Effectiveness and Exergy Destruction Analysis of Evaporator in Organic Rankine Cycle

Kyoung Hoon Kim, and Chul Ho Han

***Abstract*---**This paper carries out a performance analysis based on the first and second laws of thermodynamics for source heat exchanger in organic Rankine cycle when the heat source is low-temperature energy in the form of sensible heat. In the analysis, effects of the selection of working fluid and turbine inlet pressure are investigated on the performance of the evaporator including the effectiveness of heat transfer, exergy destruction, pinch point, and second law efficiency. The results show that the heat exchanger effectiveness and the exergy destruction increase but the pinch-point enthalpy increases as the turbine inlet pressure increases.

***Keywords*–**organic Rankine cycle, source heat exchanger, effectiveness, exergy destruction.

I. Introduction

ENERGY demand has been rapidly increasing but the fossil fuel to meet the demand is being drained, an efficient use of low-grade energy source such as geothermal energy, exhaust gas from gas turbine system, biomass combustion, or waste heat from various industrial processes becomes more and more important. The waste heat which is merely discharged due to the lack of effective conversion method contributes to the thermal pollution and environmental problem [1]-[2]. The organic Rankine cycle (ORC) and the power generating system using binary mixture as a working fluid have attracted much attention as they are proven to be the most feasible methods to achieve high efficiency in converting the low-grade thermal energy to more useful forms of energy [3]-[7].

The ORC is commonly accepted as an established technology to convert low grade heat source into electricity. Furthermore, ORCs are designed for unmanned operation with little maintenance. Due to the excellent characteristics, several ORC waste heat recovery plants are already in operation [8]. ORC is a Rankine cycle where an organic fluid is used instead of water as working fluid. Since ORC system exhibits great flexibility, high safety and low maintenance requirements in recovering low-grade heat, it has been widely accepted as a promising technology for conversion of low grade heat into useful work and electricity. Drescher and Bruggemann [9] studied the ORC in solid biomass power and heat plants and they proposed a method to find suitable thermodynamic fluids for ORCs in biomass plants and found that the family of alkylbenzenes showed the highest efficiency. Heberle and Brueggemann [10] reported an analysis of a combined heat and power generation for geothermal re-sources with series and parallel circuits of an ORC. Dai et al [11] identified isobutane and R236ea as efficient working fluids by using a generic optimization algorithm.

Hung et al. [12] examined Rankine cycles using organic fluids which are categorized into three groups of wet, dry and isentropic fluids. They pointed out that dry fluids have disadvantages of reduced net work due to superheated vapor at turbine exit, and wet fluids of the moisture content at turbine inlet, so isentropic fluids are to be preferred. Kim and Perez-Blanco [13] presented a thermodynamic analysis of cogeneration of power and refrigeration based on ORC activated by low-grade sensible energy. Lee and Kim [14] reported energy and exergy analyses for a combined cycle consisting of an ORC and a liquefied natural gas (LNG) Rankine cycle for the recovery of low-grade heat sources and LNG cold energy.

In heat exchangers of power generation cycles for recovery of thermal energy sources in the form of sensible heat energy, assessment of pinch point is somewhat complex. Kim et al. [15] proposed efficient and novel methods for pinch point assessments in source heat exchanger or condenser in ammonia-water based power generation cycles. They also presented the first and second thermodynamic law analysis for heat recovery vapor generator (HRVG) of ammonia-water mixture when the heat source is low- temperature energy in the form of sensible heat [16].

This study presents a performance analysis of the evaporator in organic Rankine cycle based on the first and second laws of thermodynamics for the recovery of low-grade heat source. Parametric study is carried out on the performance of the source heat exchanger depending on the working fluid and turbine inlet pressure.

