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Exergoeconomic performance optimization of an endoreversible intercooled regenerated Brayton cogeneration plant
Part 1: Thermodynamic model and parameter analyses

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Abstract
A thermodynamic model of an endoreversible intercooled regenerative Brayton heat and power cogeneration plant coupled to constant-temperature heat reservoirs is established by using finite time thermodynamics in Part 1 of this paper. The heat resistance losses in the hot-, cold- and consumer-side heat exchangers, the intercooler and the regenerator are taken into account. The finite time exergoeconomic performance of the cogeneration plant is investigated. The analytical formulae about dimensionless profit rate and exergetic efficiency are derived. The numerical examples show that there exists an optimal value of intercooling pressure ratio which leads to an optimal value of dimensionless profit rate for the fixed total pressure ratio. There also exists an optimal total pressure ratio which leads to a maximum profit rate for the variable total pressure ratio. The effects of intercooling, regeneration and the ratio of the hot-side heat reservoir temperature to environment temperature on dimensionless profit rate and the corresponding exergetic efficiency are analyzed. At last, it is found that there exists an optimal consumer-side temperature which leads to a double-maximum dimensionless profit rate. The profit rate of the model cycle is optimized by optimal allocation of the heat conductance of the heat exchangers in Part 2 of this paper.

Keywords: Finite time thermodynamics, Endoreversible intercooled regenerative Brayton cogeneration plant, Exergoeconomic performance, Profit rate, Exergetic efficiency.

1. Introduction
The heat and power cogeneration plants are more advantageous in terms of energy and exergy efficiencies than plants which produce heat and power separately [1]. It is important to determine the optimal design parameters of the cogeneration plants. By using classical thermodynamics, Rosen *et al.* [2] performed energy and exergy analyses for cogeneration-based district energy systems, and exergy methods are employed to evaluated overall and component efficiencies and to identify and assess thermodynamic losses. Khaliq [3] performed the exergy analysis of a gas turbine trigeneration system for combined production of power heat and refrigeration and investigated the effects of overall pressure ratio, turbine inlet temperature and pressure drop on the exergy destruction. Reddy and Butcher [4] investigated the exergetic efficiency performance of a natural gas-fired intercooled reheat gas turbine cogeneration system and analyzed the effects of intercooling, reheat and total pressure ratio on the

performance of the cogeneration plant. Khaliq and Choudhary [5] evaluated the performance of intercooled reheat regenerative gas turbine cogeneration plant by using the first law (energetic efficiency) and second law (exergetic efficiency) of thermodynamics and investigated the effects of overall pressure ratio, cycle temperature ratio and pressure losses on the performance of the cogeneration plant. Vieira *et al.* [6] maximized the profit of a complex combined-cycle cogeneration plant using a professional process simulator which leading to a better compromise between energetic efficiency and cost, and the results of the exercises show that the optimal plant operating conditions depend nontrivially on the economic parameters, also the effects of exported steam mass flow rate and DMP (difference marketable price) on the optimal performances are discussed.

