Multi-Objective Optimization of Organic Rankine Cycle for Low-Grade Waste Heat Recovery

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Abstract. The organic Rankine cycle (ORC) is considered as one of the most viable technology to recover low-grade waste heat. A multi-objective optimization model is established to simultaneously derive the maximum exergy efficiency and the minimum electricity production cost (EPC) of a specific ORC system by employing the genetic algorithm (GA). Evaporation temperature and condensation temperature are selected as decision variables. At first, variations of exergy efficiency and EPC with evaporation temperature and condensation temperature are investigated respectively using R245fa, R245ca, R600, R600a, R601 and R601a as working fluids. Subsequently, a multi-objective optimization is performed and the Pareto frontiers for various working fluids are obtained. Results indicate that performance of the specific ORC system with R245fa as working fluid is better that with other working fluids.

1 Introduction

The global primary energy consumption and carbon emissions in recent decades are shown in Fig. 1 and Fig. 2, respectively [1]. Obviously, the global primary energy consumption and the carbon dioxide emissions present a gradual increasing trend, which would result in energy shortage and severe environmental issues. Therefore, it is essential to exploit new energy resource and implement energy conservation and emission reduction.

Fig. 1. The global primary energy consumption in recent decades.

It is reported that the low-grade waste heat directly discharged to environment accounts for 50% or more of the total heat produced in industry [2]. Afterwards, more and more scholars dedicated to waste heat recovery and put forward some solutions, such as Kalina cycle, organic Rankine cycle (ORC), Goswami cycle, etc [3]. Afterwards, ORC was proved to be a promising and superior technology to effectively convert low-grade waste heat into power, and applied in many fields, including solar, geothermal, biomass, ocean thermal energy and waste heat recovery [4-5].

In the past few decades, an enormous amount of researches have been carried out on ORC, focusing on working fluid selection [6-7], expander design [7-8], cycle configuration modification [9-10], optimum design of heat exchanger [11-12], system optimization[13-14], and experimental research on ORC system [15-16], etc.

Fig. 2. The carbon emissions in recent decades.

Regarding to optimization researches on ORC, there are single-objective and multi-objective optimizations based on different evaluation criteria. Generally, the evaluation criteria could be categorized into three classes which are thermodynamic indicators including net power output, thermal efficiency, exergy efficiency [17]; economic indicators including electricity production cost (EPC) [9, 18-20], levelized energy cost (LEC) [20], specific investment cost (SIC) [21], net present value (NPV), depreciated payback period (DPBP) [22]; and environmental impact through life cycle assessment (LCA) [23-24].

In this paper, a multi-objective optimization model was established to evaluate the effects of evaporation temperature and condensation temperature on exergy

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efficiency and EPC of a specific ORC system for waste heat recovery. And genetic algorithm (GA) was employed to derive the Pareto frontier. R245fa, R245ca, R600, R600a, R601 and R601a were selected as working fluids to analyse thermodynamic and economic performances of the ORC system.

2 Description and modeling of the ORC system

2.1 System description
The ORC mainly comprises of preheater, evaporator, turbine, condenser, and pump, as described in Fig.3. The high temperature and high pressure vapor expands and converts thermal energy into power in turbine, driving the generator to produce electricity. Then the low pressure vapor exiting from the turbine is condensed into liquid in condenser and dissipates heat to cooling water. Subsequently, the low temperature and low pressure liquid is pumped into preheater and evaporator where it is heated into saturated liquid and saturated vapor in sequence by hot water. Then a new cycle begins. The corresponding T-s diagram of these thermodynamic processes is shown in Fig.4.

2.2 Thermodynamic modeling
For the sake of simplify, there are some assumptions for the ORC mentioned in the research, which are as follows:
- The ORC system is under stable subcritical condition.
- Heat dissipation to the environment from heat exchangers and pipelines are ignored.
- Heat transfer processes in all heat exchangers are isobaric, and pressure drop in pipelines is regarded to be zero.

