fluid needed to apply the Patel-Teja equation are given in TABLE I, where M, T_{cr}, P_{cr}, and ω are molar mass, critical temperature, critical pressure, and acentric factor, respectively [25].

The most important parameter of the ORC combined with ERC is the turbine inlet pressure (TIP) because it affects both power generation and refrigeration output. In addition refrigeration performance is influenced by the turbine outlet pressure (TOP), condensing temperature, and evaporator temperature in a complicated manner. The main reason is the strong dependence of the entrainment ratio of the ejector on these parameters. In this study the exergetical performance of the system is investigated by varying TIP in a range of 6 to 30 bar. While the TOP, the turbine inlet and condensing temperature are fixed at 6 bar, 130°C and 25°C, respectively. Other basic calculation data are summarized as follows. Coolant temperature: 15°C, environment temperature: 15°C, evaporator temperature: -20°C, temperature of cooled space: -5°C, pinch point temperature difference: 5°C, isentropic efficiency of pump: 0.7, isentropic efficiency of turbine: 0.85, nozzle efficiency: 0.95, mixing efficiency: 0.95, diffuser efficiency: 0.95.

Only a part of the exergy supplied to the system by the source fluid is recovered as net power production and refrigeration exergy. The rest of it are either destroyed at the system components or lost by the source and coolant exhaust. Fig. 2 shows all percentage consumptions of supplied exergy relative to the source input exergy, E_{si}, for R123 which has medium critical temperature of the five working fluids considered. Recall that sum of all exergy consumption ratios is unity. For R123 TIP is restricted in a range of about 6.5 to 14.5 bar. The lower limit corresponds to the minimum workable TIP which must be greater than the TOP of 6 bar. The upper limit is the TIP when the turbine inlet quality is just one. The recovered exergy (η_{ex}), lost exergy due to source exhaust (D_{so}), destroyed exergy at the boiler (D_b), condenser (D_{cd}), ejector (D_j), and evaporator (D_e) are prevalent. The exergy loss due to coolant exhaust (D_{co}) and the exergy destruction at the turbine (D_t), pump (D_p), and expansion valve (D_v) are relatively small. This estimate of order of magnitude of exergy consumption is similar to the other working fluids. Therefore the dependences of the exergy consumption ratios on TIP are investigated only for the major elements.

Fig. 3 shows the ratio of exergy loss due to source exhaust versus TIP for various working fluids. Except isobutane TIP is limited above in order to satisfy the superheated inlet condition. For each fluid, exergy loss due to source exhaust increases almost linearly with TIP. It is the result of the increased source outlet temperature because the heat load of the boiler becomes smaller as the TIP gets higher for the same inlet temperature. For the same reason of reduced heat load, exergy loss due to source exhaust is higher for the fluids with higher critical temperature.

Ratios of exergy destruction of boiler and condenser are demonstrated in Fig. 4 with respect to TIP for various working fluids. Exergy destructions of boiler and condenser are comparable in magnitude and decrease with TIP for all working fluids. This is mainly the result of the effect of reduced mass

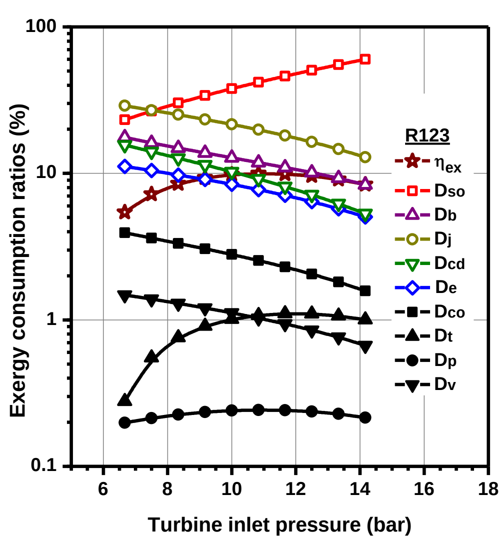

Fig. 2 Exergy consumption ratio of each element versus turbine inlet pressure for R123.

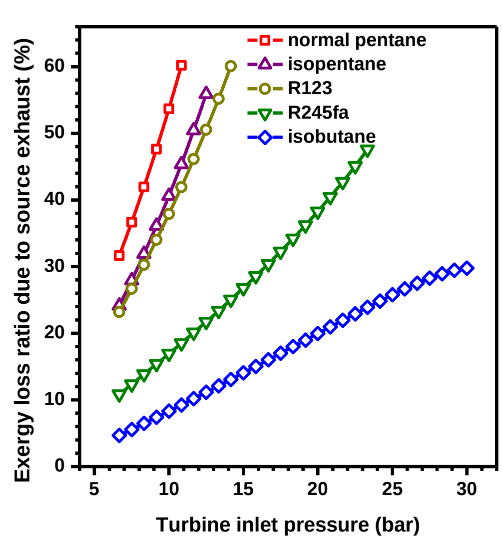

Fig. 3 Exergy loss ratio due to source exhaust versus turbine inlet pressure for various working fluids.