Finite-time thermodynamics (FTT) [7-18] is a powerful tool for analyzing and optimizing performance of various thermodynamic cycles and devices. In recent years, some authors have performed the performance analysis and optimization for various cogeneration plants by using finite-time thermodynamics. Bojic [19] investigated the annual worth of an endoreversible Carnot cycle cogeneration plant with the sole irreversibility of heat resistance. Sahin *et al.* [20] performed exergy output rate optimization for an endoreversible Carnot cycle cogeneration plant and found that the lower the consumer-side temperature, the better the performance. Erdil *et al.* [21] optimized the exergetic output rate and exergetic efficiency of an irreversible combined Carnot cycle cogeneration plant under various design and operating conditions and found that the optimal performance stayed approximately constant with consumer-side temperature. Atmaca *et al.* [22] performed the exergetic output rate, energy utilization factor (EUF), artificial thermal efficiency and exergetic efficiency optimization of an irreversible Carnot cycle cogeneration plant. Ust *et al.* [23] provided a new exergetic performance criterion, exergy density, which includes the consideration of the system sizes, and investigated the general and optimal performances of an irreversible Carnot cycle cogeneration plant. In industry, Brayton cycle is widely used and some authors are interested in the cogeneration plants composed of various Brayton cycles. Yilmaz [24] optimized the exergy output rate and exergetic efficiency of an endoreversible simple gas turbine closed-cycle cogeneration plant, investigated the effects of parameters on exergetic performance and found that the lower the consumer-side temperature, the better the performance. Hao and Zhang [25, 26] optimized the total useful-energy rate (including power output and useful heat rate output) and the exergetic output rate of an endoreversible Joule-Brayton cogeneration cycle by optimizing the pressure ratio and analyzed the effects of parameters on the optimal performances. Ust *et al.* [27, 28] proposed a new objective function called the exergetic performance coefficient (EPC), and optimized an irreversible regenerative gas turbine closed-cycle cogeneration plant with heat resistance and internal irreversibility [27] and an irreversible Dual cycle cogeneration plant with heat resistance, heat leakage and internal irreversibility [28].

Exergoeconomic (or thermoeconomic) analysis and optimization [29, 30] is a relatively new method that combines exergy with conventional concepts from long-run engineering economic optimization to evaluate and optimize the design and performance of energy systems. Salamon and Nitzan [31] combined the endoreversible model with exergoeconomic analysis for endoreversible Carnot heat engine with the only loss of heat resistance. It was termed as finite time exergoeconomic analysis [32-38] to distinguish it from the endoreversible analysis with pure thermodynamic objectives and the exergoeconomic analysis with long-run economic optimization. Furthermore, such a method has been extended to endoreversible Carnot heat engine with complex heat transfer law [39], universal endoreversible heat engine [40], generalized irreversible Carnot heat engine [41], generalized irreversible Carnot heat pump [42] and universal irreversible steady flow variable-temperature heat reservoir heat pump [43]. On the bases of Refs. [32-38]. Tao *et al.* [44, 45] performed the finite time exergoeconomic performance analysis and optimization for an endoreversible simple [44] and regenerative [45] gas turbine closed-cycle heat and power cogeneration plant coupled to constant temperature heat reservoirs by optimizing the heat conductance allocations among the hot-, cold- and consumer-side heat exchangers, the regenerator and the pressure ratio of the compressor.

As to now, there is no work concerning the finite time thermodynamic analysis and optimization for endoreversible intercooled regenerative Brayton cogeneration cycle in the open literatures. In this paper, a thermodynamic model of an endoreversible intercooled regenerative Brayton heat and power cogeneration plant coupled to constant-temperature heat reservoirs is established and the performance investigation is performed by using finite time exergoeconomic analysis. The intercooling process and the heat resistance losses in the hot-, cold-, consumer-side heat exchangers and the regenerator are taken into account. The analytical formulae about dimensionless profit rate and exergetic efficiency are deduced. The two cases with fixed and variable total pressure ratios are discussed, and the effects of design parameters on general and optimal performances of the cogeneration plant are analyzed by detailed numerical examples. The intercooling pressure ratio and the total pressure ratio are optimized, and the corresponding exergetic efficiency is obtained.

2. Cycle model

The *T-s* diagram of the heat and power cogeneration plant composed of an endoreversible intercooled regenerative Brayton closed-cycle coupled to constant-temperature heat reservoirs is shown in Figure 1. Processes 1-2 and 3-4 are isentropic adiabatic compression process in the low- and high-pressure compressors, while the process 5-6 is isentropic adiabatic expansion process in the turbine. Process 2-3 is an isobaric intercooling process in the intercooler. Process 4-7 is an isobaric absorbed heat process and process 6-8 is an isobaric evolved heat process in the regenerator. Process 7-5 is an isobaric absorbed heat process in the hot-side heat exchanger and process 9-1 is an isobaric evolved heat process in the cold-side heat exchanger. Process 8-9 is an isobaric evolved heat process in the consumer-side heat exchanger.

