Energy, Conventional and Advanced Exergy Analysis of Low Temperature Geothermal Binary-flashing Cycle Using Zeotropic Mixtures

Yuan Zhao

Powerchina HuaDong Engineering Corporation Limited, Hangzhou, 311122, China

Bowen Du

Powerchina HuaDong Engineering Corporation Limited, Hangzhou, 311122, China

Shunyi Chen

Powerchina HuaDong Engineering Corporation Limited, Hangzhou, 311122, China

Jun Zhao

Key Laboratory of Efficient Utilization of Low and Medium Grade Energy (Tianjin University), MOE, Tianjin, 300350, China

Lingbao Wang (✉ wanglb@ms.giec.ac.cn)

Guangzhou Institute of Energy Conversion <https://orcid.org/0000-0001-7395-8103>

Research

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Energy, conventional and advanced exergy analysis of low temperature geothermal binary-
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Yuanzhao^{a,b}, Bowen Du^a, Shunyi Chen^a, Jun Zhao^b, Lingbao Wang^{c,*}

^a Powerchina HuaDong Engineering Corporation Limited, Hangzhou, 311122, China

^b Key Laboratory of Efficient Utilization of Low and Medium Grade Energy (Tianjin University),
MOE, Tianjin, 300350, China

^c Guangzhou Institute of Energy Conversion, Chinese Academy of Sciences, Guangzhou 510640,
China

*Corresponding author. Tel.: +86 20 87057792; fax: +86 20 87057791.

E-mail address: wanglb@ms.giec.ac.cn (LB Wang)

Abstract:

Due to deep utilization of geobrine and high net power output, binary flashing cycle (BFC) is
deemed to be the future geothermal energy power generation technology. The BFC using
R245/R600a zeotropic mixtures is presented in this paper. The thermodynamic model of the system
is built, and energy, conventional and advanced exergy analysis are carried out, to reveal the real
optimization potential. It is demonstrated that the optimal composition mass fraction of R245fa and
dryness of working fluid at the evaporator outlet ranges are 0.30~0.50 and 0.40~0.60, considering
the thermodynamic performance and the flammability of the mixtures, simultaneously.
Conventional exergy analysis indicates that the maximum exergy destruction occurs in condenser,
followed by expander, evaporator, flashing tank, preheater, high-pressure pump and low-pressure

1 pump. While the advanced exergy analysis reveals that the expander should be given the first
2 priority for optimization, followed by condenser and evaporator. The BFC has a large potential for
3 improvement due to higher avoidable exergy destruction, about 48.6% of the total system exergy
4 destruction can be reduced. And the interconnections among system components are not very strong,
5 owing to small exogenous exergy destructions. It also demonstrates the effectiveness of advanced
6 exergy analysis, and the approach can be extended to other energy conversion systems to maximize
7 the energy and exergy savings for sustainable development.

8 **Keywords:** Advanced exergy analysis; Geothermal energy; Binary flashing cycle; Zeotropic
9 mixtures; Inerting mass concentration.

10 11 12 **1. Introduction**

13 Energy is the material basis of human activities and social development. With the rapid growth
14 of worldwide population and advance of science and technology, the ever increasing demand for
15 energy has brought serious global problems, such as energy shortage and environmental pollution.
16 Renewable clean energy utilization has become an extraordinary development trend to settle such
17 issues [1]. China has announced her pledge to reach peak emission by 2030 and carbon neutrality
18 by 2060 [2]. Renewable energies will play more and more important role, especially for solar and
19 wind energy [3-4]. While the high intermittency and variability, and low predictability and
20 controllability lead to huge challenges for the security and management of power grid, limiting the
21 large-scale application. Geothermal energy is characterized by wide distribution, abundant reserves
22 and high stability [5]. Therefore, the geothermal energy can be used as the base load to guarantee

1 the safe and stable operation of the power grid [6].

2 Organic rankine cycle (ORC) is regarded as a promising technology to exploit low-medium
3 geothermal energy [7]. Due to the relatively low efficiency of the basic ORC system, many efforts
4 has been made to boost the thermodynamic performance to strengthen popularization and
5 application. As a deeply improved ORC system, binary flashing cycle (BFC) is regarded as the
6 future geothermal power generation system. The investigations on the BFC mainly include system
7 performance comparison with ORC [8-11], double-evaporator ORC [12], and organic flash cycle
8 [13], working fluid selection [14-15], and operating parameters optimization [16-18]. Conclusions
9 can be drawn from the literatures that compared with the ORC, more net power output can be
10 achieved by the BFC; hydrocarbon working fluids appear to have excellent thermodynamic
11 performance; due to the reduction in the entropy generation resulted by the temperature glide effect,
12 zeotropic mixtures may be good alternatives; working fluid dryness at the evaporator outlet and
13 flashing temperature are the main parameters to be optimized owing to the inherent characteristic.

14 As can be seen, the studies on the BFC are very limited. The environmental friendly nature of the
15 working fluid has not been considered sufficiently. And most of the studies focus on the energy
16 analysis, only taking consideration of energy quantity. As an important supplement to energy
17 analysis, exergy analysis has already been proven to be a powerful tool to identify the location,
18 magnitude and sources of thermodynamic inefficiencies. The exergy destruction of every
19 component can be obtained through conventional exergy analysis. While this approach does not
20 consider interdependencies among system components and the real optimization potential of all
21 components. As a great progress, Tsatsaronis et al. [19] proposed an advanced exergy analysis
22 method by splitting the exergy destruction into endogenous/exogenous categories to further assess

