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Research paper

Exergoeconomic performance optimization for a regenerative closed-cycle gas turbine combined heat and power plant

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ABSTRACT

A combined heat and power plant with regenerative closed-cycle gas turbine is considered. The heat transfer losses, irreversible adiabatic losses and irreversible pressure drop losses are considered in the plant. The expressions of dimensionless profit rate and exergy efficiency are gained. The optimal heat conductance distributions and compressor pressure ratio are derived based on maximum dimensionless profit rate and maximum exergy efficiency objectives, respectively. The results show that the heat conductance distributions and compressor pressure ratio have their optimal values which results in maximum dimensionless profit rate. The curve of maximum dimensionless profit rate and the corresponding exergy efficiency is loop-shaped one. With the increases in heat reservoir temperature ratio τ_1 , price ratios a and b as well as total heat exchanger inventory U_T , the maximum dimensionless profit rate increases. With the decrease in a and increases in τ_1 , b and U_T , the corresponding exergy efficiency increases. The heat reservoir temperature ratio τ_2 has its optimal value which leads to double maximum dimensionless profit rate. Moreover, the maximum exergy efficiency performance of the plant is also analyzed. Because the exergoeconomic performance of irreversible regenerative closed-cycle gas turbine cogeneration plant has not been investigated by the researchers before, the exergoeconomic performance optimization is conducted in this paper, and the results obtained can offer new design guidelines for this kind of cogeneration plants.

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1. Introduction

Combined heat and power plant (CHPP) has a higher energy utilization rate compared with the pure power plant (Wang et al., 2017; Han et al., 2017; Mehrpooya et al., 2019; Urbanucci et al., 2019). Many scholars have conducted parameters optimizations to improve CHPP performances (Caliano et al., 2017; Wang et al., 2018; Abbasi et al., 2018; Wang et al., 2019; Sheykhi et al., 2019; Wu et al., 2019) by using classical thermodynamics.

Finite-time thermodynamics (FTT) (Andresen et al., 1984; Bejan, 1996; Chen et al., 1999; Chen and Xia, 2018; Abam et al., 2018; Lucia and Grisolia, 2019; Chen and Xia, 2019; Chen and Li, 2020) has exhibited its advantages in performance optimizations of various thermodynamic processes and cycles. Among them, FTT optimizations for CHPPs are one of the hot issues. Bojic

(1997) considered the heat transfer losses (HTLs) of a CHPP with Carnot cycle, and optimized its thermoeconomic performance by changing the heat conduction distributions. Kaushik et al. (2005) and Atmaca et al. (2009) discussed the exergetic performances of a Carnot CHPP, and found that the irreversibility parameters had evident effects on its performances. Atmaca et al. (2016) further studied the CHPP applications for the housing complex, and obtained a short payback period of the CHPP. Besides Carnot CHPPs, Ust et al. (2007) discussed the exergy performance coefficient of a CHPP with Dual cycle, and pointed out that this coefficient showed a good compromise between exergy output rate (EOR) and entropy generation rate (EGR). Yilmaz et al. (2008) optimized the EOR of a CHPP with simple Brayton cycle, and pointed out that a lower temperature of the thermal consumer (TTC) led to a higher EOR. Hao and Zhang (2013) analyzed the useful-energy rate (UER) and efficiency of a CHPP with open Brayton cycle, and found that the optimal results derived by maximizations of UER and efficiency were different when the ratio of power to heat was larger. Dubey et al. (2016) calculated the UER of a CHPP with reheat Joule–Brayton cycle, and found that the performance of

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Nomenclature	
<i>a</i>	Price ratio
<i>b</i>	Price ratio
<i>C</i>	Thermal capacity rate, kW/K
<i>D</i>	Pressure recovery coefficient
<i>E_H</i>	Net exergy output rate (EOR), kW
<i>E_P</i>	EOR of power, kW
<i>E_K</i>	Thermal EOR, kW
<i>P</i>	Power output, kW
<i>p</i>	Pressure, Pa
<i>Q</i>	Heat transfer rate, kW
<i>T</i>	Temperature, K
<i>U</i>	Heat conductance, kW/K
<i>u</i>	Heat conductance distribution
Greek symbols	
γ	Effectiveness of the heat exchanger
η	Exergy efficiency
λ	Process efficiency
Π	Profit rate, \$
π	Pressure ratio
σ	Entropy generation rate, kW/K
φ	Price, \$/kW
τ	Temperature ratio for heat reservoir
ω	Temperature ratio for process 1–2 s
Subscripts	
$\bar{\Pi}$	Maximum dimensionless profit rate
<i>H</i>	Hot-side
<i>K</i>	Thermal consumer-side
<i>L</i>	Cold-side
max	Maximum
opt	Optimal
<i>R</i>	Regenerator
<i>T</i>	Total
η	Maximum exergy efficiency
0	Ambient
1, 2, 2s, 3, 4, 5, 5s, 6, 7	State points
Superscript	
—	Dimensionless