II. System Analysis

The schematic diagram of ORC is shown in Fig. 1. The working fluid is compressed in pump from a saturated liquid state (state 1) to a compressed liquid state (state 2). The fluid is then heated in the evaporator using the sensible heat from the source fluid to a saturated or superheated vapor state (state 3). After the mechanical energy is obtained (state 4) owing to the expansion process in turbine, the flow enters the condenser and exchanges heat with the coolant, then returns to state 1. The source fluid enters the evaporator with mass flow rate m_s and temperature T_s. On the other hand, working fluid enters the

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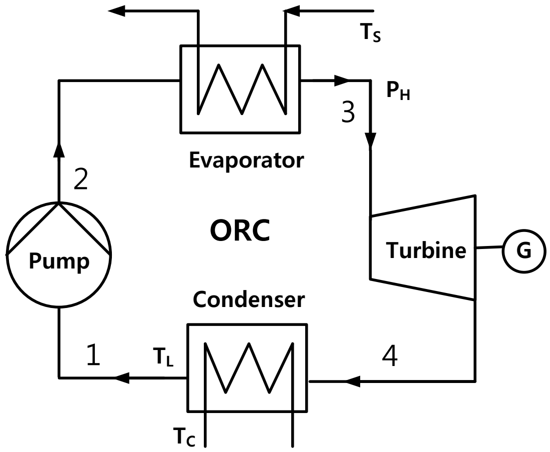

Fig. 1. Schematic diagram of ORC and source heat exchanger

evaporator with mass flow rate m_w temperature T_2, and turbine inlet pressure P_H, is heated by the hot source stream, and then leaves the heat exchanger with turbine inlet temperature T_H=T_3.

The important assumptions in the present cycle analysis are as follows [16]: 1) The fluid flow and heat transfer rates are steady. 2) Heat loss between the system and environment is negligible so that heat transfer occurs only between hot and cold fluid streams in the heat exchangers. 3) Pressure drop due to flows inside heat exchangers is negligible so that the pressure inside a heat exchanger is maintained constant. 4) The kinetic energy and the potential energy changes of the fluids in and out of the heat exchangers are negligible. 5) The longitudinal heat conduction in the tube walls is negligible. 6) The working fluid at the turbine inlet is a saturated vapor.

For a specified mass flow rate of the source fluid m_s, the mass flow rate of the working fluid m_w , the mass flow rate of coolant m_c , and heat transfer at evaporator Q can be determined from the energy balances at the evaporator and condenser and the conditions of pinch temperature difference as follows [12]:

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

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

$$Q = m_w(h_3 - h_2) = m_s c_{ps}(T_s - T_{sout}) \tag{3}$$

where h is the specific enthalpy of the mixture, T_{sout} is the source exhaust temperature, T_{cout} is the coolant outlet temperature, c_p is the isobaric specific heat, and subscripts s, w, and c denote the source, working fluid, and coolant, respectively. And the condition of pinch temperature difference can be determined as the temperature difference between hot and cold streams in a heat exchanger is as same as the prescribed value of ΔT_{pp} [2], [15]:

$$\min[T_{hot} - T_{cold}] = \Delta T_{pp} \tag{4}$$

Let us define as the dimensionless enthalpy at pinch point H_{pp} and enthalpy ratio at pinch point x_{pp} as follows:

$$x_{pp} = \frac{h_{pp} - h_f}{h_g - h_f} \tag{5}$$

$$H_{pp} = \frac{h_{pp} - h_2}{h_3 - h_2} \tag{6}$$

where h_{pp}, h_g, and h_f represent the enthalpies of working fluid at pinch point, saturated liquid and saturated vapor, respectively.

Exergy is a measure of the maximum capacity of a system to perform useful work as it proceeds to a specified final state in equilibrium with its surroundings. Exergy is generally not conserved as energy, but destructed in the system. The specific exergy is defined as [17]:

$$e = h - h_0 - T_0(s - s_0) \tag{7}$$

where s is the specific entropy and the subscript 0 denotes the dead state. The specific exergy of source fluid at temperature T can be evaluated approximately in accordance to the following equation:

$$e = c_s[T - T_0 - T_0 \ln(T / T_0)] \tag{8}$$

The effectiveness ε, second law efficiency η, and exergy destruction D of evaporator are defined as follows:

$$\varepsilon = \frac{T_s - T_{sout}}{T_s - T_2} \tag{9}$$

$$\eta = \frac{m_w(e_3 - e_2)}{m_s(e_s - e_{sout})} \tag{10}$$

$$D = m_s(e_s - e_{sout}) - m_w(e_3 - e_2) \tag{11}$$

In this work, the thermodynamic properties of the working fluids are calculated by the Patel–Teja equation of state [18], [19].