1 the interdependencies among system components. Shortly afterwards Tsatsaronis and Moungh-Ho
2 [20] presented the other advanced exergy analysis method by dividing exergy destruction into
3 unavoidable/avoidable parts from the viewpoint of system component real optimization potential.
4 The combination of such two classification methods was referred to current advanced exergy
5 analysis. The advanced exergy analysis has been applied to various simple and complex
6 thermodynamic systems, such as ORC, ORC-modified system, ORC-based multi-generation system.
7 Dai et al. [21] conducted evaluated ORC system performance study using 12 Hydrocarbons working
8 fluids from the viewpoint of energetic and advanced exergetic analysis. Nami et al. [22] carried out
9 the advanced exergy analysis of geothermal driven ORC system, and indicated that the internal
10 exergy destruction accounts for 72% of the total exergy destruction. Gökgedik et al. [23] applied
11 advanced exergy analysis method to a real geothermal power plant to quantitatively evaluate the
12 thermodynamic performance improvement potential. The exergy efficiency can be improved from
13 9.60% to 15.40% by system components optimization. Wang et al. [24] presented advanced exergy
14 analysis of dual-loop ORC using zeotropic mixture, and revealed the improvement potential of each
15 component. Liao et al. [25] revealed that the technical modification of expander and condenser in
16 top ORC is beneficial to the ORC-ORC system efficiency improvement due to higher endogenous
17 avoidable exergy destruction. Chen et al. [26] constructed a novel and flexible ORC experimental
18 facility, and analyzed the experimental data to quantify the improvement potential employing the
19 advanced exergy analysis method. Montazerinejad et al. [27] applied both conventional and
20 advanced exergy analysis to combined cooling, heating and power (CCHP) system, and found that
21 expander yields the largest endogenous investment cost rate. Khosravi et al. [28] examined the
22 individual components of the optimal cycle from advanced exergy perspectives, and indicated that

1 the steam generator and expander have the first and second priorities for optimization. Anvari et al.
2 [29] applied advanced exergy analysis to identify components with high improvement potentials in
3 a tri-generation system producing heat, cold and power. It is indicated that over 32% of the total
4 exergy destruction are avoidable. Ambriz-Díaz et al. [30] performed an advanced exergy assessment
5 of polygeneration plant achieving cascade use of geothermal energy. It is revealed that 10.61 kW
6 and 2.28 kW of exergy destruction in the main heat exchanger and ORC can be avoided by design
7 variables optimization. Zhang et al. [31] presented thermodynamic improvement of integrated
8 transcritical CO₂ energy storage and ORC system. And it is indicated that thermal oil heat exchanger
9 should be firstly optimized.

10 To the best of the author's knowledge, there is almost no investigation of BFC using zeotropic
11 mixtures, with full consideration of the flammability of the mixtures. And the real optimization
12 potential of BFC has not been presented. To fill up the research gap, the present work focuses on
13 the advanced exergy analysis of the BFC using zeotropic mixtures to reveal the real optimization
14 potential of system as well as interactions among system components. As a modified ORC system,
15 a critical parameter, i.e. working fluid dryness at the evaporator outlet has been added in the BFC
16 compared with the ORC. The synergic optimization of the mixtures composition mass fraction and
17 dryness at the evaporator outlet is significant. The results are expected to provide meaningful
18 information for system optimization and practical guidance. In present study, the zeotropic mixtures
19 are chosen based on ideal thermodynamic and physical properties, environmental-friendly
20 characteristics with less Global Warming Potential (GWP) and Ozone-Depletion Potential (ODP),
21 and flammability. The R245fa and R600a are selected as the zeotropic mixtures [32-33], with the
22 thermophysical properties listed in Table 1.

Table 1 Thermophysical properties of R245fa and R600a

	Molecular	Normal	Critical	Critical	ODP	GWP	Flammability
Fluid	mass	boiling	temperature	pressure			
	(g/mol)	point (°C)	(°C)	(Mpa)			
R245fa	134.05	15.1	154.0	3.65	0	1030	Non-fammable
R600a	58.12	-11.7	134.7	3.63	0	3	Fammable

2 System Description

The schematic layout of BFC is depicted in Fig. 1. It mainly consists of a preheater, an evaporator, a separator, an expander, a generator, a flashing tank, a condenser, a high-pressure pump and a low-pressure tank. The high temperature geofluid passes through the evaporator and heats the working fluid into gas-liquid state. The gas-liquid two-phase current enters the separator, at which the gas-phase working fluid is separated and then is fed into the high-pressure stage of the expander to produce work. The liquid-phase working fluid that remains in the separator flows into the flashing tank. The gas-phase working fluid from the flashing tank is then passed into the low-pressure state of the expander to produce work. The liquid-phase working from the flashing tank is pumped into the evaporator. The working fluid exhaust from the expander passes through the condenser where it is condensed to liquid. The working fluid is then pumped into the preheater to absorb the heat of the geofluid from the evaporator to continue cycle operation.

Fig. 1 Schematic layout of BFC

3. System modeling

To facilitate the theoretical analysis, the mathematical model is built with the following assumptions:

- (1) The system operates under steady state condition;
- (2) The system ignores heat loss of heat exchangers;
- (3) The system neglects pressure losses in heat exchangers and pipelines;
- (4) The exergy destruction in the separator is ignored;
- (5) The geobrine is assumed to be pure water.

3.1 Energy analysis

The general mass and energy balance equations can be defined as:

$$\sum m_{in} - \sum m_{out} = 0 \quad (1)$$

$$\sum m_{in} X_{in} - \sum m_{out} X_{out} = 0 \quad (2)$$

$$(\sum m_{in} h_{in} + \sum Q_{in}) - (\sum m_{out} h_{out} + \sum Q_{out}) + W = 0 \quad (3)$$

where m is mass flow rate, kg/s; h is the specific enthalpy, kJ/kg; X is mass fraction of binary mixture; Q is heat transfer rate, kW; W is power, kW; the subscripts “in” and “out” denote inlet and outlet, respectively. The Eqs. 1-3 are the basis for the BFC detailed energy analysis.

The mass balance for the separator is given by

$$m_2 = m_{wf} x_{eva} \quad (4)$$

$$m_{10} = m_{wf} (1 - x_{eva}) \quad (5)$$

$$m_1 X_1 = m_2 X_2 + m_{10} X_{10} \quad (6)$$

where m_{wf} is the total working fluids mass flow rate, kg/s; x is the working fluid dryness degree at the evaporator outlet; the subscript “eva” denotes the evaporator; the subscript numbers represent the BFC working state points shown in Fig. 1.

The mass balance of the flashing tank is given by

$$m_{11} = m_{10} x_{FT} = m_{wf} (1 - x_{eva}) x_{FT} \quad (7)$$

$$m_{12} = m_{10} (1 - x_{FT}) = m_{wf} (1 - x_{eva}) (1 - x_{FT}) \quad (8)$$

$$m_{10} X_{10} = m_{11} X_{11} + m_{12} X_{12} \quad (9)$$

where the subscript “FT” denotes the flashing tank. Note that the flashing temperature is assumed to be the average of evaporation temperature and condensation temperature [16].