the CHPP could be evidently improved by adopting reheater. Yang et al. (2014a, 2016) conducted the exergy optimizations of the CHPPs with intercooled regenerated Brayton cycles, and obtained the optimal consumer-side temperatures leading to optimal exergy performances. Moreover, the trigeneration systems (Chen et al., 2013; Mohammadi et al., 2017; Mirzaee et al., 2019) were also investigated by some scholars based on FTI.

Besides the optimization objectives mentioned above, exergoeconomic (or thermoeconomic) optimizations (Tsatsaronts, 1993; El-Sayed, 2003) have also been attracted by some scholars. Salomon and Nitzan (1981) introduced the exergoeconomic concept (Tsatsaronts, 1993; El-Sayed, 2003) into the performance analyses of endoreversible models, formed a new analysis method – finite time exergoeconomic analysis (FTEA)

(Salamon and Nitzan, 1981; Chen et al., 1997, 2004; Ding et al., 2011; Kan et al., 2010; Feng et al., 2010; Ge et al., 2019; Wu et al., 2000; Chen et al., 1996; Tao et al., 2009; Yang et al., 2011; Chen et al., 2014; Yang et al., 2014b; Chen et al., 2011; Feng et al., 2011). The dimensionless profit rate (DPR) was maximized in the FTEA, and the corresponding exergy efficiency (EE) was named as finite time exergoeconomic performance bound (FTEPB). The FTEA is widely used in the cycle performance evaluations, such as Carnot heat engine cycles (Chen et al., 1997, 2004), universal two-heat-reservoir heat engine cycle (Ding et al., 2011), universal two-heat-reservoir refrigerator cycle (Kan et al., 2010), universal two-heat-reservoir heat pump cycle (Feng et al., 2010), complex Brayton cycle (Ge et al., 2019), quantum Stirling heat engine cycle (Wu et al., 2000), absorption cycles (Chen et al., 1996), CHPPs with simple and complex Brayton cycles (Tao et al., 2009; Yang et al., 2011; Chen et al., 2014; Yang et al., 2014b), and trigeneration plants with simple Brayton cycle (Chen et al., 2011; Feng et al., 2011), etc. FTEA was conducted in the researches mentioned above from the viewpoint of long-run economic optimization, and new performance bounds were provided different from those derived by traditional analyses.

In this paper, the FTEA will be introduced into the performance optimizations of a CHPP with regenerative closed-cycle gas turbine (RCCGT). Three kinds of losses will be considered in the CHPP. The expressions of DPR and EE of CHPP will be derived. The heat conductance distributions (HCDs) and compressor pressure ratio (CPR) will be optimized based on maximum DPR and maximum EE objectives, respectively. Characteristics of optimal results of CHPP versus heat reservoir temperature ratios, price ratios and total heat exchanger inventory (THEI) will be analyzed. The result differences between this paper and previous researches will be compared. Two innovations of this paper are introducing the FTI theory and FTEA method into the performance optimization of the irreversible RCCGT, and new guidelines for the CHPP designers will be offered.

2. Cycle model

Figs. 1 and 2 show the flow diagram as well as the relationship of temperature and specific entropy of a CHPP with RCCGT, respectively. The heat reservoir temperatures of the heat exchangers (HEs) at hot-, cold- and consumer-sides (HEHCCSs) are T_H , T_L and T_K , respectively, and the working fluid (WF) runs between the heat reservoirs. The compression and expansion processes 1–2 and 4–5 in the compressor and turbine are non-isentropic ones. The CPR of the compressor is $\pi = p_2/p_1$. The isobaric processes 3–4 and 7–1 are heat absorbing and releasing processes, and processes 2–3 and 5–6 are those in the regenerator. The processes 1–2–3–4–5–6–7–1 form the whole cycle of CHPP, and processes 1–2s–3–4–5s–6–7–1 are the corresponding endoreversible cycle. The differences between the endoreversible and irreversible RCCGTs are reflected in the two adiabatic processes. For the former cycle, processes 1–2 s and 4–5 s are isentropic processes, which are the deal processes. For the latter cycle, the irreversible factors in the compressor and turbine are considered, and the more practical non-isentropic processes 1–2 and 4–5 are adopted instead of processes 1–2 s and 4–5 s.