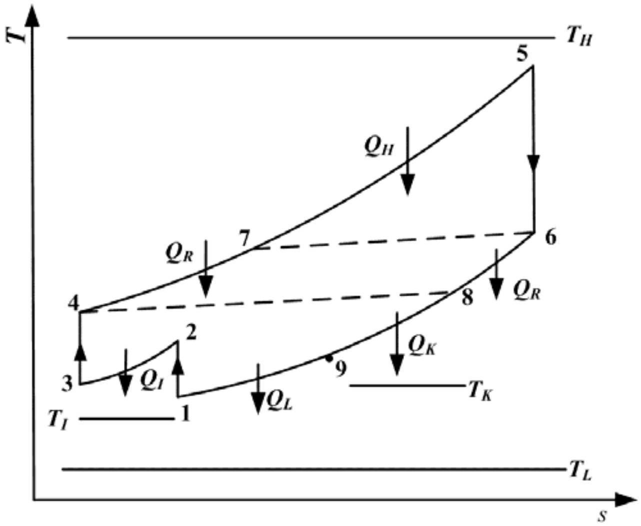

Figure 1. *T-s* diagram for the cycle model

Assuming that the working fluid used in the cycle is an ideal gas with constant thermal capacity rate (mass flow rate and specific heat product) C_{wf}. The hot-, cold- and consumer-side heat reservoir temperatures are T_H, T_L and T_K respectively, and the intercooling fluid temperature is T_I. The heat exchangers between the working fluid and the heat reservoirs, the regenerator and the intercooler are counter-flow. The conductances (heat transfer surface area and heat transfer coefficient product) of the hot-, cold- and consumer-side heat exchangers, the intercooler and the regenerator are U_H, U_L, U_K, U_I, U_R , respectively. According to the heat transfer processes, the properties of working fluid and the theory of heat exchangers, the rate (Q_H) of heat transfer from heat source to the working fluid, the rate (Q_L) of heat transfer from the working fluid to the heat sink, the rate (Q_K) of heat transfer from the working fluid to the heat consuming device, the rate (Q_I) of heat exchanged in the intercooler, and the rate (Q_R) of heat regenerated in the regenerator are, respectively, given by:

$$Q_H = U_H \frac{(T_5 - T_7)}{\ln\left[(T_H - T_7)/(T_H - T_5)\right]} = C_{wf}(T_5 - T_7) = C_{wf} E_H (T_H - T_7) \tag{1}$$

$$Q_L = U_L \frac{(T_9 - T_1)}{\ln\left[(T_9 - T_L)/(T_1 - T_L)\right]} = C_{wf}(T_9 - T_1) = C_{wf} E_L (T_9 - T_L) \tag{2}$$

$$Q_K = U_K \frac{(T_8 - T_9)}{\ln\left[(T_8 - T_K)/(T_9 - T_K)\right]} = C_{wf}(T_8 - T_9) = C_{wf} E_K (T_8 - T_K) \tag{3}$$

$$Q_I = U_I \frac{(T_2 - T_3)}{\ln\left[(T_2 - T_I)/(T_3 - T_I)\right]} = C_{wf}(T_2 - T_3) = C_{wf} E_I (T_2 - T_I) \tag{4}$$

$$Q_R = C_{wf}(T_7 - T_4) = C_{wf}(T_6 - T_8) = C_{wf} E_R (T_6 - T_4) \tag{5}$$

where E_H, E_L, E_K, E_I and E_R are the effectivenesses of the hot-, cold-, consumer-side heat exchangers, the intercooler and the regenerator, respectively, and are defined as:

$$\begin{gathered} E_H = 1 - \exp(-N_H),\ E_L = 1 - \exp(-N_L),\ E_K = 1 - \exp(-N_K) \\ E_I = 1 - \exp(-N_I),\ E_R = N_R/(N_R + 1) \end{gathered} \tag{6}$$