III. Results and Discussions

In the present study, thermodynamic analysis of evaporator in ORC is carried out. Eight fluids of R134a, R152a, propane, isobutane, butane, R245fa, R123, and isopentane are considered. The basic thermodynamic data for the working fluids are listed in ascending order of critical temperature in Table 1 [20]. The source fluid is considered as water at T_s = 150°C with m_s = 1 kg/s. The basic data of the system variables are as follows: condensation temperature T_L = 40°C, coolant temperature T_c = 25°C, dead state temperature T_0 = 298.15 K, pinch temperature difference ΔT_{pp} = 10°C, isentropic pump efficiency η_p = 0.85, isentropic turbine efficiency η_t = 0. The key parameter in this study is the turbine inlet pressure, P_H.

The mass flow ratio is plotted against the turbine inlet pressure in Fig. 2 for various working fluids. The mass flow

TABLE I
BASIC THERMODYNAMIC DATA OF WORKING FLUIDS.

Substance	M(kg/kmol)	T_{cr}(K)	P_{cr}(bar)	ω
R134a	102.031	380.00	36.90	0.239
R152a	66.051	386.60	44.99	0.263
propane	44.096	396.82	42.49	0.152
isobutane	58.123	408.14	36.48	0.177
butane	58.123	425.18	37.97	0.199
R245fa	134.048	427.20	36.40	0.372
R123	136.467	456.90	36.74	0.282
isopentane	72.150	460.43	33.81	0.228

is defined as the ratio of the mass flow rate of working fluid to the source fluid. The mass flow ratio decreases with increasing turbine inlet pressure. It is because as the turbine inlet pressure increases, the corresponding evaporation temperature of working fluid increases, which leads to higher temperature of source exhaust and consequently to lower heat transfer in the evaporator. It can be seen from the figure that there exists an upper limit of turbine inlet pressure for the working fluids with high critical temperature such as R123 and isopentane, where the heat transfer is reduced to zero.

Fig. 3 shows the variations of enthalpy ratio at pinch point in evaporator. As is seen in the Eq. (5), the enthalpy ratio is same as the quality of the fluid when it is higher than zero and lower than unity. When the enthalpy ratio is lower than zero, the lower enthalpy ratio becomes, the greater degree of subcooling of the working fluid becomes. When the enthalpy ratio is greater than unity, the higher the enthalpy ratio becomes, the greater the degree of superheating of the working fluid becomes. As is seen in the figure that for most working fluids the pinch point occurs

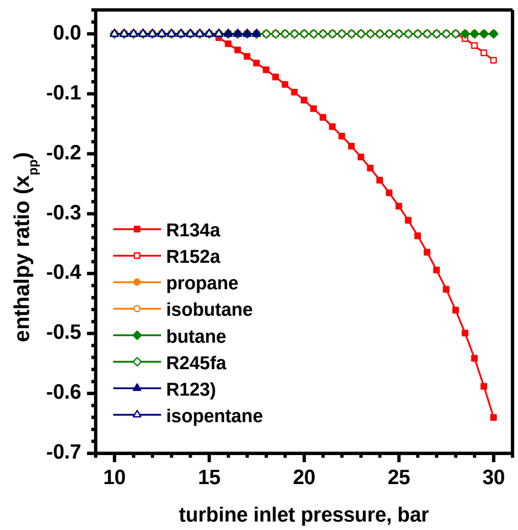

Fig. 3. Variations of enthalpy ratio at pinch point with turbine inlet pressure for various working fluids.

at which the enthalpy ratio is unity so when the working fluid is saturated vapor. But for working fluids with lower critical temperature such as R152 or R134a, the enthalpy ratio decreases with increasing turbine inlet pressure. This means that the pinch point occurs when the working fluid is subcooled.