The two pressure recovery coefficients are given as:

$$D_1 = p_4/p_2, D_2 = p_1/p_5. \quad (1)$$

The heat transfer rate (HTR, Q_H) of hot-side HE is:

$$Q_H = C_{wf}(T_4 - T_3) = C_{wf}\gamma_H(T_H - T_3), \quad (2)$$

where C_{wf} is the rate of thermal capacity, and the effectiveness γ_H is calculated by:

$$\gamma_H = 1 - e^{-(U_H/C_{wf})}, \quad (3)$$

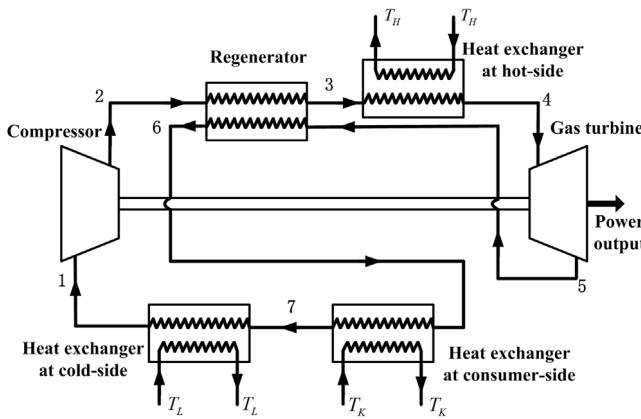


Fig. 1. Flow diagram of CHP plant composed of a regenerative closed-cycle gas turbine.

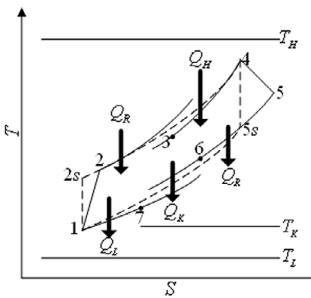


Fig. 2. T-s diagram of CHP plant composed of a regenerative closed-cycle gas turbine.

where U_H is the heat conductance. The HTRs (Q_L , Q_K and Q_R) of the cold- and consumer-side HEs and regenerator can be derived similar with Eqs. (2) and (3) (Tao et al., 2009), and the corresponding meanings of the symbols are similar with those of the hot-side HE.

3. Performance relations

The temperature ratio for process 1–2 s can be expressed as:

$$\omega = T_{2s}/T_1 = \pi^m, \quad (4)$$

where m is a constant, and π is the CPR. For endoreversible cycle in Fig. 2, one has:

$$T_1 T_4 = T_{2s} T_{5s}. \quad (5)$$

Combining Eqs. (1), (4) and (5) yields:

$$T_4/T_{5s} = (p_4/p_5)^m/(D_1 D_2)^m = \omega. \quad (6)$$

The power output (PO) of the CHPP, i.e. the EOR derived from PO, is:

$$E_P = P = Q_H - Q_L - Q_K. \quad (7)$$

According to the definition of exergy and exergy balance equation, the net thermal exergy rate (NTER), thermal EOR and EGR of CHPP can be expressed as:

$$E_H = c_H Q_H - c_L Q_L = E_P + E_K + T_0 \sigma, \quad (8)$$

$$E_K = Q_K [1 - \frac{T_0 \cdot \ln(T_6/T_7)}{T_6 - T_7}], \quad (9)$$

$$\sigma = \frac{Q_L}{T_L} - \frac{Q_H}{T_H} + \frac{Q_K \cdot \ln(T_6/T_7)}{T_6 - T_7}, \quad (10)$$

where T_0 is the ambient temperature, and $c_i = 1 - T_0/T_i$ ($i = H, L$). The profit rate of CHPP is:

$$\Pi = \varphi_P E_P + \varphi_K E_K - \varphi_H E_H, \quad (11)$$

where φ_H , φ_K and φ_P are the prices of the NTER, thermal EOR and PO, respectively.