where $N_i (i = H, L, K, I, R)$ are the numbers of heat transfer units of the hot-, cold-, consumer-side heat exchangers, the intercooler and the regenerator, respectively, and are defined as: $N_i = U_i / C_{wf}$.
Defining that the working fluid isentropic temperature ratios for the low-pressure compressor and the total compression process are x and y, i.e. $x = T_2/T_1$, $y = T_5/T_6$. According to the properties of endoreversible cycle, one has:

$$x = \pi_1^{(k-1)/k},\ y = \pi^{(k-1)/k},\ T_4 = T_3 y x^{-1} \tag{7}$$

where π_1 is the intercooling pressure ratio which satisfies $\pi_1 \geq 1$, and π is the total pressure ratio which satisfies $\pi \geq \pi_1$. k is the specific heat ratio of working fluid.

3. Formulae about dimensionless profit rate and exergetic efficiency

Assuming that the environment temperature is T_0, the total rate of exergy input of the cogeneration plant is:

$$e_H = Q_H(1 - T_0/T_H) - Q_L(1 - T_0/T_L) - Q_I(1 - T_0/T_I) \tag{8}$$

According to the first law of thermodynamics, the power output (the exergy output rate of power) of the cogeneration plant is:

$$P = Q_H - Q_L - Q_I - Q_K \tag{9}$$

The entropy generation rate (σ) of the cogeneration plant is:

$$\sigma = Q_L/T_L + Q_I/T_I + Q_K/T_K - Q_H/T_H \tag{10}$$

From the exergy balance for the cogeneration plant, one has:

$$e_H = P + e_K + T_0\sigma \tag{11}$$

where e_K is thermal exergy output rate, i.e. the exergy output rate of process heat, and $T_0\sigma$ is the exergy loss rate.
Combining equations (8)-(11) yields the thermal exergy output rate:

$$e_K = Q_K(1 - T_0/T_K) \tag{12}$$

Assuming that the prices of exergy input rate, power output and thermal exergy output rate are φ_H, φ_P and φ_K, respectively, and the profit rate of cogeneration plant is defined as:

$$\Pi = \varphi_P P + \varphi_K e_K - \varphi_H e_H \tag{13}$$

when $\varphi_P = \varphi_K = \varphi_H$, equation (13) becomes:

$$\Pi = \varphi_P (P + e_K - e_H) = -\varphi_P T_0 \sigma \tag{14}$$

The maximum profit rate objective is equivalent to a minimum entropy generation rate objective in this case.
When $\varphi_P = \varphi_K$ and $\varphi_H / \varphi_P \to 0$, equation (13) becomes:

$$\Pi = \varphi_P (P + e_K) \tag{15}$$

The maximum profit rate objective is equivalent to a maximum total exergy output rate objective in this case.
Combining equations (1)-(5) with (7)-(12) yields the inlet temperature (T_1) of the low-pressure compressor:

$$T_1 = \frac{2yc_2c_3E_IT_I(c_1c_4 - c_4 + yE_R) + 2x(y - c_4E_R)(E_LT_L + c_3E_KT_K) + xc_1c_2c_3E_HT_H}{2x[y - c_4E_R + yc_2c_3c_5(c_4 - c_1c_4 - yE_R)]} \tag{16}$$

where $c_1 = 2(1 - E_R)$, $c_2 = 1 - E_K$, $c_3 = 1 - E_L$, $c_4 = 1 - E_H$, and $c_5 = 1 - E_I$.
The power output is:

$$P = \frac{\begin{array}{l} C_{wf}E_HT_K[xc_2c_3T_H(1 - E_R) - xy(T_1 - E_LT_L - c_3E_KT_K) + c_2c_3y^2E_R(xc_5T_1 + \\ E_IT_I)] - xc_4C_{wf}(1 - E_R)[c_2E_LT_K(T_1 - T_L) + c_2c_3E_IT_K(xT_1 - T_I)] - xc_4C_{wf} \\ E_KT_K(1 - E_R)(T_1 - E_LT_L - c_3T_K) \end{array}}{xc_2c_3c_4T_K(1 - E_R)} \tag{17}$$