The dimensionless enthalpy at pinch point which was defined in Eq. (6) is illustrated in Fig. 4 for various working fluids. It can be seen from the figure that the dimensionless enthalpy increases with increasing turbine inlet pressure. It is because as the turbine inlet pressure increases, the corresponding evaporation becomes high, so the potion of heating for subcooled liquid in the total heat transfer in the evaporator increases.

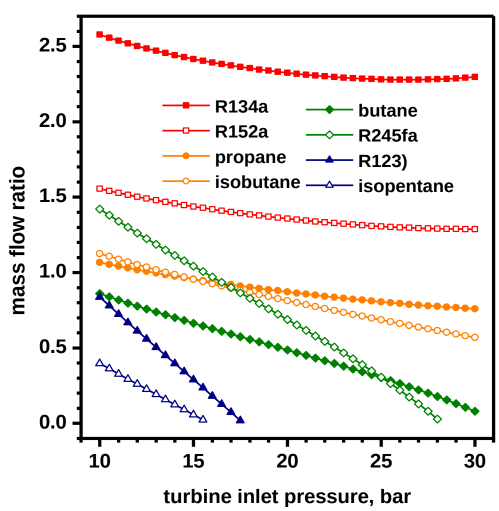

Fig. 2. Variations of mass flow ratio with turbine inlet pressure for various working fluids.

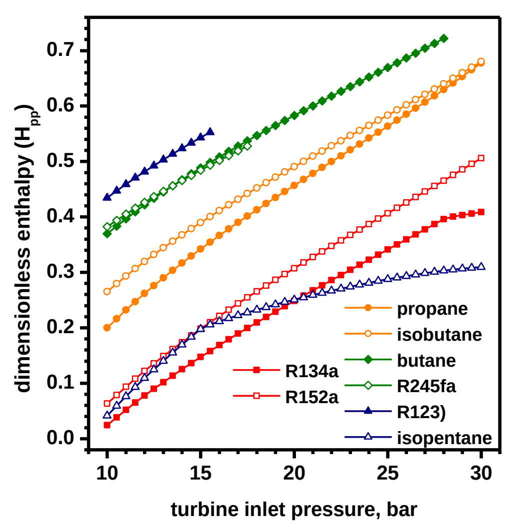

Fig. 4. Variations of dimensionless enthalpy at pinch point with turbine inlet pressure for various working fluids.

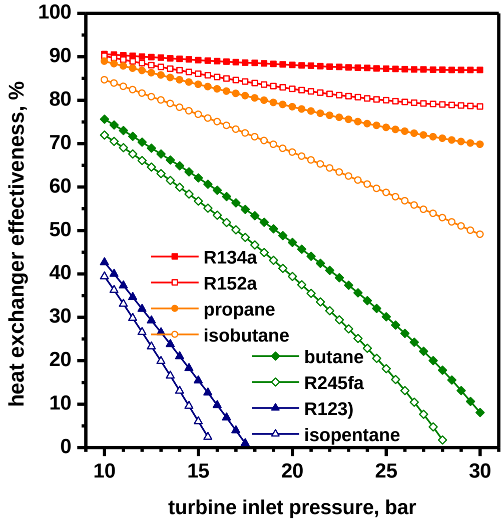

Fig. 5. Variations of heat exchanger effectiveness of evaporator with turbine inlet pressure for various working fluids.

The effectiveness of evaporator is plotted against the turbine inlet pressure in Fig. 5 for various working fluids. It can be seen from the figure that the effectiveness decreases with increasing turbine inlet pressure, because the vaporization heat of working fluid decreases with increasing turbine inlet pressure. For a specified turbine inlet pressure, the effectiveness becomes lower as the critical temperature of working fluid increases, since a high critical temperature of working fluid results in a low heat transfer in the evaporator and consequently a low effectiveness of evaporator.

