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Thermodynamic Performance of a Combined Power and Ejector Refrigeration Cycle

Hyung Jong Ko and Kyoung Hoon Kim

***Abstract*—**In this study thermodynamic performance analysis of a combined organic Rankine cycle and ejector refrigeration cycle is carried out for use of low-grade heat source in the form of sensible energy. Special attention is paid to the effects of system parameters including the turbine inlet temperature and turbine inlet pressure on the characteristics of the system such as ratios of mass flow rate, net work production, and refrigeration capacity as well as the coefficient of performance and exergy efficiency of the system. Results show that for a given source the coefficient of performance increases with increasing of the turbine inlet pressure. However, the exergy efficiency has an optimal condition with respect to the turbine inlet pressure.

***Keywords*—**Coefficient of performance, ejector refrigeration cycle, exergy efficiency, low-grade energy, organic rankine cycle.

I. Introduction

MOST of the low-grade thermal energy is merely discarded, since it is difficult to efficiently convert them into electricity by conventional methods. For the past decades, 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 [1]-[4].

Velez et al. [5], Raj et al. [6], Tchanche et al. [7], and Chen et al. [8] carried out the review of the state of art of research in energy conversion from low-grade energy sources. Drescher and Bruggemann [9] investigated the ORC in solid biomass power and heat plants. 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. Schuster et al. [10] mentioned numerous running applications, such as geothermal power plant, biomass fired combined heat and power plants, solar desalination plants, waste heat recovery or micro CHP. Dai et al. [11] used a genetic optimization algorithm, and identified isobutane and R236ea as efficient working fluids. Ho et al. [12] proposed a novel cycle named as organic flash cycle (OFC). In the cycle heat addition occurs with the cycle working fluid remaining in the liquid state, which would ensure near perfect temperature matching to the source stream. Heberle and Brueggemann [13] investigated the combined heat and power generation for geothermal resources with series and parallel circuits of an ORC.

Low-grade energy sources can be used for the cogeneration of power and refrigeration. This can be achieved by coupling a thermal power cycle with either absorption or ejector refrigeration cycle [14]. Recently, Dai et al. [15] proposed a combined power and refrigeration cycle, which combines the Rankine cycle and the ejector refrigeration cycle. This combined cycle produces both power output and refrigeration output simultaneously and it can be driven by the flue gas of gas turbine or engine, solar energy, geothermal energy and industrial waste heats. Li et al. [16] proposed an organic Rankine cycle with ejector (EORC) for the purpose of increasing the power output capacity and its efficiency. Habibzadeh et al. [17] performed the thermodynamic study of a thermal system which combines an organic Rankine cycle and an ejector refrigeration cycle using classical and finite-size thermodynamics.

In this study the thermodynamic performance of a combined power and ejector refrigeration cycle is carried out for use of low-grade heat source in the form of sensible energy. The effects of the system parameters including the turbine inlet temperature and pressure on the system characteristics such as ratios of mass flow rate, net work production, and refrigeration capacity as well as the coefficient of performance (COP) and exergy efficiency are investigated.

II. System Analysis

The system considered in this work is an organic Rankine cycle combined with an ejector refrigeration cycle and its schematic diagram is shown in Fig. 1. The system consists of a pump, boiler, turbine, ejector, condenser, expansion valve, and evaporator. The working fluid considered in this study is R245fa and its thermodynamic properties are calculated by the Patel-Teja equation of state [18]. The basic data of the working fluid which are needed to apply the Patel-Teja equation are shown in Table I, where M, T_{cr}, P_{cr} and ω are molecular weight, critical temperature, critical pressure, and acentric factor, respectively [19]. The system is driven by the sensible energy supplied by a low-grade heat source. Heat rejection to the coolant and refrigeration space takes place in the condenser and evaporator, respectively.