Combining Eqs. (1)–(9), the DPR can be expressed as:

$$\begin{aligned} & a\{\lambda_1\{1 - p[\gamma_R + (1 - \gamma_R)(\gamma_K + \gamma_L - \gamma_L \gamma_K)]\} - q(1 - \gamma_L) \\ & \times (1 - \gamma_K)[\gamma_R + p(1 - 2\gamma_R)]\}\gamma_H \tau_1 + \{\lambda_1[1 - \gamma_R p(1 - \gamma_H)] \\ & - q[\gamma_H + \gamma_R - \gamma_H \gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)]\}(1 - \gamma_L) \\ & \times \gamma_K \tau_2 + \gamma_L \tau_3\} + b\gamma_K(1 - \tau_2^{-1})\{\lambda_1(1 - \gamma_R)p\gamma_H \tau_1 - \lambda_1 p[1 \\ & - \gamma_R(1 - \gamma_H)]\}\tau_2 + q[\gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)][(1 - \gamma_L)\tau_2 \\ & + \gamma_L \tau_3] - \gamma_H(1 - \tau_1^{-1})\{\lambda_1[1 - \gamma_R p] - q(1 - \gamma_L)\}\gamma_K \tau_2 + \gamma_L \tau_3\} \\ & + \gamma_L(1 - \tau_3^{-1})\{(1 - \gamma_K)(1 - \gamma_R)p\lambda_1 \gamma_H \tau_1 + \lambda_1[1 - \gamma_R p(1 - \gamma_H)] \\ & \times (\gamma_K \tau_2 - \tau_3) + (1 - \gamma_K)q[\gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)]\}\tau_3\} \\ \overline{\Pi} = & \frac{a\{\lambda_1\{1 - p[\gamma_R + (1 - \gamma_R)(\gamma_K + \gamma_L - \gamma_L \gamma_K)]\} - q(1 - \gamma_L)(1 - \gamma_K)[\gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)]\}}{\lambda_1[1 - \gamma_R p(1 - \gamma_H)] - q(1 - \gamma_L)(1 - \gamma_K)[\gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)]}, \end{aligned} \quad (12)$$

where $p = 1 - \lambda_2 + \lambda_2 \omega^{-1}(D_1 D_2)^{-m}$, $q = \omega - 1 + \lambda_1$, λ_1 and λ_2 are the efficiencies of processes 1–2 and 4–5, the price ratios are defined as: $a = \varphi_P/\varphi_H$ and $b = \varphi_K/\varphi_H$, and the heat reservoir temperature ratios are defined as: $\tau_1 = T_H/T_0$, $\tau_2 = T_K/T_0$ and $\tau_3 = T_L/T_0$, respectively.

Combining Eqs. (1)–(9), the EE of the CHPP can be expressed as:

$$\eta = \frac{E_P + E_K}{E_H} = \frac{\lambda_1\{1 - p[\gamma_R + (1 - \gamma_R)(\gamma_K + \gamma_L - \gamma_L \gamma_K)]\} - q(1 - \gamma_L)}{\gamma_H(1 - \tau_1^{-1})\{\lambda_1[1 - \gamma_R p] - (1 - \gamma_L)(1 - \gamma_K)q[\gamma_R \\ \times (1 - 2\gamma_R)]\}\tau_1 - (1 - \gamma_R)q[(1 - \gamma_L)\gamma_K \tau_2 + \gamma_L \tau_3]} \\ - q[\gamma_H + \gamma_R - \gamma_H \gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)][(1 - \gamma_L)\gamma_K \tau_2 \\ + \gamma_L \tau_3] + \gamma_K(1 - \tau_2^{-1})\{\lambda_1(1 - \gamma_R)p\gamma_H \tau_1 - \lambda_1 \tau_2[1 - \gamma_R(1 \\ - \gamma_H)]\} + q[\gamma_R + p(1 - \gamma_H)(1 - 2\gamma_R)][(1 - \gamma_L)\tau_2 + \gamma_L \tau_3\}] \\ \times \gamma_H(1 - \tau_1^{-1})\{\lambda_1[1 - \gamma_R p] - (1 - \gamma_L)(1 - \gamma_K)q[\gamma_R \\ \times (1 - 2\gamma_R)]\}\tau_1 - (1 - \gamma_R)q[(1 - \gamma_L)\gamma_K \tau_2 + \gamma_L \tau_3\}] \\ - \gamma_L(1 - \tau_3^{-1})(1 - \gamma_K)(1 - \gamma_R)p\lambda_1 \gamma_H \tau_1 + \lambda_1[1 - \gamma_R \\ \times p(1 - \gamma_H)][\gamma_K \tau_2 - \tau_3] + (1 - \gamma_K)q[\gamma_R + p(1 - \gamma_H) \\ \times (1 - 2\gamma_R)]\tau_3\}. \quad (13)$$

4. Discussions

When $\varphi_P \geq \varphi_H$ and $\varphi_K \geq \varphi_H$, the CHPP may be profitable. The DPR and EE in Eqs. (12) and (13) are generalized expressions, and many results obtained in previous work are special cases of these equations.