Fig. 6 shows the exergy destruction at evaporator as a function of turbine inlet pressure for various working fluids. It is worthy to note that in this case the exergy destruction is equals to the product of dead state temperature T_0 and the entropy

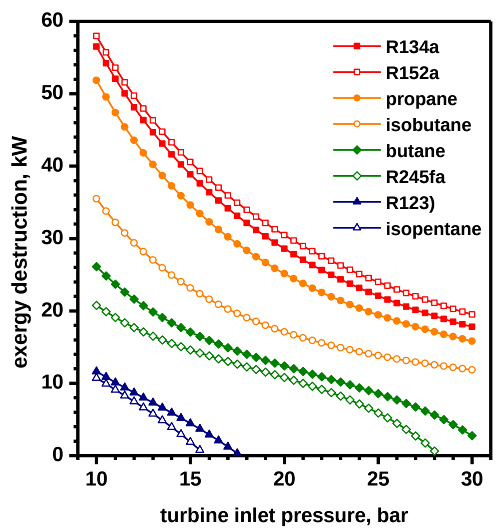

Fig. 6. Variations of exergy destruction with turbine inlet pressure for various working fluids.

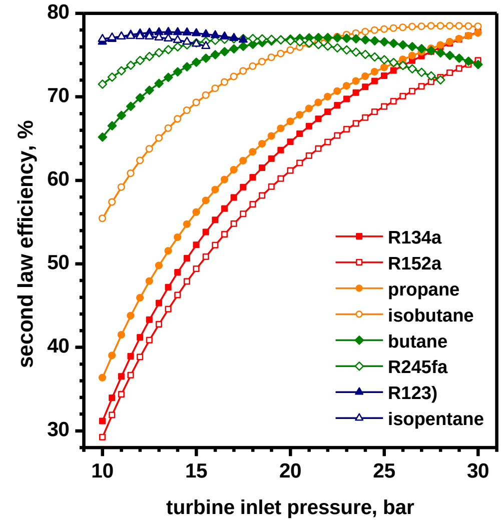

Fig. 7. Variations of second law efficiency with turbine inlet pressure for various working fluids.

generation in the evaporator. The exergy destruction decreases with increasing turbine inlet pressure. Therefore, the higher turbine inlet pressure becomes, the lower irreversibility of the evaporator becomes. It can be also seen that for a specified value of turbine inlet pressure, the exergy destruction at the evaporator shows a decreasing tendency with respect to the critical temperature of working fluid. However, the exergy destruction of R152a is higher than that of R134a.

Fig. 7 shows the second law efficiency of evaporator with respect to turbine inlet pressure for various working fluids. In the range of simulation, the efficiency increases with increasing turbine inlet pressure for the working fluids with low critical temperature such as R134a, R152a, propane and isobutane. However, for the working fluids with high critical temperature such as butane, R245fa, R123, and isopentane, the efficiency firstly increase and reaches a maximum value and then decreases again as the turbine inlet pressure increases. However, it can be observed from the figure that the maximum efficiency values of the working fluids are similar to each other. The optimum turbine inlet pressure for the second-law efficiency becomes higher as the critical temperature of working fluid becomes lower.

IV. Conclusions

A performance analysis of the evaporator in organic Rankine cycle with various working fluids is carried out based on the first and second laws of thermodynamics for the recovery of low grade finite heat source. The results show that the selection of the working fluid and the turbine inlet pressure exhibit significant effects on the thermodynamic performance of the evaporator. Results show that the heat exchanger effectiveness of evaporator or exergy destruction decreases with increasing turbine inlet pressure or critical temperature of working fluid. However, the second law efficiency increases with turbine inlet pressure for the working fluids with lower critical temperature but has a peak value with respect to the turbine inlet pressure for the working fluids of higher critical temperature.

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