The power output of the expander is given by

$$W_{exp} = m_2 (h_2 - h_{3s}) \eta_{exp} + m_{11} (h_{11} - h_{3s}) \eta_{exp} \quad (10)$$

where η_{exp} represents the expander isentropic efficiency; the subscripts “exp” and “s” denote expander and isentropic, respectively.

The mass balance of the expander is given by

$$m_3 = m_2 + m_{11} \quad (12)$$

$$m_3 X_3 = m_2 X_2 + m_{11} X_{11} \quad (13)$$

The energy balance of the condenser is given by

$$Q_{con} = m_{cf} c_{p,cf} (t_{cf,out} - t_{cf,in}) = m_4 (h_3 - h_4) \quad (14)$$

where t is the temperature, °C; c_p denotes specific heat capacity, kJ/(kg•K); the subscript “cf” represents cooling water.

The mass balance of the condenser is given by

$$m_4 = m_3 \quad (15)$$

$$X_4 = X_3 \quad (16)$$

The power consumption of the low-pressure pump is given by

$$W_{LPP} = m_5 (h_5 - h_4) = m_5 (h_{5s} - h_4) / \eta_{LPP} \quad (17)$$

where η_{LPP} represents the low-pressure pump isentropic efficiency; the subscript “LPP” represents

1 low-pressure pump.

2 The mass balance of the low-pressure pump is given by

3
$$m_3 = m_4 \quad (18)$$

4
$$X_3 = X_4 \quad (19)$$

5 The energy balance of the preheater is given by

6
$$Q_{\text{pre}} = m_{\text{hf}} c_{\text{p,hf}} (t_{\text{hf,mid}} - t_{\text{hf,out}}) = m_6 (h_6 - h_5) \quad (20)$$

7 where the subscripts “hf” and “pre” represent the geofluid and preheater, respectively.

8 The mass balance of the preheater is given by

9
$$m_6 = m_5 \quad (21)$$

10
$$X_6 = X_5 \quad (22)$$

11 The power consumption of the high-pressure pump is given by

12
$$W_{\text{HPP}} = m_7 (h_7 - h_{12}) = m_7 (h_{7s} - h_{12}) / \eta_{\text{HPP}} \quad (23)$$

13 where η_{HPP} represents the high-pressure pump isentropic efficiency; the subscript “HPP” represents
14 high-pressure pump.

15 The mass balance of the high-pressure pump is given by

16
$$m_7 = m_{12} \quad (24)$$

17
$$X_7 = X_{12} \quad (25)$$

18 The energy balance of the evaporator is given by

19
$$Q_{\text{eva}} = m_{\text{hf}} c_{\text{p,hf}} (t_{\text{hf,in}} - t_{\text{hf,mid}}) = m_1 (h_1 - h_8) \quad (26)$$

20 The mass balance of the evaporator is given by

21
$$m_1 = m_8 = m_6 + m_7 \quad (27)$$

22
$$m_1 X_1 = m_8 X_8 = m_6 X_6 + m_7 X_7 \quad (28)$$

1 The net power output is given by

2
$$W_{\text{net}} = W_{\text{exp}} - W_{\text{LPP}} - W_{\text{HPP}} \quad (29)$$

3 where the subscript “net” represents net.

4 The thermal efficiency is given by

5
$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{pre}} + Q_{\text{eva}}} = \frac{W_{\text{net}}}{m_{\text{hf}} c_{\text{p,hf}} (T_{\text{hf,in}} - T_{\text{hf,out}})} \quad (30)$$

6 where η_{th} represents the thermal efficiency.

7 **3.2 Conventional exergy analysis**

8 The exergy balance of the BFC is given by

9
$$\sum_j \left(1 - \frac{T_0}{T_j} \right) Q_j - W + \sum m_{\text{in}} e_{\text{in}} - \sum m_{\text{out}} e_{\text{out}} - E_{\text{D}} = 0 \quad (31)$$

10 where T_0 is ambient temperature, °C; e is specific exergy, J/kg; E_{D} is exergy destruction, W.

11 The specific entropy is given by

12
$$e_j = (h_j - h_0) - T_0 (s_j - s_0) \quad (32)$$

13 where h_0 is specific enthalpy under ambient state, J/kg; s_0 is specific entropy under ambient state,
14 J/(kg·K).

15 The exergy destruction for k-th component is given by

16
$$E_{\text{D,k}} = E_{\text{F,k}} - E_{\text{P,k}} \quad (33)$$

17 where E_{F} and E_{P} are exergy input and exergy output, respectively, W.

18 The exergy balance of the BFC is given by

19
$$E_{\text{F,tot}} = E_{\text{P,tot}} - E_{\text{D,tot}} - E_{\text{L}} \quad (34)$$

20 where E_{L} is the exergy consumption, W.

21 The exergy destructions of flashing tank, expander, condenser, low-pressure pump, preheater, high-

pressure and evaporator are given by

$$E_{D,FT} = E_{F,FT} - E_{P,FT} = E_{10} - (E_{11} + E_{12}) \quad (35)$$

$$E_{D,exp} = E_{F,exp} - E_{P,exp} = (E_2 + E_{11} - E_3) - W_{exp} \quad (36)$$

$$E_{D,con} = E_{F,con} - E_{P,con} = (E_3 - E_4) - (E_{cf,out} - E_{cf,in}) \quad (37)$$

$$E_{D,LPP} = E_{F,LPP} - E_{P,LPP} = W_{LPP} - (E_5 - E_4) \quad (38)$$

$$E_{D,pre} = E_{F,pre} - E_{P,pre} = (E_{hf,mid} - E_{hf,out}) - (E_6 - E_5) \quad (39)$$

$$E_{D,HPP} = E_{F,HPP} - E_{P,HPP} = W_{HPP} - (E_7 - E_{12}) \quad (40)$$

$$E_{D,eva} = E_{F,eva} - E_{P,eva} = (E_{hf,in} - E_{hf,out}) - (E_1 - E_8) \quad (41)$$

The total exergy destruction of the BFC is given by

$$E_D = E_{D,FD} + E_{D,exp} + E_{D,con} + E_{D,LPP} + E_{D,pre} + E_{D,HPP} + E_{D,eva} \quad (42)$$

The exergetic efficiency of the BFC is given by

$$\eta_{ex} = \frac{W_{net}}{E_{in}} = \frac{W_{net}}{Q_{pre} \left(1 - \frac{T_0}{T_{m,pre}} \right) + Q_{gen} \left(1 - \frac{T_0}{T_{m,eva}} \right)} \quad (43)$$

3.3 Advanced exergy analysis

The exergy can identify the system inefficient components, nevertheless the interactions among components and the actual energy saving potential are not taken into account. The advanced exergy analysis considers the detailed interaction among components and intends to improve the quality of the results achieved from conventional exergy analysis.