(1) When $\gamma_R = 0$, $\lambda_1 = \lambda_2 = 1$ and $D_1 = D_2 = 1$, Eqs. (12) and (13) become the DPR and EE of CHPP with endoreversible simple Brayton cycle (SBC). However, the EEs are also different when the calculation methods of thermal EORs are different in this paper and available reference.

(2) When $\gamma_R = 0$ and $D_1 = D_2 = 1$, Eqs. (12) and (13) become the DPR and EE of CHPP with irreversible SBC.

(3) When $\lambda_1 = \lambda_2 = 1$ and $D_1 = D_2 = 1$, Eqs. (12) and (13) become the DPR and EE of CHPP with endoreversible regenerative gas turbine (Tao et al., 2009).

(4) When $\varphi_P = \varphi_K = \varphi_H$, Eq. (11) becomes:

$$\Pi = \varphi_P(E_P + E_K - E_H) = -\varphi_P T_0 \sigma. \quad (14)$$

The maximum DPR objective is simplified into minimum EGR objective.

(5) When $\varphi_P/\varphi_H \rightarrow \infty$ and $\varphi_P = \varphi_K$, Eq. (11) is simplified into:

$$\Pi = \varphi_P(E_P + E_K). \quad (15)$$

The maximum DPR objective is simplified into maximum exergy production objective.

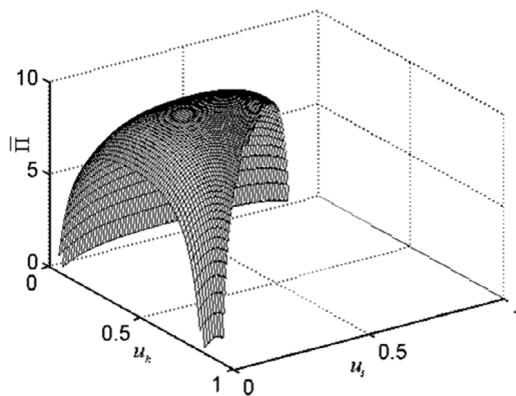


Fig. 3. Three-dimensional characteristic of $\bar{\Pi}$ versus u_h and u_l .

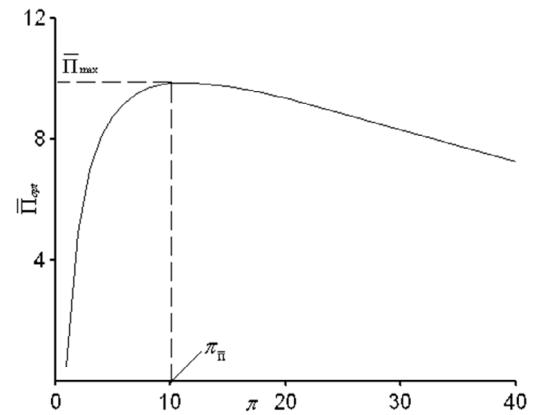


Fig. 4. Characteristic of $\bar{\Pi}_{opt}$ versus π .

5. Performance optimizations

According to Eq. (12), the DPR $\bar{\Pi}$ is a function of $a, b, D_1, D_2, \lambda_1, \lambda_2, \tau_1, \tau_2, \tau_3, \pi, U_H, U_L, U_K$ and U_R . Similar with the constraints for regenerative Brayton cycles in Refs. Tao et al. (2009), Yang et al. (2011), Chen et al. (2014) and Yang et al. (2014b), the constraint of THEI (U_T) is considered in the performance optimization of the CHPP with RCCGT.

When U_T is unchanged, the definitions of the four HCDs are expressed as:

$$(u_h, u_l, u_k, u_r) = (U_H, U_L, U_K, U_R)/U_T, \quad (16)$$

where the sum of the four HCDs is 1. Numerical calculation shows that the DPR reach its maximum when $u_r = 0$, which illustrates that the regeneration leads to a decrement of DPR.