TABLE I
Basic Data of the Working Fluid

Substance	M (kg/kmol)	T_{cr} (K)	P_{cr} (bar)	ω
R245fa	134.048	427.2	36.40	0.3724

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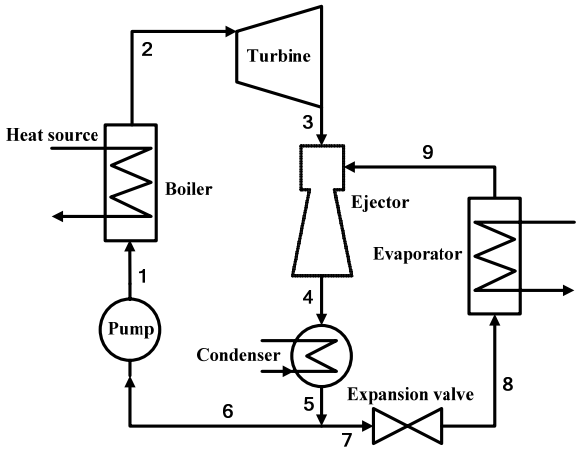

Fig. 1 Schematic diagram of the system

In order to investigate the thermodynamic performance of the system the following simplifications are utilized.

1) The source and coolant fluids are water at temperature of T_s and T_c, respectively.
2) The working fluid enters the turbine as superheated vapor with temperature of T_h and pressure of P_h.
3) The working fluid leaves the condenser as saturated liquid at temperature of T_{cd}.
4) The minimum temperature difference between the hot and cold streams in the boiler and condenser is held at a prescribed value of pinch point temperature difference, ΔT_{PP}.
5) Pressure drop and heat loss in the system are negligible.
6) The pump and turbine have constant isentropic efficiencies of η_p and η_t, respectively.
7) The flow in the ejector is one-dimensional and the velocities at the inlet and outlet are negligible. Mixing of the primary and secondary flows occurs at constant pressure. The effects of irreversibility at the nozzle, mixing, and diffuser sections can be taken into account by using the nozzle, mixing, and diffuser efficiencies of η_n, η_m, and η_d, respectively [15].

Using these assumptions and referring to Fig. 1 the thermodynamic states at 1 to 9 are determined as follows. The points 1 and 2 are determined using P_h, η_p and P_h, T_h, respectively. Similarly the point 3 is determined using the turbine outlet pressure P_m and η_t. The state at the exit 4 and the entrainment ratio $r_{e/t}$ of the ejector are determined iteratively using the conservation equations and P_m, evaporator pressure P_e, and condenser pressure P_{cd} and η_n, η_m, and η_d. It is noted that P_e is the saturation pressure at the evaporator temperature T_e and $r_{e/t}$ is equal to the ratio of mass flow rate of the evaporator to turbine. The working fluids at 5, 6, and 7 are at the same thermodynamic states and are saturated liquids at T_{cd}. The corresponding saturation pressure defines P_{cd}. The relations $P_8 = P_e$ and $h_8 = h_7$ determine point 8. The exit of evaporator 9 corresponds to the saturated vapor at temperature T_e.

It is desirable to circulate more working fluid and less coolant fluid for a given mass flow rate of source fluid, unless the pinch point condition is violated. The ratios of mass flow rate of working fluid and coolant fluid to that of source fluid can be determined from the energy balance and the pinch point condition. Note that the mass flow rate in the condenser is equal to the sum of those in the turbine and evaporator:

$$m_{cd} = m_t + m_e \tag{1}$$

The rates of heat input, net work production and refrigeration output can be calculated from

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

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

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

The rate of exergy input to the system by the source fluid and the exergy output associated with refrigeration can be calculated as

$$E_{in} = m_s c_{ps}\{T_s - T_0 - T_0 \ln(T_s / T_0)\} \tag{5}$$

$$E_e = Q_e(T_0 - T_{cs}) / T_{cs} \tag{6}$$

where the subscript 0 refers to the dead state and T_{cs} is the temperature of the cooled space.

As the system cogenerates power and refrigeration output, the overall performance of the system can be assessed with the coefficient of performance (COP) and the exergy efficiency η_{ex}. These are defined as follows:

$$COP = (W_{net} + Q_e) / Q_{in} \tag{7}$$

$$\eta_{ex} = (W_{net} + E_e) / E_{in} \tag{8}$$

III. Results and Discussions

In this study source fluid is water with temperature of T_s = 150°C. The performance of the system is parametrically investigated for the varying parameter values. The inlet (P_h) and outlet (P_m) pressure of turbine are varied from 5 to 20 bar and from 3 to 4 bar, respectively, while the turbine inlet (T_h) and condenser (T_{cd}) temperature ranging from 100 to 120°C and from 22 to 30°C are considered. Other basic data for analysis are as follows; T_c = 15°C, T_e = -20°C, T_{cs} = -5°C, ΔT_{PP} = 10°C, η_p = 0.7, η_t = 0.85, η_n = 0.95, η_m = 0.95, η_d = 0.95.

