Exergy Analysis of a Combined Power and Ejector Refrigeration Cycle

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Abstract—Assessment of exergetical performance of a combined organic Rankine cycle and ejector refrigeration cycle is carried out for the effective recovery of low-temperature waste heat. Effects of the turbine inlet and outlet pressure, and turbine inlet and condenser temperature are parametrically investigated on the exergy destruction (anergy) ratio of each component of system and exergy efficiency of the combined cycle. Results show that injector and condenser have second highest anergy after the boiler and source exhaust. The exergy efficiency can have peak values with respect to turbine inlet pressure, while it decreases monotonically with turbine outlet pressure and condenser temperature.

Keywords—energy, cogeneration, ejector refrigeration cycle, exergy efficiency, organic Rankine cycle.

I. INTRODUCTION

There has been an increasing interest in the methods to effectively convert the low-temperature waste heat into useful forms of energy. The waste heat which is merely discharged due to the lack of effective conversion method contributes to the thermal pollution and environmental problem [1]. The organic Rankine cycle (ORC) has been considered as one of the most feasible methods to achieve high efficiency in recovering low-temperature waste heat [2]. Among other studies, [3], [4] and [5] carried out the reviews on the state of the art of research in energy conversion from low-grade energy sources. Velez et al. [6] conducted a comparative study of working fluids for ORC operating at low temperature. Schuster et al. [7] presented a state of the art of ORC applications such as geothermal power plants, biomass fired combined heat and power plants, solar desalination plants, waste heat recovery and micro cogeneration of heat and power. Wang et al. [8] proposed a theoretical thermal efficiency model based on an ideal ORC to analyze the influence of working fluid properties.

One of the advantages of ejector refrigeration cycle (ERC) is its comparatively simple structure, which is the result of replacing the compressor of conventional refrigeration cycle by ejector with no moving parts. In addition ERC can be driven by low-grade energy sources using organic fluid as working fluid. ERC can be used for the cogeneration of power and refrigeration by combining it with various power cycles. Dai et al. [9] proposed a novel combined power and refrigeration cycle, in which a turbine is added between boiler and ejector of ERC. An exergy analysis and a parametric analysis of the combined cycle using R123 as working fluid were performed to guide the thermodynamic improvement of this cycle and to evaluate the effects of the key parameters on the cycle performance. This combined cycle was investigated by Zheng and Weng [10] for the working fluid of R245fa without considering the exergy loss of the heat addition process. The modified version of the combined cycle of [9] was studied by Wang et al. [11] for the same fluid, in which only a part of vapor was extracted from the turbine in the midst of expansion to the ejector. This modified version was also analyzed by Habibzadeh et al. [12] for different working fluid.

For improving the performance of a thermodynamic cycle, exergy analysis is regarded as an appropriate tool since it can locate and estimate the waste and loss of useful energy of a cycle [13]. In this study an assessment of exergetical performance of a combined power and ejector refrigeration cycle which has been investigated by [9], [10] is carried out for the effective utilization of low-temperature waste heat with R245fa as working fluid. The effects of the system parameters including the turbine inlet temperature and pressure are investigated parametrically on the anergy ratio in each component of system and the exergy efficiencies of cycle.

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 system consists of a pump, boiler, turbine, ejector, condenser, expansion valve, and evaporator. The working fluid of the combined cycle is R245fa and its thermodynamic properties are calculated by the Patel-Teja equation of state [14], [15]. The basic data of the working fluid needed when applying the Patel-Teja equation are listed in TABLE 1, where $M$, $T_c$, $P_c$, and $\omega$ are molecular weight, critical temperature, critical pressure, and acentric factor, respectively [16]. The combined cycle 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.

In this study the following assumptions are used for the
1) The working fluid enters the turbine as superheated vapor, and leaves the condenser and evaporator as saturated liquid and saturated vapor, respectively.
2) The temperature difference between the hot and cold streams in the boiler and condenser is higher than the pinch point temperature difference.
3) Pressure drop and heat loss in the system are negligible.
4) The pump and turbine have constant isentropic efficiency.
5) The flow in the ejector is one-dimensional and the mixing and diffuser sections can be taken into account by using their efficiencies.