The exergy destruction rate of the k-th component is split into two parts, named unavoidable and avoidable exergy destructions, as follows

$$E_{D,k} = E_{D,k}^{UN} + E_{D,k}^{AV} \quad (44)$$

where $E_{D,k}^{UN}$ is unavoidable exergy destruction caused by technical and economic limitations, W ;

1 $E_{D,K}^{AV}$ is avoidable exergy destruction, which can be reduced by technical progress, W.

2 The unavoidable exergy destruction is given by

3
$$E_{D,k}^{UN} = E_{P,k} \times \left(\frac{E_{D,k}}{E_{P,k}} \right)^{UN} \quad (45)$$

4 The exergy destruction of the k-th component can also be divided into endogenous and exogenous
5 parts. The endogenous exergy destruction is associated with the irreversibility of the k-th component
6 itself, while the exogenous exergy destruction is related to the irreversibility of other components.

7
$$E_{D,k} = E_{D,k}^{EN} + E_{D,k}^{EX} \quad (46)$$

8 where $E_{D,K}^{EN}$ is endogenous exergy destruction, W; $E_{D,K}^{EX}$ is exogenous exergy destruction, W.

9 In combination with the two splitting methods, the exergy destruction can be divided into four parts
10 as follows:

11
$$E_{D,k} = E_{D,k}^{AV,EN} + E_{D,k}^{AV,EX} + E_{D,k}^{UN,EN} + E_{D,k}^{UN,EX} \quad (47)$$

12 where $E_{D,K}^{AV,EN}$, $E_{D,K}^{AV,EX}$, $E_{D,K}^{UN,EN}$ and $E_{D,K}^{UN,EX}$ are avoidable-endogenous, avoidable-exogenous,
13 unavoidable-endogenous and the unavoidable-exogenous exergy destruction, W.

14 The detailed exergy destructions can be calculated as follows:

15
$$E_{D,k}^{UN,EN} = E_{P,k}^{EN} \times \left(\frac{E_{D,k}}{E_{P,k}} \right)^{UN} \quad (48)$$

16
$$E_{D,k}^{UN,EX} = E_{D,k}^{UN} - E_{D,k}^{UN,EN} \quad (49)$$

17
$$E_{D,k}^{AV,EN} = E_{D,k}^{EN} - E_{D,k}^{UN,EN} \quad (50)$$

18
$$E_{D,k}^{AV,EX} = E_{D,k}^{AV} - E_{D,k}^{AV,EN} \quad (51)$$

19

20 3.4 Flammability of the zeotropic mixtures

21 R600a is flammable. R245fa can be used as retardant. To ensure safety, the volume concentration

of the nonflammable working fluids in the zeotropic mixtures must be greater than minimum inerting concentration.

The minimum inerting volume concentration of the retardants in the zeotropic mixtures is given by

$$C_{\min} = \frac{0.21 \ln \left(\frac{0.05}{v_{\max}} \right) (C_{\text{st}} - 100)}{\Phi - 0.21 \ln \left(\frac{0.05}{v_{\max}} \right)} \quad (52)$$

where $C_{\min}$ is the minimum inerting volume concentration, %; $v_{\max}$ is the maximum flame propagation velocity of the flammable working fluid, m/s; C_{st} is stoichiometric concentration, %; Φ is the suppression coefficient of flammable retardant refrigerant.

The suppression coefficient of flammable retardant refrigerant can be obtained by group contribution method. The suppression coefficient of R245fa is 0.29.

The minimum inerting mass concentration of R245fa in R245fa/R600a mixtures can be calculated by

$$X_{\min} = \frac{C_{\min} \cdot M_{\text{R245fa}}}{C_{\min} \cdot M_{\text{R245fa}} + (1 - C_{\min}) \cdot M_{\text{R600a}}} \quad (53)$$

where, $X_{\min}$ is the minimum inerting mass concentration of R245fa, %; M_{R245fa} and M_{R600a} are the molar mass of R245fa and R600a, respectively, kg/kmol.

According to Eqs. 52-53, the minimum inerting mass concentration of R245fa in R245fa/R600a mixtures is 0.239.

Based on the above assumptions and equations, zeotropic mixtures composition optimization and advanced exergy analysis of the BFC are conducted. The fundamental database REFPROP 9.1 is used to describe the working fluids properties [34]. The operating conditions and parameters are listed in Table 2.

Table 2 Operating conditions and parameters

Parameters	Values
Inlet temperature of geobrine (°C)	90
Inlet temperature of cooling water (°C)	20
Degree of subcooling (°C)	3
Pinch point temperature of heat exchanger (°C)	5
Isentropic efficiency of expander	0.7
Isentropic efficiency of pump	0.6
Ambient temperature (°C)	20
Ambient pressure (kPa)	101.325

4. Model verification

The comparison between the results of the developed model and the ones in the literature is conducted to demonstrate its accuracy. The comparison under the identical operational conditions and working fluid is presented in Fig. 2, where it can be observed that the numerical prediction is close to the referenced data. As can be seen, only small differences exist between the results, indeed, the largest relative error of thermal efficiency is lower than 1.30%. It indicates a good agreement between the presented and referenced model.