In the combined cycle the power generation performance is mainly influenced by the turbine inlet conditions, while the refrigeration characteristics are affected by the conditions at the turbine outlet and condenser. Therefore the former is investigated by varying P_h and T_h, and the latter by varying P_m and T_{cd}.

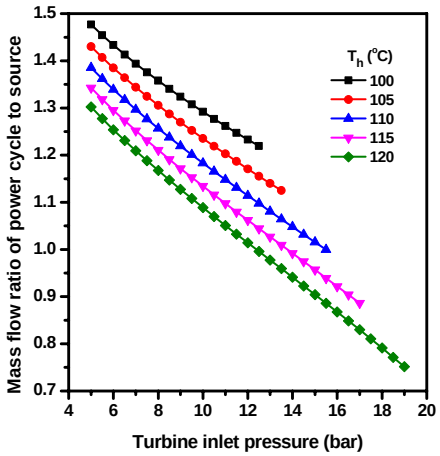

Fig. 2 Dependence of mass flow ratio of power cycle to source on the turbine inlet conditions

Fig. 2 shows the dependence of mass flow ratio of power cycle to source fluid, $r_{t/s}$, on the turbine inlet conditions for P_m = 3 bar and T_{cd} = 30°C. Turbine inlet pressure is limited such that the working fluid enters turbine as a superheated vapor. For example P_h < 12.5 bar for T_h = 100°C. $r_{t/s}$ is an important variable because it characterizes the performance on a base of unit mass of source fluid. As a high P_h or T_h means a big heat load during heating process, $r_{t/s}$ is lower for higher P_h or T_h. The dependence of $r_{t/s}$ on P_h is almost linear but the dependence on P_m or T_{cd} is weak.

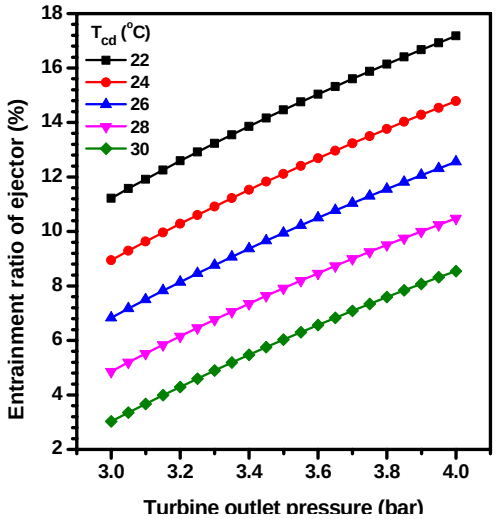

Fig. 3 Dependence of entrainment ratio of ejector on turbine outlet pressure and condenser temperature

Fig. 3 shows the dependence of entrainment ratio of ejector, $r_{e/t}$, on turbine outlet pressure and condenser temperature for P_h = 10 bar and T_h = 110°C. $r_{e/t}$ is defined as the ratio of secondary flow rate to primary flow rate in the ejector, and equal to the ratio of mass flow rate in evaporator to that in turbine. $r_{e/t}$ characterizes the refrigeration performance. $r_{e/t}$ increases almost linearly with P_m since high P_m implies high energy of the primary flow which is needed to entrain the secondary flow. On the contrary high T_{cd} lowers entrainment ratio since it means a high back pressure of ejector resulting in a high compression ratio. The dependence of entrainment ratio on the turbine inlet condition is very weak.

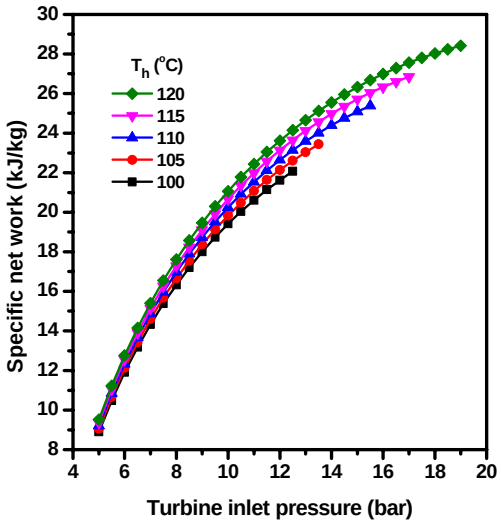

Fig. 4 Dependence of net work per unit mass of working fluid on the turbine inlet conditions