Actually, the regeneration is useful for energy saving of CHPP. To avoid the contradiction in the optimization, the HCD of regeneration is fixed. For the fixed U_T, u_r, u_h, u_l, u_k and π will be varied to research for the maximum value of DPR. In the optimizations, $a = 10, b = 6$ (Tao et al., 2009; Yang et al., 2011; Chen et al., 2014; Yang et al., 2014b; Chen et al., 2011; Feng et al., 2011), $C_{wf} = 1.0 \text{ kW/K}$, $U_T = 10 \text{ kW/K}$, $k = 1.4$, $\lambda_1 = \lambda_2 = 0.85$, $D_1 = D_2 = 0.95$, $\tau_1 = 5$, $\tau_2 = 1.4$ and $\tau_3 = 1$ are set. The relationship of $\bar{\Pi}$, u_h and u_l is shown in Fig. 3. When the other parameters are fixed, the DPR can be maximized by varying u_h and u_l simultaneously. When the HCD u_h is further varied, the optimal DPR ($\bar{\Pi}_{opt}$) can be obtained. The characteristic of $\bar{\Pi}_{opt}$ versus the CPR π is shown in Fig. 4. π has its optimal value ($\pi_{\bar{\Pi}}$) which leads to maximum DPR ($\bar{\Pi}_{max}$) of CHPP, and the corresponding HCDs at $\bar{\Pi}_{max}$ condition are signed as $(u_h)_{\bar{\Pi}}$, $(u_l)_{\bar{\Pi}}$ and $(u_k)_{\bar{\Pi}}$, respectively.

5.1. Optimal heat conductance distributions and compressor pressure ratio

The characteristics of $(u_h)_{\bar{\Pi}}$, $(u_l)_{\bar{\Pi}}$, $(u_k)_{\bar{\Pi}}$ and $\pi_{\bar{\Pi}}$ versus τ_1 , a and U_T are shown in Figs. 5–7, respectively. Similar characteristics for τ_2 and b are also analyzed by numerical calculations. They illustrate that $(u_l)_{\bar{\Pi}}$ increases when τ_2 , a and U_T increase, and the tendency is reversed when τ_1 and b increase; $(u_k)_{\bar{\Pi}}$ increases when τ_1 and b increase, and the tendency is reversed when τ_2 and a increase; $\pi_{\bar{\Pi}}$ increases when τ_1 , a and U_T increase, and the tendency is reversed when b increases; $(u_h)_{\bar{\Pi}}$ does not evidently change when the discussed parameters vary. One can see that most of the design parameters have significant impacts on the optimal values of design variables, which illustrates the importance of parameter analyses in this section.

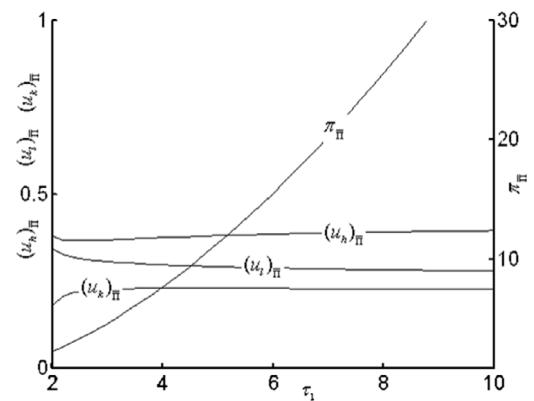


Fig. 5. Characteristics of $(u_h)_{\bar{\Pi}}$, $(u_l)_{\bar{\Pi}}$, $(u_k)_{\bar{\Pi}}$ and $\pi_{\bar{\Pi}}$ versus τ_1 .

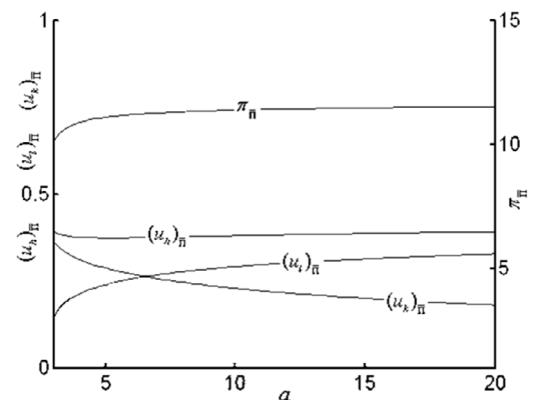


Fig. 6. Characteristics of $(u_h)_{\bar{\Pi}}$, $(u_l)_{\bar{\Pi}}$, $(u_k)_{\bar{\Pi}}$ and $\pi_{\bar{\Pi}}$ versus a .