On the basis of the aforementioned analysis, the turbine power output can be calculated by

$$W_{turb} = m_{wf}(h_1 - h_2)$$

(1)

The turbine isentropic efficiency is expressed as

$$\eta_{turb} = \frac{(h_1 - h_2) / (h_1 - h_{2s})}{(h_1 - h_2)}$$

(2)

Where, $h_{2s}$ is the enthalpy at the turbine outlet through isentropic expansion.

The exergy destruction rate in turbine is given by

$$i_{turb} = \dot{E}_1 - \dot{E}_2 - W_{turb}$$

(3)

The heat transfer rate released from working fluid to cooling source is calculated by

$$\dot{Q}_{cond} = m_{wf}(h_2 - h_4)$$

(4)

The exergy destruction rate in condenser is given by

$$i_{cond} = \dot{E}_2 - \dot{E}_4 + \dot{E}_{10} - \dot{E}_{12}$$

(5)

The pump power consumed is given by

$$W_{pump} = m_{wf}(h_5 - h_4)$$

(6)

The pump isentropic efficiency is expressed as

$$\eta_{pump} = \frac{(h_{5s} - h_4) / (h_5 - h_4)}{(h_{5s} - h_4)}$$

(7)

Where, $h_{5s}$ is the enthalpy at the pump outlet through isentropic compression.

The exergy destruction rate in pump is given by

$$i_{pump} = \dot{E}_4 - \dot{E}_5 + W_{pump}$$

(8)

The heat transfer rates absorbed from heat source in preheater and evaporator are calculated respectively by

$$\dot{Q}_{preh} = m_{wf}(h_6 - h_5)$$

$$\dot{Q}_{evap} = m_{wf}(h_1 - h_6)$$

(9)

The pinch point temperature differences in condenser and evaporator are defined as

$$\Delta T_{pp,cond} = T_3 - T_{11}$$

$$\Delta T_{pp, evap} = T_5 - T_6$$

(10)

The exergy destruction rates in preheater and evaporator are given by

$$i_{preh} = \dot{E}_5 - \dot{E}_6 + \dot{E}_5 - \dot{E}_6$$

$$i_{evap} = \dot{E}_6 - \dot{E}_7 + \dot{E}_7 - \dot{E}_8$$

(11)

$$\dot{E}_4 = m_{wf}[(h_1 - h_{amb, wf}) - T_{amb}(s_1 - s_{amb, wf})]$$

(12)
Where, \( h_{\text{ambi},\text{w}} \) and \( s_{\text{ambi},\text{w}} \) are enthalpy and entropy of working fluid at state \( i (i = 1 \sim 6) \) under ambient conditions, respectively; \( T_{\text{ambi}} \) is ambient temperature.

\[
E_i = m_i \left[ (h_i - h_{\text{ambi},h}) - T_{\text{ambi}}(s_i - s_{\text{ambi},h}) \right] \tag{13}
\]

Where, \( h_{\text{ambi},h} \) and \( s_{\text{ambi},h} \) are the enthalpy and entropy of heat source at state \( i (i = 7 \sim 9) \) under ambient conditions, respectively.

\[
E_i = m_i \left[ (h_i - h_{\text{ambi},c}) - T_{\text{ambi}}(s_i - s_{\text{ambi},c}) \right] \tag{14}
\]

Where, \( h_{\text{ambi},c} \) and \( s_{\text{ambi},c} \) are the enthalpy and entropy of cooling source at state \( i (i = 10 \sim 12) \) under ambient conditions, respectively.