The thermal exergy output rate is:

$$e_K = \frac{C_{wf}E_K(T_K - T_0)(T_1 - E_LT_L - c_3T_K)}{c_2c_3T_K} \tag{18}$$

Defining price ratios: $a = \varphi_P / \varphi_H$, $b = \varphi_K / \varphi_H$, and Π can be nondimensionalized by using $\varphi_H C_{wf} T_0$:

$$\overline{\Pi} = \frac{\varphi_P P + \varphi_K e_K - \varphi_H e_H}{\varphi_H C_{wf} T_0} = \frac{(a-1)P + (b-1)e_K - T_0\sigma}{C_{wf}T_0} \tag{19}$$

The exergetic efficiency (η_{ex}) is defined as the ratio of total exergy output rate to total exergy input rate:

$$\eta_{ex} = \frac{P + e_K}{e_H} = \frac{P + e_K}{P + e_K + T_0\sigma} \tag{20}$$

where

$$\begin{array}{l} \sigma = C_{wf}\{E_L(T_1 - T_L)/(c_3T_L) + E_I(xT_1 - T_I)/T_I + E_K(T_1 - E_LT_L - c_3T_K)/(c_2c_3T_K) - \\ \quad E_H[xc_2c_3T_H(1 - E_R) - xy(T_1 - E_LT_L - c_3E_KT_K) + c_2c_3y^2E_R(xc_5T_1 + E_IT_I)]/[xc_2c_3 \cdot \\ \quad c_4T_H(1 - E_R)]\} \end{array}$$

According to equation (19), the dimensionless profit rate ($\overline{\Pi}$) of the endoreversible intercooled regenerative Brayton cogeneration plant coupled to constant-temperature heat reservoirs is the function of the intercooling pressure ratio (π_1) and the total pressure ratio (π) when the other boundary condition parameters (T_H, T_L, T_I, T_K, T_0, C_{wf}, E_H, E_L, E_K, E_I, E_R) are fixed.

4. Numerical examples

To see how the parameters influence the dimensionless profit rate, detailed numerical examples are provided. Defining four temperature ratios: $\tau_1 = T_H/T_0$, $\tau_2 = T_L/T_0$, $\tau_3 = T_I/T_0$, and $\tau_4 = T_K/T_0$, which are the ratios of the hot-, cold- and consumer-side heat reservoir temperatures and intercooling fluid temperature to environment temperature, respectively. In the calculations, $k = 1.4$, $C_{wf} = 1.0kW/K$, $\tau_2 = \tau_3 = 1$ and $\tau_4 = 1.2$ are set. According to analysis in Ref. [46], $a = 10$ and $b = 6$ are set.

4.1 The total pressure ratio is fixed

Assuming that $\pi = 18\,(1 < \pi_1 \le 18)$. The effect of E_R on the characteristic of $\overline{\Pi}$ versus π_1 with $E_H = E_L = E_I = E_K = 0.8$ and $\tau_1 = 5.0$ is shown in Figure 2. The effect of E_I on the characteristic of $\overline{\Pi}$ and η_{ex} versus π_1 with $E_H = E_L = E_R = E_K = 0.8$ and $\tau_1 = 5.0$ is shown in Figure 3.