5.2. Dimensionless profit rate versus exergy efficiency characteristic

The characteristic of optimal DPR ($\bar{\Pi}_{opt}$) versus the corresponding EE ($\eta_{\bar{\Pi}opt}$) with different u_r is shown in Fig. 8. It shows that $\bar{\Pi}_{opt} - \eta_{\bar{\Pi}opt}$ characteristic is leaf-shaped one. The maximum DPR ($\bar{\Pi}_{max}$) and the FTEPB ($\eta_{\bar{\Pi}}$) can be derived at the highest point of the leaf-shaped curve. Furthermore, the maximum EE (η_{max}) and the corresponding DPR ($\bar{\Pi}_\eta$) can be also obtained. When $u_r \approx 0.1$, the optimization results derived by DPR and EE maximizations are equal to each other.

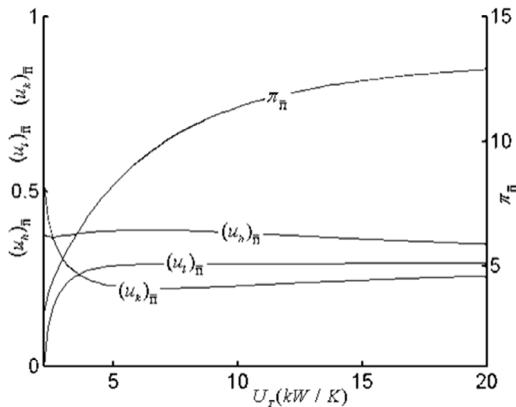


Fig. 7. Characteristics of $(u_h)_\Pi$, $(u_l)_\Pi$, $(u_k)_\Pi$ and π_Π versus U_T .

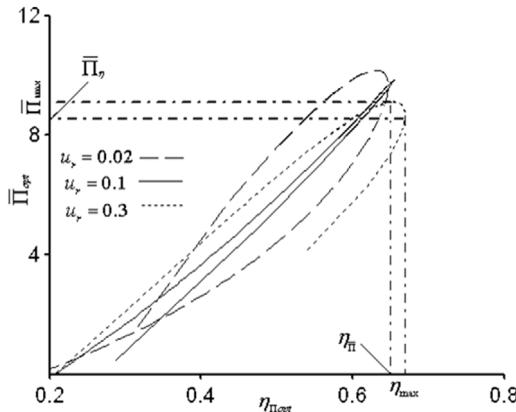


Fig. 8. Influence of u_r on characteristic of $\bar{\Pi}_{opt}$ versus $\eta_{\Pi_{opt}}$.

5.3. Maximum dimensionless profit rate and finite time exergoeconomic performance bound

The characteristics of maximum DPR ($\bar{\Pi}_{max}$) versus τ_1 , U_T and τ_2 are shown in Figs. 9–11, respectively. Similar characteristics for a and b are also analyzed by numerical calculations. One can see that, $\bar{\Pi}_{max}$ increases when τ_1 , a , b and U_T increase; the FTEPB ($\eta_{\bar{\Pi}}$) increases when τ_1 , b and U_T increase, and the tendency is reversed when a increases; τ_2 has an optimal value which leads to double maximum value of the DPR. The optimal TTC exists, which is different from the results obtained in Refs. Ust et al. (2007), Yilmaz et al. (2008), Hao and Zhang (2013) and Dubey et al. (2016). One can see that the design parameters have significant impacts on the performances of CHPP, which provides some opportunities to improve their performances.

5.4. Maximum exergy efficiency and the corresponding dimensionless profit rate

The variations of η_{max} and $\bar{\Pi}_\eta$ with U_T are shown in Fig. 12. From the figure, η_{max} and $\bar{\Pi}_\eta$ increase when U_T increases. Similar analyses show that η_{max} and $\bar{\Pi}_\eta$ increase when τ_1 and τ_2 increase. However, the CHPP becomes a pure heat supply plant when $\tau_2 > 1.44$, and the maximum η_{max} is vanished. Therefore, an appropriate τ_2 should be chosen in the designs of CHPP.

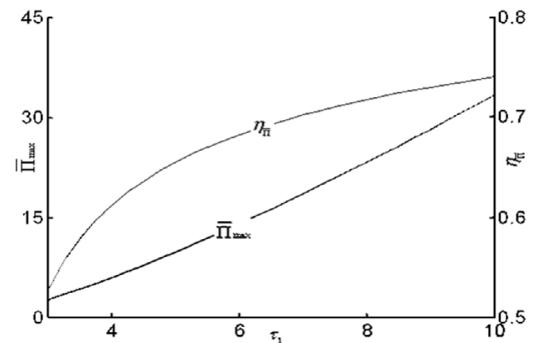


Fig. 9. Variations of $\bar{\Pi}_{max}$ and $\eta_{\bar{\Pi}}$ with τ_1 .