The net power output of the ORC system is given by

\[
W_{\text{net}} = W_{\text{turb}} - W_{\text{pump}} \tag{15}
\]

Then the total exergy destruction rate of the ORC system is calculated by

\[
\dot{I}_{\text{tota}} = \dot{I}_{\text{turb}} + \dot{I}_{\text{cond}} + \dot{I}_{\text{pump}} + \dot{I}_{\text{preh}} + \dot{I}_{\text{evap}} \tag{16}
\]

The exergy efficiency of the ORC system is defined as

\[
\eta_{\text{exer}} = \frac{W_{\text{net}}}{W_{\text{net}} + \dot{I}_{\text{tota}}} \tag{17}
\]

### 2.3 Economic modeling

In the economic analysis, the module costing technique, proposed by Guthrie [25] to evaluate the preliminary cost of a chemical plant, is employed in the economic modelling of the ORC system [19-20]. Furthermore, EPC is selected to evaluate the economic performance of the ORC system. The investment costs of the major equipment are considered in the following analysis.

The total investment cost of the ORC system is defined as the sum of the bare module costs of all components.

\[
C_{\text{tot}} = C_{\text{BM,preh}} + C_{\text{BM,evap}} + C_{\text{BM,turb}} + C_{\text{BM,cond}} + C_{\text{BM,pump}} \tag{18}
\]

The bare module cost, including direct cost and indirect cost, is defined as

\[
C_{\text{BM}} = C_p F_{\text{BM}} \tag{19}
\]

Where, \( C_p \) is the purchased cost under base conditions, \( F_{\text{BM}} \) is the bare module cost factor.

\[
\log_{10} C_p = K_1 + K_2 \log_{10}(A) + K_3 \left[ \log_{10}(A) \right]^2 \tag{20}
\]

Where, \( A \) is the heat transfer area of heat exchanger or capacity of turbine and pump; \( K_1, K_2, K_3, B_1, B_2 \) are constants which are illustrated in Table 1; \( F_M \) is the material factor; \( F_p \) is the pressure factor, given by

\[
F_{\text{BM}} = B_1 + B_2 F_M F_p \tag{21}
\]

\[
\log_{10} F_p = C_1 + C_2 \log_{10}(P) + C_3 \left[ \log_{10}(P) \right]^2 \tag{22}
\]

Where, \( P \) is the operating pressure in bar; \( C_1, C_2, C_3 \) are constants, listed in Table 1.

Taking inflation into consideration, the Chemical Engineering Plant Cost Index (CEPCI) is adopted to adjust the cost changes, and the total capital cost of the ORC system in 2018 is derived [26]

\[
C_{\text{tot,2018}} = C_{\text{tot,2012}} \cdot \text{CEPCI}_{2018} / \text{CEPCI}_{2012} \tag{23}
\]

Where, \( \text{CEPCI}_{2012} = 584.6 \), \( \text{CEPCI}_{2018} = 603.1 \) [27].

In the field of waste heat recovery, the cost of waste heat source is free [28]. However, the operation and maintenance cost is considered. The EPC is defined as [10, 18-19]

\[
EPC = \left( CRF \cdot C_{\text{tot,2018}} + C_{\text{OM}} \right) / (W_{\text{net}} \cdot h_{\text{fl}}) \tag{24}
\]

Where, \( C_{\text{OM}} \) is the operation and maintenance cost and assumed to be 1.5\% of \( C_{\text{tot,2018}} \); \( h_{\text{fl}} \) is the annual operation time of the ORC system, which is 7500 h; \( CRF \) is the capital recovery cost (CRF), indicated as

\[
CRF = y(1+y)^{LT} \left[ (1+y)^{LT} - 1 \right] \tag{25}
\]

Where, \( y \) is the interest rate, 5\%; \( LT \) is the life time of the ORC system, 20 years.

### 2.4 Multi-objective optimization modeling

#### 2.4.1 Objective functions

In order to evaluate the thermodynamic and economic performances of a specific ORC system, a multi-objective optimization is carried out to simultaneously derive the maximum exergy efficiency and the minimum EPC. Decision variables involve evaporation temperature and condensation temperature.