Using these assumptions and referring to Fig. 1 the thermodynamic states at 1 to 9 are determined for given condenser and evaporator temperatures. In this procedure the exit state 4 and entrainment ratio of the ejector are determined iteratively from the conservation equations of mass, momentum, and energy.

The ratios of mass flow rate of working fluid and coolant fluid to that of source fluid, $m_s$, are determined from the energy balance and pinch point condition. Note that the mass flow rate in the condenser is equal to the sum of those in the turbine and evaporator.

$$m_{cd} = m_t + m_e$$  \hspace{1cm} (1)

The ratio, $r_{exh} = m_e/m_t$, is called the entrainment ratio of ejector.

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

$$Q_e = m_e (h_g - h_e)$$ \hspace{1cm}  \hspace{1cm} (2)

$$W_{net} = W_i - W_p = m_t [(h_i - h_f) - (h_i - h_e)]$$ \hspace{1cm}  \hspace{1cm} (3)

$$Q = m_t (h_i - h_e) - m_e (h_g - h_e)$$ \hspace{1cm}  \hspace{1cm} (4)

The specific exergy $e$ and the rates of exergy input to the system, loss due to the exhaust of source fluid, and exergy output associated with refrigeration are written as

$$e = h - h_0 - T_0 (s - s_0)$$  \hspace{1cm} (5)

$$E_{in} = m_s c_p (T_i - T_0 - T_0 \ln (T_i/T_0))$$ \hspace{1cm}  \hspace{1cm} (6)

$$E_{exh} = m_s c_p (T_{exh} - T_0 - T_0 \ln (T_{exh}/T_0))$$ \hspace{1cm}  \hspace{1cm} (7)

$$E_{e} = Q_e (T_0 - T_{exh})/T_{ex}.$$ \hspace{1cm} (8)

Here $h$, $s$, and $c_p$ mean the specific enthalpy, entropy, and isobaric specific heat. The subscript 0 refers to the environment state, and $T_i$, $T_{exh}$ and $T_{ex}$ are the temperatures of the source inlet, source exhaust and cooled space, respectively.

The rate of exergy destruction or anergy of adiabatic system can be calculated from the difference of the rate of exergy input and output. At the boiler, for example,

$$D_b = E_{in} - E_{exh} + m_e (e_1 - e_2).$$ \hspace{1cm} (9)

The anergy ratio of each component is defined as the ratio of its anergy to exergy input by the source fluid given in (6).

Two kinds of expressions for the exergy efficiency of the cycle defined by

$$\eta_{2a} = (W_{net} + E_e)/E_{in},$$ \hspace{1cm} (10)

$$\eta_{2b} = (W_{net} + E_e)/(E_{in} - E_{exh}),$$ \hspace{1cm} (11)

Are considered; one is based on the exergy input, the other net exergy delivered to the system through the boiler. $\eta_{2a}$ represents an estimation of exergy efficiency by assuming that exergy of the exhaust source is merely discharged to environment. On the contrary $\eta_{2b}$ is an optimistic estimation with a hope that the exergy of exhaust can be fully recovered. Note that two efficiencies are equal if the source exhaust temperature is equal to that of environment.

III. RESULTS AND DISCUSSIONS

In this study the exergetical performance of the combined power and ejector refrigeration cycle using R245fa as working fluid is investigated for the varying parameter values. Water is considered as source fluid with inlet temperature of 150°C. The inlet and outlet pressure of turbine are varied from 5 to 20 bar and from 3 to 4 bar, respectively, while the turbine inlet and condenser temperature ranging from 100 to 120°C and from 22 to 30°C are considered. Other basic data for analysis 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: 10°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.

<table>
<thead>
<tr>
<th>Substance</th>
<th>$M$ (kg/kmol)</th>
<th>$T_e$ (K)</th>
<th>$P_e$ (bar)</th>
<th>$\omega$</th>
</tr>
</thead>
<tbody>
<tr>
<td>R245fa</td>
<td>134.048</td>
<td>427.20</td>
<td>36.40</td>
<td>0.3724</td>
</tr>
</tbody>
</table>

**TABLE I: BASIC DATA FOR THE WORKING FLUID**
Fig. 2 shows the dependence of anergy ratio of source exhaust on turbine inlet pressure (TIP) and turbine inlet temperature (TIT), which is the portion of exergy lost by discharging to the environment. The range of TIP is restricted such that the working fluid enters the turbine as a superheated vapor. The anergy ratio of source exhaust almost linearly increases with TIP for fixed TIT. Although the difference is not large, anergy ratio of source exhaust is higher for higher TIT with the same TIP.