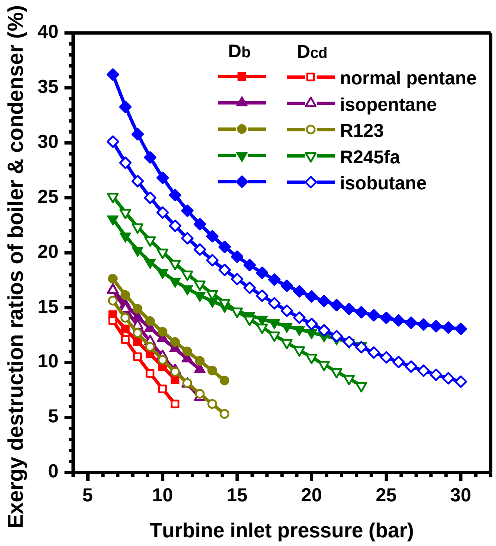

Fig. 4 Exergy destruction ratios of boiler and condenser versus turbine inlet pressure for various working fluids.

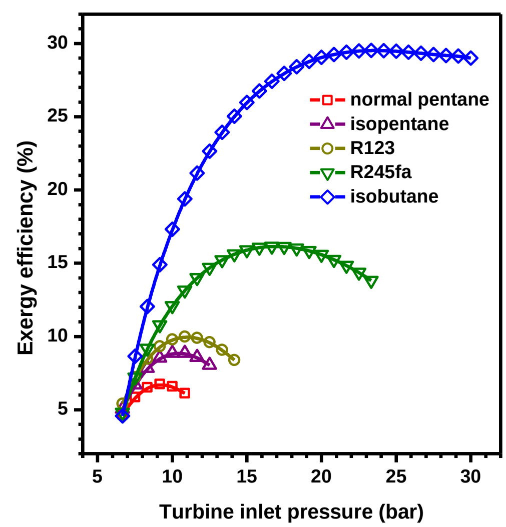

Fig. 6 Dependence of exergy efficiency on turbine inlet pressure for various working fluids.

flow rates to which the heat transfer rates at the boiler and the condenser are directly proportional. For a source mass flow rate of m_s = 1 kg/s, the mass flow rate of working fluid through the turbine ranges roughly from m_t = 0.3 to 1.3 kg/s depending on the working fluid and decreases with TIP. The reason why m_t decreases with TIP is that less fluid can be circulated satisfying the pinch point condition at the boiler. The entrainment ratio of ejector also decreases slightly with TIP. Exergy destruction of the working fluid with lower critical temperature is bigger both at the boiler and the condenser.

Fig. 5 demonstrates the exergy destruction ratios of ejector and evaporator with respect to TIP for various working fluids. The exergy destruction of ejector is more than two times of that of evaporator since the mass flow rate of the ejector is larger than the evaporator. Exergy destruction monotonically decreases as TIP increases both in the ejector and the evaporator since the mass flow decreases with TIP. The dependence of exergy destruction on the critical temperature of the working fluid is not uniform. However, the rate of change of the exergy destruction ratio with respect to TIP is faster for the fluid with higher critical temperature.

Fig. 6 shows the dependence of exergy efficiency of the system on TIP for various working fluids. Total recovered exergy is the sum of net power production and refrigeration exergy output. The effect of raising TIP on the power cycle is twofold: increase of enthalpy drop and decrease of mass flow rate. Because of these competing effects of TIP, the net power production curve shows a hill shape behavior with respect to

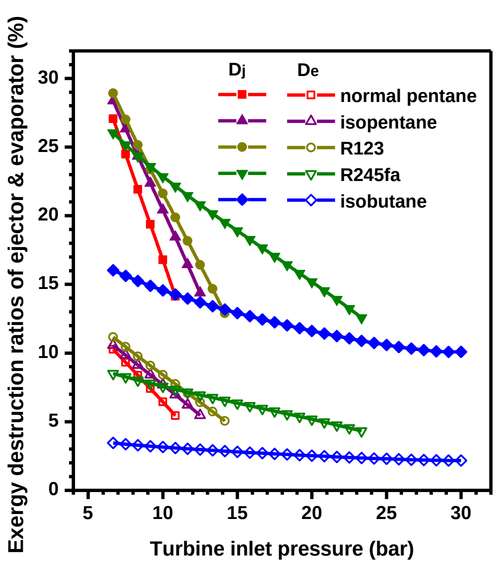

Fig. 5 Exergy destruction ratios of ejector and evaporator versus turbine inlet pressure for various working fluids.