The multi-objective functions are defined as

\[
\text{Minimize } F_1(X) = -\eta_{\text{exer}} \tag{26}
\]

\[
F_2(X) = EPC \
\]

\[
T_{\text{cond}} < T_{\text{evap}} < T_{\gamma} \tag{27}
\]

Subject to \( T_{\text{evap}} < T_{\text{crit}} \)

\[
T_{\text{cond}} > T_{\text{ambi}} \tag{28}
\]

Where, \( T_{\text{crit}} \) is the critical temperature of working fluid.
Table 1. Parameter values for equations in economic modelling [26].

| Parameter | K1    | K2    | K3    | C1     | C2    | C3    | B1 | B2 | FM | FBM |
|-----------|-------|-------|-------|--------|-------|-------|----|----|----|-----|
| Preheater | 4.3247| -0.3030| 0.1634| 0.0388 | -0.1127| 0.0818| 1.63| 1.66| 1.81|
| Evaporator| 4.6420| 0.3698| 0.0025| 0      | 0      | 0     | 1.63| 1.66| 1.81|
| Turbine   | 2.2476| 1.4965| -0.1618| 0      | 0      | 0     | -  | -  | -  | 1.38|
| Condenser | 4.3247| -0.3030| 0.1634| 0      | 0      | 0     | 1.63| 1.66| 1.81|
| Pump      | 3.3892| 0.0536| 0.1538| 0      | 0      | 0     | 1.89| 1.35| 2.2724|

2.4.2 Genetic algorithm

The controlled elitist GA, which is a variant of NSGA-II [29], is employed to solve the multi-objective optimization model of the ORC system. An elitist GA always favors individuals with better fitness value. A controlled elitist GA also favors individuals that can help increase the diversity of the population even if they have a lower fitness value. It is important to maintain the diversity of population for convergence to an optimal Pareto front. Two features of the controlled elitist GA, fraction and distance, control the elitism. Fraction limits the number of individuals on the Pareto front (elite members). If the spread, a measure of the movement of the Pareto front, is small, the algorithm stops. The corresponding calculation procedures were developed using MATLAB.

Table 2. The main operating parameters of the ORC system.

| Parameter                        | value  | Parameter                        | value |
|----------------------------------|--------|----------------------------------|-------|
| Heat source temperature          | 395 K  | Pump isentropic efficiency       | 75%   |
| Heat source mass flow rate       | 27.7 kg/s | Cooling source temperature     | 300 K |
| Pinch point temperature difference in evaporator | 10 K   | Turbine isentropic efficiency    | 87%   |
| Pinch point temperature difference in condenser | 5 K    |                                 |       |

3 Results and discussion

Properties of the organic working fluids were calculated by REFPROP v9.0 software from the National Institute of Standards and Technology (NIST). Firstly, effects of evaporation temperature and condensation temperature on exergy efficiency and EPC of the ORC system were studied. The main operating parameters of the ORC system are listed in Table 2. Then, the multi-objective optimization was conducted and the Pareto frontiers with different working fluids were derived and compared.

3.1 Effects of evaporation temperature on exergy efficiency and EPC of the ORC system

The evaporation temperature increases from 333 to 373 K when the condensation temperature is at 313 K. Effects of exergy efficiency and EPC with evaporation temperature are depicted in Fig. 5 and Fig.6, respectively.

From Fig. 5, it is indicated that the exergy efficiency of the ORC system presents a significant increasing trend first and then increases gently with evaporation temperature, which coincides with the conclusion in [20]. Reasons are as follows. Higher evaporation temperature yields larger specific enthalpy drop in turbine. However, the heat source temperature at the evaporator outlet increases, which would result in a lower heat transfer rate in evaporator. Therefore, the working fluid mass flow rate would decrease. Under the comprehensive effects of the increasing specific enthalpy drop and the continuously decreasing mass flow rate, the net power output of the ORC system firstly increases, reaches a peak, and then decreases with increasing evaporation temperature. Although the total exergy destruction rates diminish monotonously, the exergy efficiencies with various working fluids keep rising.
consistent with the result in [18]. The changing trend is shown in Fig. 6.