Fig. 2 Comparison of simulation result between the present work and Ref. [12]

5. Results and discussions

5.1 Synergy optimization of mixtures composition mass fraction and dryness

The largest difference between ORC and BFC is the working fluid dryness at the evaporator outlet. One of the key issues of geothermal power generation using zeotropic mixtures is the design

1 of composition mass fraction. The input parameters for the synergy optimization of mixtures
 2 composition mass fraction and dryness at the evaporator outlet are listed in Table 1. The influences
 3 of composition mass fraction and dryness on the net power output, thermal efficiency, exergy
 4 efficiency and exergy destruction are illustrated in Figs. 3-6. As can be seen, the net power output,
 5 thermal efficiency and exergy efficiency present similar variation trend. There exist an approximate
 6 triangular region in the central zone of Figs. 3-5, at which the BFC yields the best thermodynamic
 7 performance. The optimal composition mass fraction of R245fa and dryness of working fluid at the
 8 evaporator outlet ranges are 0.30~0.70 and 0.30~0.80, at which the flammability of the mixtures is
 9 suppressed. From Fig. 6, the exergy destruction firstly increases and then decrease with the rising
 10 composition mass fraction of R245fa. While in the low composition mass fraction of R245fa ranges
 11 (0~0.50), the dryness at the evaporator outlet has weaker influence on the exergy destruction. To
 12 achieve the minimum exergy destruction, the optimal composition mass fraction of R245fa and
 13 dryness of working fluid at the evaporator outlet ranges are 0.25~0.50 and 0.40~0.60.
 14 Comprehensive consideration the thermodynamic performance, the optimal composition mass
 15 fraction of R245fa and dryness of working fluid at the evaporator outlet ranges are 0.30~0.50 and
 16 0.40~0.60.

17

18 **Fig. 3** Variations of W_{net} with x and X

19

20 **Fig. 4** Variations of η_{th} with x and X

21

22 **Fig. 5** Variations of η_{ex} with x and X

Fig. 6 Variations of E_d with x and X

5.2 Conventional exergy analysis

Based on analysis above, the composition mass fraction of R245fa and dryness of working fluid at the evaporator outlet are set to be 0.40 and 0.50, respectively, to facilitate the following analysis. The exergy destruction distribution of each component in the BFC is depicted in Fig.7. For writing convenience, evaporator, expander, flashing tank, condenser, low-pressure pump, high-pressure pump and preheater are abbreviated as “Eva”, “Exp”, “FT”, “Con”, “LPP”, “HPP” and “Pre”. As can be observed, the maximum exergy destruction is occurred in condenser (4.024 kW), accounting for 26.45% of the total exergy destruction. It is indicated that the condenser should be given the first priority for optimization. The exergy destructions of expander, evaporator and flashing tank are 3.866 kW, 2.939 kW and 2.699 kW, accounting for 25.40%, 19.32%, and 17.74%, respectively. The exergy destructions of preheater, high-pressure pump and low-pressure pump can be ignored. The exergy destruction in the heat exchangers, including preheater, evaporator and condenser accounting for 52.40% of the total exergy destruction. A great amount of energy quality is lost, or there is larger destruction rate in the heat exchanger, because huge energy with high temperature (high quality) is transferred to the working fluid/cooling water at low temperature (low quality) under irreversible process. As a consequence, in order to cut down the total exergy destruction for improving the exergy efficiency, more attention should be paid to heat exchanger optimization.

Fig. 7 Exergy destruction of each component in the BFC

5.3 Advanced exergy analysis

On the basis of the results derived from conventional exergy analysis, advanced exergy analysis is conducted to reveal the realistic improvement potential. For splitting exergy destruction into unavoidable and avoidable exergy destruction, the unavoidable operating condition of each unit need to be set. The main assumptions for real (actual operating conditions used for conventional analysis), unavoidable (with extremely high efficiency), and theoretical (theoretical maximum efficiency used to simulate the theoretical cycle) operating conditions of the components are listed in Table 3.

Table 3. Assumptions for the real, theoretical, and unavoidable operating conditions.

Component	Parameter	Real	unavoidable	theoretical
Heat exchanger	Pitch point temperature difference (°C)	5[35]	0.3[36]	0
Expander	Isentropic efficiency	0.7[37]	0.95[38]	1
Low-pressure pump	Isentropic efficiency	0.6[39]	0.95[38]	1
High-pressure pump	Isentropic efficiency	0.6[39]	0.95[38]	1
Flashing tank	Isentropic efficiency	Isenthalpic	0.95	1

Under the working conditions in Table 3, the state point parameters for real, unavoidable, and

1 theoretical operating conditions are listed in Tables 4-6. As can be seen, the net power output of
2 theoretical operating condition is 92.8% and 7.08% higher than those of real and unavoidable
3 operating conditions. The thermal efficiency of theoretical operating condition is 68.63% and 6.32%
4 higher than those of real and unavoidable operating conditions. The exergy efficiency of theoretical
5 operating condition is 78.16% and 6.64% higher than those of real and unavoidable operating
6 conditions.

7 Table 4 State point parameters for real operating conditions

State point	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg·K ⁻¹)	Mass flow rate (kg/s)	Dryness	Composition mass fraction
1	70.005	880.435	368.740	1.579	2.1632	0.141	0.600
2	70.005	880.435	546.160	2.097	0.3059	1	0.587
3	39.716	299.692	522.270	2.114	0.622	1.053	0.581
4	27.000	299.692	259.190	1.247	0.622	0	0.581
5	27.511	880.435	260.340	1.248	0.622	Subcooling	0.581
6	50.346	880.435	302.610	1.384	0.622	0	0.581
7	50.352	880.435	301.210	1.379	1.5413	Subcooling	0.608
8	50.346	880.435	301.610	1.380	2.1632	Subcooling	0.600
9	70.000	880.435	339.680	1.494	2.1632	0	0.600
10	70.005	880.435	339.520	1.494	1.8573	0	0.602
11	50.003	531.511	529.820	2.088	0.3161	1	0.576

12	50.003	531.511	300.490	1.378	1.5413	0	0.608
$W_{\text{net}} = 7.876 \text{ kW}; \eta_{\text{th}} = 4.590\%; \eta_{\text{ex}} = 32\%$							