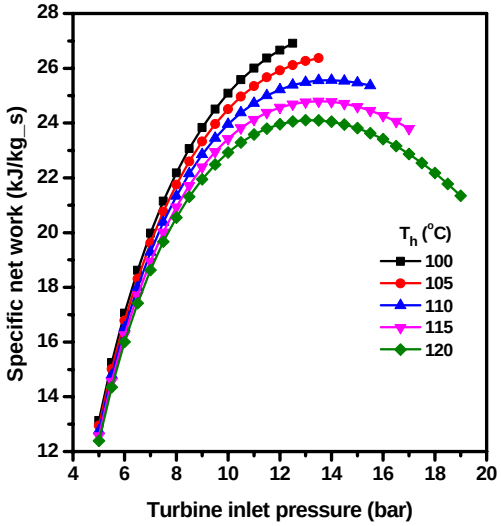

Fig. 5 Dependence of net work per unit mass of source fluid on the turbine inlet conditions

The dependences of the specific net work on the turbine inlet conditions for P_m = 3 bar and T_{cd} = 30°C are given for a unit mass of working fluid, $w_{n/t} = W_{net}/m_t$, in Fig. 4 and for a unit mass of source fluid, $w_{n/s} = W_{net}/m_s$, in Fig. 5, respectively. Note a relation $w_{n/s} = w_{n/t}\, r_{t/s}$. For the pressure range considered, $w_{n/t}$ increases with P_h and T_h, which can be readily expected since the enthalpy drop increases with them for fixed P_m. For fixed T_h, the dependence of $w_{n/s}$ on P_h is similar to $w_{n/t}$ for relatively low P_h range. However it begins to decrease with P_h from peak points. Moreover $w_{n/s}$ is smaller for higher T_h. These different behaviors may be explained through the factor $r_{t/s}$ shown in Fig. 2. For a given low-grade energy source more efficient power

generation is expected if the system is operated for the TIP giving peak value of $w_{n/s}$.

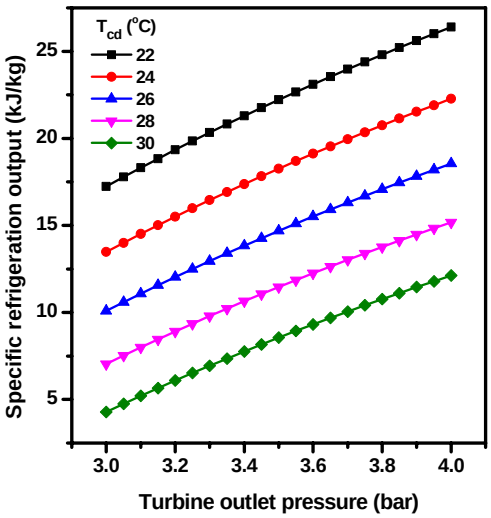

Fig. 6 Refrigeration output per unit mass of working fluid versus turbine outlet pressure and condenser temperature

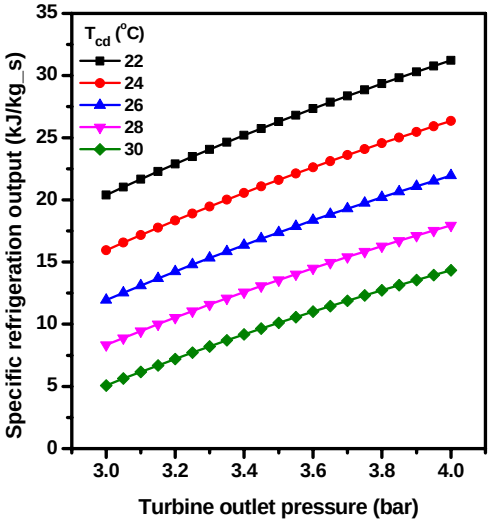

Fig. 7 Refrigeration output per unit mass of source fluid versus turbine outlet pressure and condenser temperature