3.2 Effects of condensation temperature on exergy efficiency and EPC of the ORC system

With constant evaporation temperature, variations of exergy efficiency and EPC of the ORC system with condensation temperature are exploited.

As demonstrated in Fig. 7, the exergy efficiency sharply declines with the increasing condensation temperature, which is in agreement with the result in [20]. Due to the unchanged evaporation temperature, the heat transfer rate in evaporator is maintained at a constant value. Consequently, the working fluid mass flow rate remains constant. However, the specific enthalpy drop in turbine would shrink owing to the increasing condensation temperature. Accordingly, the net power output of the ORC system remarkably decreases. Additionally, the increasing condensation temperature enables a decline in the total exergy destruction rate. Although both of the net power output and the exergy destruction rate decrease, the decline rate of the former is higher than that of the latter. Thus, the exergy efficiencies present a decreasing trend for all working fluids.

The bare module costs of turbine and pump decrease as condensation temperature increases, which results in a reduction of the total cost $C_{\text{total,2018}}$. In addition, the decreasing rate of $C_{\text{total,2018}}$ is lower than that of the net power output of the ORC system. As a result, the EPC exhibits an increasing trend with the increase in condensation temperature, depicted in Fig. 8.

3.3 Multi-objective optimization results

The multi-objective optimization results of the ORC system by using GA are discussed in this part. The Pareto frontiers for various working fluids are described in Fig. 9. As shown in the figure, variation of EPC and exergy efficiency of the ORC system with various working fluids present a similar tendency. Moreover, the exergy efficiencies present a slighter increasing trend than EPC. Taking R245fa for example, when the exergy efficiency varies from 46.9% to 54.3%, the EPC increases from 0.0477 to 0.0623 dollars per kW⋅h. The relative growth rates are 15.9% and 30.6%, respectively.

Furthermore, it can be easily found that improvement of exergy efficiency would enlarge the EPC of the ORC system. Conversely, reduction of EPC would lead to lower exergy efficiency. Evidently, it is impossible to simultaneously obtain the minimum EPC and the maximum exergy efficiency. That is to say, a compromise solution is needed to balance the economic cost and thermodynamic performance.
regarded as the Pareto frontiers. The optimal ranges of the exergy efficiency and EPC fluctuate from 51.3% to 52.2% and from 0.0505 to 0.0524 dollars.

![Fig. 10. Pareto frontier of EPC versus exergy efficiency with R245fa.](image)

4 Conclusion

In the present work, a multi-objective optimization has been implemented taking economic investment and thermodynamic performance of a specific ORC system for low-grade waste heat recovery into consideration. With R245fa, R245ca, R600, R600a, R601 and R601a as working fluids, thermodynamic and economic models were established. At first, effects of evaporation temperature and condensation temperature on exergy efficiency and EPC are investigated, respectively. Then the Pareto frontiers for various working fluids, thermodynamic and economic models were established. At first, effects of evaporation temperature and condensation temperature on exergy efficiency and EPC are investigated, respectively. Then the Pareto frontiers for various working fluids are obtained by solving the multi-objective optimization model using GA. The main conclusions are:

1. The exergy efficiency of the ORC system presents a pronounced increasing trend first and then increases gently with evaporation temperature. However, the EPC increases after an initial decrease as the evaporation temperature increases.
2. The exergy efficiency sharply decreases with the increasing condensation temperature, whereas the EPC exhibits an increasing trend with the increasing condensation temperature.
3. Compared the Pareto frontiers for various working fluids, the comprehensive performance of the ORC system with R245fa as working fluid is better than those with other fluids mentioned in the present work. Furthermore, the Pareto frontier solutions are derived.

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