1 Table 5 State point parameters for unavoidable operating conditions

State point	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg·K ⁻¹)	Mass flow rate (kg/s)	Dryness	Composition mass fraction
1	70.007	880.435	377.840	1.605	2.163	0.186	0.600
2	70.007	880.435	546.040	2.096	0.402	1	0.587
3	36.068	299.646	516.990	2.097	0.690	1.032	0.583
4	27.000	299.646	259.160	1.247	0.690	0	0.583
5	27.276	880.435	259.880	1.247	0.690	Subcooling	0.583
6	50.207	880.435	302.270	1.383	0.690	0	0.583
7	50.212	880.435	300.920	1.378	1.473	Subcooling	0.608
8	50.207	880.435	301.350	1.379	2.163	Subcooling	0.600
9	70.000	880.435	339.680	1.494	2.163	0	0.600
10	70.007	880.435	339.460	1.494	1.761	0	0.603
11	50.004	531.495	529.740	2.088	0.288	1	0.576
12	50.004	531.495	300.460	1.378	1.473	0	0.608
$W_{\text{net}} = 14.179 \text{ kW}; \eta_{\text{th}} = 7.280\%; \eta_{\text{ex}} = 53.460\%$							

2 Table 6 State point parameters for theoretical operating conditions

State	Temperature	Pressure	Enthalpy	Entropy	Mass	Dryness	Composition
-------	-------------	----------	----------	---------	------	---------	-------------

point	(°C)	(kPa)	(kJ/kg)	(kJ/kg·K ⁻¹)	flow		mass
				¹⁾	rate		fraction
					(kg/s)		
1	70.0072	880.4352	378.42	1.6071	2.1629	0.1886	0.6
2	70.0072	880.4352	546.03	2.0961	0.4079	1	0.587
3	35.2434	299.6433	515.85	2.0928	0.6944	1.0275	0.5825
4	27	299.6433	259.16	1.2465	0.6944	0	0.5825
5	27.2554	880.4352	259.85	1.2465	0.6944	Subcooling	0.5825
6	50.1945	880.4352	302.24	1.3825	0.6944	0	0.5825
7	50.1999	880.4352	300.89	1.3775	1.4685	Subcooling	0.6083
8	50.1945	880.4352	301.32	1.3791	2.1629	Subooling	0.6
9	70	880.4352	339.68	1.4942	2.1629	0	0.6
10	70.0072	880.4352	339.46	1.4935	1.755	0	0.603
11	50.0036	531.4933	529.73	2.0881	0.2865	1	0.5761
12	50.0036	531.4933	300.46	1.3775	1.4685	0	0.6083

$$W_{\text{net}} = 15.183 \text{ kW}; \eta_{\text{th}} = 7.740\%; \eta_{\text{ex}} = 57.010\%$$

1 In comparison with conventional exergy analysis, advanced exergy analysis further details
 2 the system exergy destruction and clarify the optimization direction. The
 3 avoidable/unavoidable exergy destruction of each component is illustrated in Fig. 8. It can be
 4 seen, the avoidable and unavoidable exergy destructions of the expander are 3.387 kW and
 5 0.4878 kW. The avoidable exergy destruction of the expander is the largest among all the
 6 components due to the occurrence of the expansion depressurization process, which is an

intensely irreversible process. And the avoidable exergy destruction of the expander accounts for 87.6% of the total exergy destruction of the expander. That is to say, most of the expander exergy destruction can be avoidable. The avoidable and unavoidable exergy destructions of the condenser are 2.565 kW and 1.459 kW. About 63.7% of the condenser exergy destruction can be avoidable. The avoidable and unavoidable exergy destructions of the evaporator are 1.148 kW and 1.790 kW. About 39.1% of the evaporator exergy destruction can be avoidable. The avoidable and unavoidable exergy destructions of the preheater are 0.363 kW and 0.646 kW. About 36.0% of the evaporator exergy destruction can be avoidable. Both the low-pressure pump and high-pressure pump exergy destructions are very little, and most of them are unavoidable. All the flashing tank exergy destruction is unavoidable. To sum up, the expander should draw the most attention, followed by condenser, evaporator, preheater, high-pressure pump and low-pressure pump. If there is only unavoidable exergy destruction in the BFC system, the total system exergy destruction can be reduced from 15.214 kW to 7.813 kW. While results from the conventional exergy analysis indicates that the condenser should be firstly optimized, followed by expander and evaporator, which is contradictory with that of the advanced exergy analysis. The comparison of component improvement priority between conventional and advanced exergy method presents a large difference due mainly to the different criteria. Furthermore, conclusions can be drawn that the engineers should focus on the avoidable exergy destruction rather than the total exergy destruction.

Fig. 8 Avoidable/unavoidable exergy destruction of each component

1 The exogenous/endogenous exergy destruction of each component is displayed in Fig. 7. As
2 can be seen, for any component of the BFC system, endogenous exergy destruction account for a
3 large proportion of the total exergy destruction. It indicates that the exergy destruction of the BFC
4 is mainly caused by the irreversibilities of the components themselves rather than their interactions.
5 Thus, it can be concluded that the interconnections between the system components are imcompact.
6 Therefore, the improvement of each component should be put in the first place when system
7 optimization is required. The interdependencies among system components can be positive or
8 negative, which could be caused by mass flow change or thermodynamic property variation of
9 working fluid through the specific component owing to the introduction of additional
10 irreversibilities. The exogenous exergy destructions of the low-pressure pump, expander and
11 condenser are all negative values, which is the result of differences in mass flow between
12 endogenous and real operating conditions. That is to say, the endogenous exergy destruction is
13 greater than the real exergy destruction. For the three components, performance improvement of
14 other components not only cannot reduce the exergy reduction, but also increase it. The increasing
15 in expander isentropic efficiency is the only useful measurement to reduce its exergy reduction.

16

17 **Fig. 9** Exogenous/endogenous exergy destruction of each component

18 **6. Conclusions**

19 In present study, the BFC using R245fa/ R600a zeotropic mixtures is proposed. Taking
20 consideration of the thermodynamic performance and working fluid flammability, the optimal
21 component mass fraction and dryness at the evaporator outlet introduced by the ORC system
22 modification, are given. The energy analysis, conventional exergy analysis and advanced exergy

analysis are conducted in sequence. The significant conclusions are summarized as follows:

(1) The recommended composition mass fraction of R245fa and dryness of working fluid at the evaporator outlet ranges are 0.30~0.50 and 0.40~0.60, at which the BFC achieves the optimal thermodynamic performance, and the flammability of the working fluid can be suppressed.