Figs. 6 and 7 show the dependence of the specific refrigeration output on turbine outlet pressure and condenser temperature for P_h = 10 bar and T_h = 110°C for a unit mass of working fluid, $q_{e/t} = Q_e/m_t$ and for a unit mass of source fluid, $q_{e/s} = Q_e/m_s$, respectively. Note that $q_{e/s}$ is $q_{e/t}$ multiplied by $r_{t/s}$. Contrary to the specific net works, $q_{e/t}$ and $q_{e/s}$ show the qualitatively same behavior as the entrainment ratio, because the evaporator temperature is fixed. They increase almost linearly with respect the increase of P_m and are smaller at higher condenser temperatures. Raising the turbine outlet temperature could give more refrigeration output with a sacrifice of the reduced power generation due to the decrease of enthalpy drop in the turbine. In spite of the low entrainment ratio ranging from 3 to 18%, the refrigeration out is comparable to the power output.

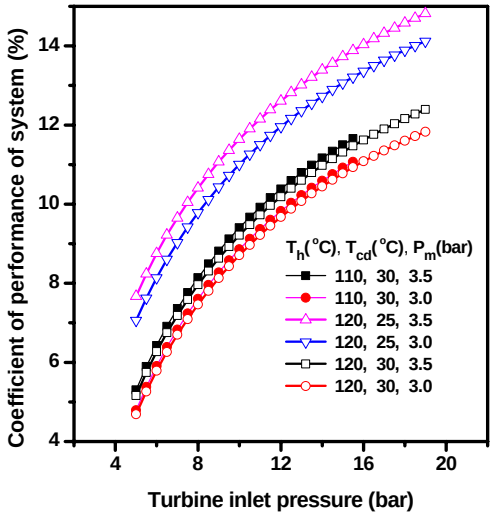

Fig. 8 Variations of the coefficient of performance with respect to the conditions of turbine and condenser

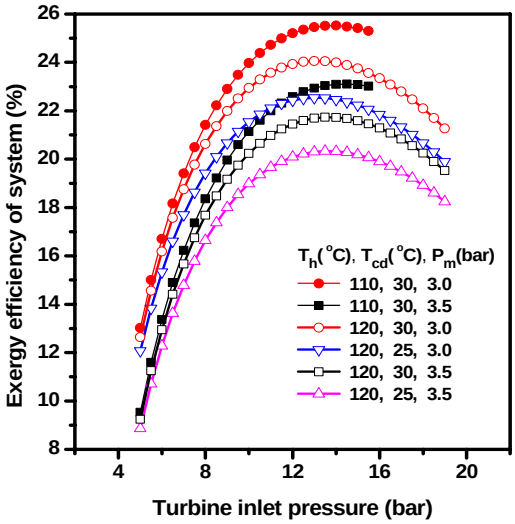

Fig. 9 Variations of the exergy efficiency with respect to the conditions of turbine and condenser

Finally the overall performance of system is investigated in terms of the coefficient of performance (*COP*) and exergy efficiency for the fixed source temperature. Fig. 8 shows the variations of *COP* with respect to the conditions of turbine and condenser. For the range of P_h considered *COP* increases with P_h. *COP* is higher for lower T_{cd} because both power and refrigeration output are larger at lower T_{cd}. On the while, *COP* is higher for higher P_m, as the increased refrigeration output overweighed the negative effect of reduced enthalpy drop. Although the difference is small *COP* is higher for lower T_h.

Variations of the exergy efficiency with respect to the conditions of turbine and condenser are shown in Fig. 9. In contrast to *COP*, exergy efficiency curves show bell shapes with respect to P_h, which implies the existence of P_h value for the optimal utilization of given energy source. The exergy

efficiency is higher for lower T_h, lower P_m, and higher T_{cd}, when other parameters are fixed.

IV. Conclusions

In this study, the performance of a combined power and ejector refrigeration cycle is thermodynamically investigated. The main results can be summarized as follows.

- Entrainment ratio of ejector increases almost linearly with turbine outlet pressure and is higher for lower condenser temperature.
- The coefficient of performance increases with turbine inlet pressure and is higher for lower condenser temperature and turbine outlet pressure.
- The exergy efficiency has an optimal condition with respect to turbine inlet pressure for a fixed source temperature. The exergy efficiency is higher for lower turbine inlet temperature, lower turbine outlet pressure, and higher condenser temperature.

Acknowledgment

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