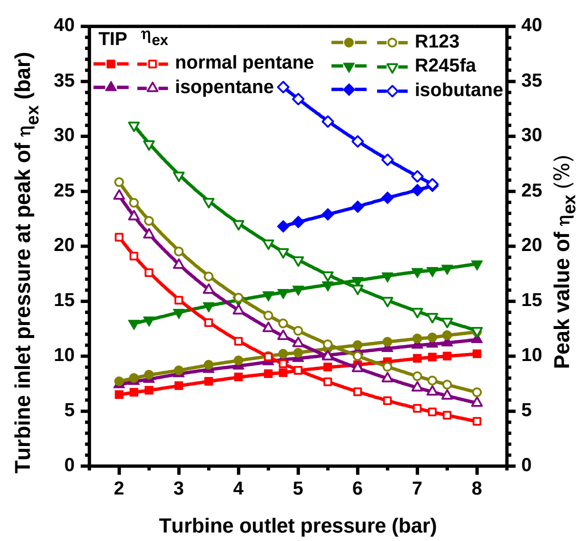

Fig. 7 Dependences of turbine inlet pressure and exergy efficiency at the peaks of η_{ex} on turbine outlet pressure for various working fluids.

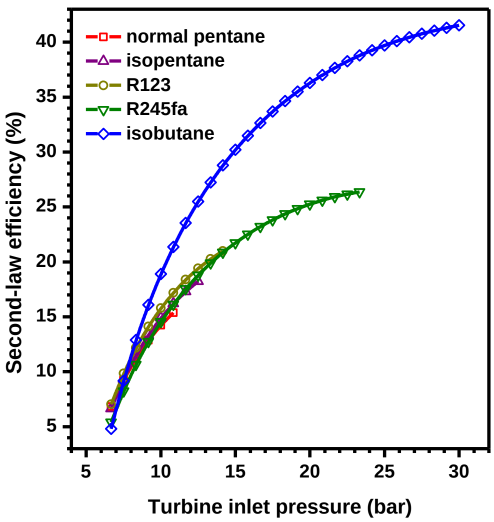

Fig. 8 Dependence of second-law efficiency on turbine inlet pressure for various working fluids.

TIP. While the exergy output associated with refrigeration monotonically decreases with TIP, mainly due to the decrease of the mass flow rate through the turbine. For most of the considered TIP range the net power production outweighs the refrigeration exergy output, so that the exergy efficiency curve has a peak for each working fluid. The exergy efficiency of the working fluid with lower critical temperature is higher than the fluid with higher critical temperature. Dependences of TIP and exergy efficiency at the peaks of the exergy efficiency curve on TOP is plotted in Fig. 7 for various working fluids. The value of TIP giving a peak of exergy efficiency increases almost linearly with TOP. The reason is because the higher inlet pressure is required in the turbine to produce the same or more power matching with the raised outlet pressure. On the contrary, the peak values of the exergy efficiency decrease with TOP. The implication of this result is that the TOP should be kept as low as possible to achieve better exergy efficiency of the system. The TIP and the exergy efficiency at the peaks of exergy efficiency curve are higher for the fluid with low critical temperature for each TOP.

Fig. 8 is the plot of the dependence of the second-law efficiency of the system on TIP for various working fluids. While the exergy efficiency is defined as the ratio of the recovered exergy to the exergy input by the source fluid, the second-law efficiency is defined as its ratio to the net exergy input. Hence the relation of $\eta_{II} = \eta_{ex} / (1 - D_{so})$ holds between the two efficiencies and the second-law efficiency is higher than the exergy efficiency for any system operating condition. The second-law efficiency increases monotonically with respect to TIP for all working fluids, to which the increasing of D_{so} has contributed. For the TIPs sufficiently higher than 6 bar which is equal to the TOP, isobutane gives the highest second-law efficiency of the considered working fluids. Except for isobutane there is only a slight difference among the second-law efficiencies of the working fluids.

IV. Conclusions

Analysis of the exergy performance based on the second law of thermodynamics is carried out for an organic Rankine cycle combined with an exergy refrigeration cycle which cogenerates power and refrigeration. The system is driven by the sensible heat of source fluid which is assumed as water with inlet temperature of 150°C. Five dry fluids are considered as the working fluid. Special attention is paid to the effects of turbine inlet pressure on the exergy loss or destruction at the components of the system and the exergy and the second-law efficiencies of the cycle. The results of the simulation can be summarized as follows.

1) The recovered exergy as power and refrigeration, lost exergy due to source exhaust, destroyed exergy at the boiler, condenser, ejector, and evaporator are dominant over the exergy loss due to coolant exhaust, exergy destruction at the turbine, pump, and expansion valve.

2) For each fluid, the exergy loss due to source exhaust increases almost linearly with turbine inlet pressure (TIP). On the contrary, the exergy destructions of boiler, condenser, ejector, and evaporator decrease with TIP.

3) The exergy efficiency curve has a peak for each working fluid and the exergy efficiency of the working fluid with lower critical temperature is higher than the fluid with higher critical temperature.

4) For each fluid the value of TIP giving peak exergy efficiencies increases almost linearly with turbine outlet pressure (TOP), but the peak value itself decreases with TOP.

5) The second-law efficiency increases monotonically with respect to TIP for all working fluids considered.

Acknowledgment

This paper was supported by Research Fund, Kumoh National Institute of Technology.

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