(2) It is indicated that the maximum exergy destruction is occurred in condenser, followed by expander, evaporator, flashing tank, preheater, high-pressure pump and low-pressure pump by conventional exergy analysis. The exergy destructions of preheater, high-pressure pump and low-pressure pump can be ignored. The condenser should be given the first priority. The exergy destruction in the heat exchangers accounts for 52.40% of the total exergy destruction

(3) The optimization sequence of BFC components deduced from the conventional and advanced methods is quite different. It is demonstrated that the priority should be given to the expander because of its largest avoidable exergy destruction, followed by condenser and evaporator. From the viewpoint of unavailability, about 48.6% of total system exergy destruction can be avoidable. And the interconnections among system components are not very strong, owing to small exogenous exergy destructions. Takes into account the interrelationships between components and the technical limitations of system components, the advanced exergy analysis could diagnose the detailed interactions among components of the BFC system and facilitate an exergoeconomic optimization.

Nomenclature

Symbols

c_p specific heat capacity ($\text{kJ/kg}\cdot\text{K}^{-1}$)

C_{st} stoichiometric concentration (%)

1	E_d	exergy destruction (W)
2	h	specific enthalpy (kJ/kg)
3	$\dot{m}$	mass flow rate (kg/s)
4	$\dot{Q}$	heat transfer rate (kW)
5	t	temperature (°C)
6	X	mass fraction of binary mixture
7	x	dryness
8	v	velocity, m/s;
9	W	work (kW)
10	Greek	
11	η	efficiency
12	Φ	suppression coefficient
13	Subscripts	
14	0	ambient condition
15	1, 2, ..., 12	state points
16	AV	avoidable
17	Con	condenser
18	EN	endogenous
19	EX	exogenous
20	Eva	evaporator
21	ex	exergy
22	Exp	expander

1 FT flash tank

2 HPP high-pressure pump

3 LPP low-pressure pump

4 Pre preheater

5 UN unavoidable

6 in inlet

7 out outlet

8 th thermal

9 wf working fluid

10 **Acronyms**

11 BFC binary flashing cycle

12 GWP global warming potential

13 LEC levelized energy cost

14 ODP ozone depletion potential

15 ORC Organic Rankine Cycle

16 **Declarations**

17 **Availability of data and materials**

18 The datasets used and/or analysed during the current study are available from the corresponding

19 author on reasonable request.

20 **Competing interests**

21 The authors declare that they have no competing interests.

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5 **Authors' contributions**

6 Yuan Zhao: Conceptualization, Methodology. Bowen Du: Software, Formal analysis. Shunyi
7 Chen: Validation, Supervision, Writing – review & editing, Visualization. Zhenzhu Yu: Validation,
8 Data curation. Lingbao Wang: Conceptualization, Methodology Writing – original draft

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11 **References**

- 12 [1] Nguyen, K. H., Kakinaka, M. (2019). Renewable energy consumption, carbon emissions, and
13 development stages: Some evidence from panel cointegration analysis. *Renewable Energy*, 132,
14 1049-1057.
- 15 [2] Davidson, M., Karplus, V. J., Zhang, D., & Zhang, X. (2021). Policies and Institutions to Support
16 Carbon Neutrality in China by 2060. *Economics of Energy & Environmental Policy*, 10(2), 7-25.
- 17 [3] Irfan, M., Elavarasan, R. M., Hao, Y., Feng, M., & Sailan, D. (2021). An assessment of
18 consumers' willingness to utilize solar energy in China: End-users' perspective. *Journal of Cleaner*
19 *Production*, 292, 126008.
- 20 [4] Magazzino, C., Mele, M., Schneider, N. (2021). A machine learning approach on the relationship
21 among solar and wind energy production, coal consumption, GDP, and CO2 emissions. *Renewable*
22 *Energy*, 167, 99-115.

- 1 [5] Kong, Y., Pang, Z., Shao, H., Hu, S. and Kolditz, O., (2014) Recent studies on hydrothermal
2 systems in China: a review. *Geothermal Energy*, 2014, 2:19.
- 3 [6] Zhang, L. X., Pang, M. Y., Han, J., Li, Y. Y., & Wang, C. B. (2019). Geothermal power in China:
4 Development and performance evaluation. *Renewable and Sustainable Energy Reviews*, 116,
5 109431.
- 6 [7] Özcan, Z., & Ekici, Ö. (2021). A novel working fluid selection and waste heat recovery by an
7 exergoeconomic approach for a geothermally sourced ORC system. *Geothermics*, 95, 102151.
- 8 [8] Michaelides, E.E., Scott, G.J., 1984. A binary-flashing geothermal power plant. *Energy* 9,
9 323–331.
- 10 [9] Shi, H., Michaelides, E.E., 1989. Binary dual-flashing geothermal power plants. *Int. J.*
11 *Energy Res.* 13, 127–135.
- 12 [10] Yuan, Z., Michaelides, E.E., 1993. Binary-flashing geothermal power plants. *J. Energy*
13 *Resour. Technol.* 115, 232–236
- 14 [11] Michaelides, E., 2016. Future directions and cycles for electricity production from
15 geothermal resources. *Energy Convers. Manage.* 107, 3–9
- 16 [12] Wang, Y.X., Wang, L.B., Li, H.S., et al., 2016. Thermodynamic calculation and
17 optimization of geothermal power generation in Ganzhi. *J. Harbin Eng. Univ.* 37,
18 873–877 (In Chinese).
- 19 [13] Mosaffa, A.H., Zareei, A., 2018. Proposal and thermoeconomic analysis of geothermal
20 flash binary power plants utilizing different types of organic flash cycle. *Geothermics*
21 72, 47–63.
- 22 [14] Edrisi, B.H., Michaelides, E.E., 2013. Effect of the working fluid on the optimum work of

1 binary-flashing geothermal power plants. *Energy* 50, 389–394

2 [15] Wang, L., Bu, X., Li, H., 2019. Investigation on geothermal binary-flashing cycle

3 employing zeotropic mixtures as working fluids. *Geothermal. Energy* 7 (1), 36.

4 [16] Liu, X., Li, H.S., Bu, X.B., et al., 2018a. Performance characteristics and working fluid

5 selection for low-temperature binary flashing cycle. *Appl. Therm. Eng.* 141, 51–60.