Fig. 3 shows the dependence of anergy ratio of boiler on TIP for various TITs. The anergy ratio of boiler is comparable in magnitude with that of source exhaust but it exhibits an opposite behavior. It is smaller for higher TIP or higher TIT. Roughly speaking the sum of anergy ratios of source exhaust and boiler is about 50%. In other words half of the exergy input by the source is destroyed or lost during the vapor generation process.

The anergy ratio of turbine is plotted as a function TIP for various TITs in Fig. 4. It increases with TIP for fixed TIT and reaches a peak value and then decreases. Even though the ratio is less than 5% over the range considered, more exergy is destroyed in the turbine as the TIT is raised.

Dependences of anergy ratio of ejector on TIP and TIT, and turbine outlet pressure (TOP) and condenser temperature are shown in Fig. 5 and Fig. 6, respectively. Like in the boiler, anergy ratio of ejector decreases as TIP or TIT is raised. This behavior is identical to the behavior that higher thermal efficiency results for higher TIP or TIT.

As seen in Fig. 6, the anergy ratio of ejector for TIP of 10 bar and TIT of 120°C increases almost linearly with TOP and is higher for lower condenser temperature. This behavior is a result of mass effect since the anergy ratio of ejector is proportional to the entrainment ratio of the ejector. For fixed
TIP and TIT, the entrainment ratio of ejector shows a nearly linearly increasing behavior with respect to TOP and is higher for low condenser temperatures [17].

Fig. 7 shows the dependence of anergy ratio of condenser on TOP and condenser temperature for TIP of 10 bar and TIT of 120°C. As for injector for fixed turbine conditions, anergy ratio of condenser increases almost linearly with increasing TOP. This is because the ratio of mass rate of condenser to source fluid exhibits such behavior. By contrast, anergy ratio of condenser is higher for higher condenser temperature since the irreversibility increases with the temperature difference in the heat exchange process. Recall that the coolant temperature is held constant at 15°C.

Finally the variations of two kinds of exergy efficiencies defined in (10) and (11) are shown in Fig. 8 with respect to TIP and TIT, and in Fig. 9 with respect to TOP and condenser temperature, respectively. The efficiency $\eta_{2a}$ represents the exergy efficiency of the cycle based on the exergy input of source fluid. In this case the exergy of the source exhaust is considered as a loss. While the efficiency $\eta_{2b}$ is based on the exergy actually delivered to the system by the source fluid. The latter is intrinsically higher than the former and the difference becomes larger as the anergy ratio of source exhaust gets higher. Therefore their difference magnifies with increasing TIP, while it remains almost unchanged with respect to TOP or condenser temperature. $\eta_{2a}$ has peak values at some TIPs for each TIT. Though $\eta_{2b}$ is monotonically increasing in the considered range of TIP, it is also expected to have peak values if the source temperature is further raised.

Both $\eta_{2a}$ and $\eta_{2b}$ for fixed turbine inlet conditions decrease with increasing TOP because the anergy of ejector and condenser exhibit the similar behavior. They are lower for
higher condenser temperatures because of the larger temperature difference in the condenser.

IV. CONCLUSIONS

An assessment of exegetical performance of a combined power and ejector refrigeration cycle is carried out for the effective recovery of low-temperature waste heat. The cycle uses R245fa as working fluid and water at 150°C as source fluid. The effects of turbine conditions and condenser temperature on the exergy destruction ratio of system component and cycle exergy efficiency are investigated. Results show that the biggest exergy destruction occurs in the heat addition process and has an almost 50% of the total exergy destruction of the cycle. The result that the exergy efficiency can have peak values with respect to turbine inlet pressure may be a useful guide to the cycle operation for the effective recovery of low temperature waste heat. A further study of the effects of system parameters such as the source temperature and evaporator temperature and the selection of working fluid on the exergetical performance of the combined cycle is required.

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