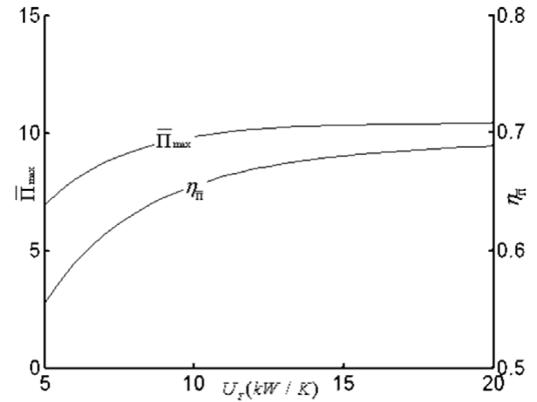


Fig. 10. Variations of $\bar{\Pi}_{max}$ and $\eta_{\bar{\Pi}}$ with U_T .

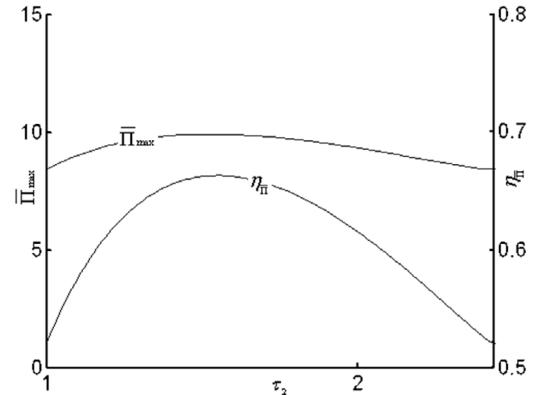


Fig. 11. Variations of $\bar{\Pi}_{max}$ and $\eta_{\bar{\Pi}}$ with τ_2 .

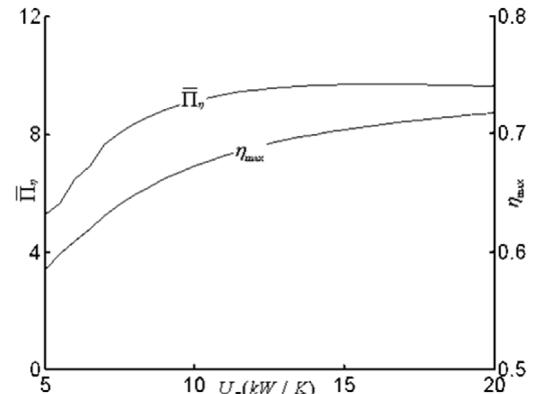


Fig. 12. Variations of η_{max} and $\bar{\Pi}_\eta$ with U_T .

6. Conclusions

A CHPP with RCCGT is investigated based on FIT theory. The HTLs, IALs and irreversible PDLs are considered. The expressions of DPR and EE of CHPP are derived. The HCDs and CPR are optimized, and the DPR and EE of CHPP are maximized. The results show that:

(1) The HCDs and CPR have their optimal values which lead to maximum value of DPR. The characteristic of maximum DPR and corresponding EE is leaf-shaped curve, and the FTEPB at the maximum DPR point is obtained. When $u_r \approx 0.1$, the optimization results derived by DPR and EE maximizations are equal to each other.

(2) The maximum DPR increases when τ_1, a, b and U_T increase. The corresponding EE increases with the decrease in a and increases in τ_1, b and U_T . The optimal TTC exists in this paper, which is different from the previous results. One can see that most of the design parameters have significant impacts on the optimal design variables derived by exergoeconomic performance optimizations, which illustrates the importance of parameter analyses.

(3) Moreover, η_{\max} and $\overline{\Pi}_\eta$ increase with the increases in τ_1, τ_2 and U_T . However, the CHPP becomes a pure heat supply plant when $\tau_2 > 1.44$, and the maximum η_{\max} is vanished. Therefore, an appropriate τ_2 should be chosen to avoid pure heat supply of CHPP.

Because the exergoeconomic performance of irreversible RC-CGT has not been investigated by the researchers before, the exergoeconomic performance optimization is conducted in this paper. Many new optimal results are obtained, which can offer new guidelines for CHPP designers.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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