6 [17] Wang, L., Li, H., Bu, X., 2020. Multi-objective optimization of Binary Flashing Cycle

7 (BFC) driven by geothermal energy. *Appl. Therm. Eng.* 166, 114693

8 [18] Wang, L., Li, H., & Bu, X. (2021). Thermo-economic investigation of binary flashing cycle for

9 enhanced geothermal system. *Geothermics*, 89, 101951.

10 [19] G. Tsatsaronis. Strengths and limitations of exergy analysis

11 A. Bejan, E. Mamut (Eds.), *Thermodynamic optimization of complex energy systems*, Kluwer

12 Academic Publishers, Dordrecht (1999), pp. 93-100

13 [20] G. Tsatsaronis, P. Moungh-Ho.

14 On avoidable and unavoidable exergy destructions and investment costs in thermal systems

15 *Energy Convers Manag*, 43 (2002), pp. 1259-1270

16 [21] Dai, B., Zhu, K., Wang, Y., Sun, Z., & Liu, Z. (2019). Evaluation of organic Rankine cycle by

17 using hydrocarbons as working fluids: Advanced exergy and advanced exergoeconomic analyses.

18 *Energy Conversion and Management*, 197, 111876.

19 [22] Nami, H., Nemati, A., & Fard, F. J. (2017). Conventional and advanced exergy analyses of a

20 geothermal driven dual fluid organic Rankine cycle (ORC). *Applied Thermal Engineering*, 122, 59-

21 70.

22 [23] Gökgedik, H., Yürüsoy, M., & Keçebaş, A. (2016). Improvement potential of a real geothermal

1 power plant using advanced exergy analysis. *Energy*, 112, 254-263.

2 [24] Wang, Z., Xia, X., Pan, H., Zuo, Q., Zhou, N., & Xie, B. (2021). Fluid selection and advanced
3 exergy analysis of dual-loop ORC using zeotropic mixture. *Applied Thermal Engineering*, 185,
4 116423.

5 [25] Liao, G., Jiaqiang, E., Zhang, F., Chen, J., & Leng, E. (2020). Advanced exergy analysis for
6 Organic Rankine Cycle-based layout to recover waste heat of flue gas. *Applied Energy*, 266, 114891.

7 [26] Chen, J., Zheng, X., Guo, G., Luo, X., Chen, Y., & Yang, Z. (2019). A flexible and multi-
8 functional organic Rankine cycle system: Preliminary experimental study and advanced exergy
9 analysis. *Energy Conversion and Management*, 187, 339-355.

10 [27] Montazerinejad, H., Ahmadi, P., & Montazerinejad, Z. (2019). Advanced exergy, exergo-
11 economic and exergo-environmental analyses of a solar based trigeneration energy system. *Applied
12 Thermal Engineering*, 152, 666-685.

13 [28] Khosravi, H., Salehi, G. R., & Azad, M. T. (2019). Design of structure and optimization of
14 organic Rankine cycle for heat recovery from gas turbine: The use of 4E, advanced exergy and
15 advanced exergoeconomic analysis. *Applied Thermal Engineering*, 147, 272-290.

16 [29] Anvari, S., Saray, R. K., & Bahlouli, K. (2015). Conventional and advanced exergetic and
17 exergoeconomic analyses applied to a tri-generation cycle for heat, cold and power production.
18 *Energy*, 91, 925-939.

19 [30] Ambriz-Díaz, V. M., Rubio-Maya, C., Ruiz-Casanova, E., Martinez-Patino, J., & Pastor-
20 Martínez, E. (2020). Advanced exergy and exergoeconomic analysis for a polygeneration plant
21 operating in geothermal cascade. *Energy Conversion and Management*, 203, 112227.

22 [31] Zhang, Y., Liang, T., Yang, C., Zhang, X., & Yang, K. (2020). Advanced exergy analysis of an

1 integrated energy storage system based on transcritical CO₂ energy storage and Organic Rankine
2 Cycle. *Energy Conversion and Management*, 216, 112938.

3 [32] Yang, J., Gao, L., Ye, Z., Hwang, Y., & Chen, J. (2021). Binary-objective optimization of latest
4 low-GWP alternatives to R245fa for organic Rankine cycle application. *Energy*, 217, 119336.

5 [33] Feng, Y., Hung, T., Zhang, Y., Li, B., Yang, J., & Shi, Y. (2015). Performance comparison of
6 low-grade ORCs (organic Rankine cycles) using R245fa, pentane and their mixtures based on the
7 thermoeconomic multi-objective optimization and decision makings. *Energy*, 93, 2018-2029.

8 [34] E.W. Lemmon, M.L. Huber, M.O. McLinden. NIST Standard Reference Database 23:
9 Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, Standard
10 Reference Data Program. National Institute of Standards and Technology, Gaithersburg, MD (2013)

11 [35] Bai T, Yu J, Yan G. Advanced exergy analysis on a modified auto-cascade freezer cycle with
12 an ejector. *Energy*, 2016, 113: 385-398.

13 [36] Hu Y. Advanced exergy analysis for a solar double stage absorption chiller[D]. Pittsburgh, USA:
14 Carnegie Mellon University, 2012.

15 [37] Lemort V, Quoilin S, Cuevas C, et al. Testing and modeling a scroll expander integrated into
16 an organic Rankine cycle. *Applied Thermal Engineering*, 2009, 29(14-15): 3094-3102.

17 [38] Fallah M, Mahmoudi SMS, Yari M, et al. Advanced exergy analysis of the Kalina cycle applied
18 for low temperature enhanced geothermal system. *Energy Conversion and Management*, 2016, 108:
19 190-201.

20 [39] Imran M, Usman M, Park BS, et al. Multi-objective optimization of evaporator of organic
21 Rankine cycle (ORC) for low temperature geothermal heat source. *Applied Thermal Engineering*,
22 2015, 80: 1-9.

1

2

3

Figures

Figure 1

Schematic layout of BFC

Figure 2

Comparison of simulation result between the present work and Ref. [12]

Figure 3

Variations of W_{net} with x and X

Figure 4

Variations of η_{th} with x and X

Figure 5

Variations of η_{ex} with x and X

Figure 6

Variations of E_d with x and X

Figure 7

Exergy destruction of each component in the BFC

Figure 8

Avoidable/unavoidable exergy destruction of each component

Figure 9

Exogenous/endogenous exergy destruction of each component

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