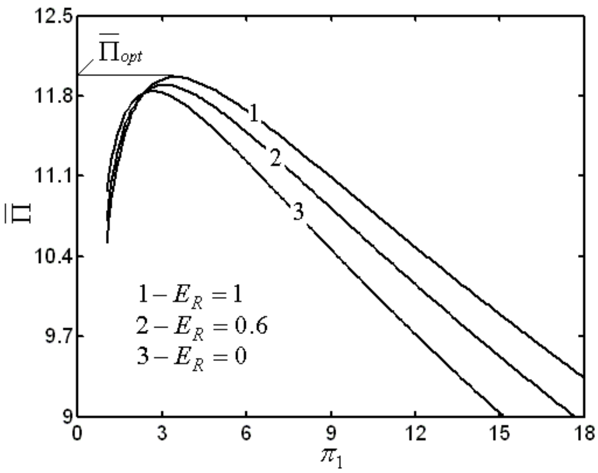

Figure 2. Effect of E_R on the characteristic of $\overline{\Pi}$ versus π_1

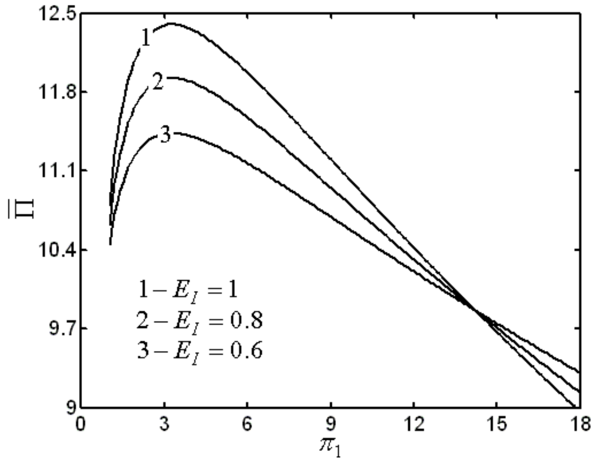

Figure 3. Effect of E_I on the characteristic of $\overline{\Pi}$ versus π_1

It can be seen from Figure 2 that there exists an optimal value of intercooling pressure ratio ($(\pi_1)_{\overline{\Pi}_{opt}}$) which corresponds to an optimal value of dimensionless profit rate ($\overline{\Pi}_{opt}$). Also there exists a critical intercooling pressure ratio ($(\pi_1)_{c1}$). When $\pi_1 < (\pi_1)_{c1}$, the calculation illustrates that the outlet temperature of turbine is lower than the outlet temperature of high-pressure compressor, i.e. $T_6 < T_4$, and the regenerative process will lead to heat loss in this case, and $\overline{\Pi}$ decreases with the increase of E_R. When $\pi_1 > (\pi_1)_{c1}$, one has $T_6 > T_4$, and $\overline{\Pi}$ increases with the increase of E_R. The calculation illustrates that when the fixed π is large, the critical point ($(\pi_1)_{c1}$) will reach the right-side of the curve.

It can be seen from Figure 3 that there exists another critical intercooling pressure ratio ($(\pi_1)_{c2}$). The calculation illustrates that with the increase of E_I, e_K decreases rapidly, e_H increases rapidly, and P changes slowly. When $\pi_1 > (\pi_1)_{c2}$, $\overline{\Pi}$ decreases with the increase of E_I. When $\pi_1 < (\pi_1)_{c2}$, $\overline{\Pi}$ increases with the increase of E_I. The calculation illustrates that no matter that the fixed π is large or small, the critical point ($(\pi_1)_{c2}$) will be always at the right-side of the peak value of the curve.

4.2 The total pressure ratio is variable

The effects of τ_1 on the characteristics of the optimal dimensionless profit rate ($\overline{\Pi}_{opt}$) and the corresponding exergetic efficiency ($(\eta_{ex})_{\overline{\Pi}_{opt}}$) versus π is shown in Figure 4. It can be seen that there exists an optimal value of total pressure ratio ($(\pi)_{\overline{\Pi}_{max}}$) (The value of the intercooling pressure ratio is also optimal in this case) which corresponds to a maximum value of dimensionless profit rate ($\overline{\Pi}_{\max}$). $(\eta_{ex})_{\overline{\Pi}_{opt}}$ also exists a extremum with respect to π. With the increase of τ_1, $\overline{\Pi}_{opt}$ and $(\eta_{ex})_{\overline{\Pi}_{opt}}$ increase. Figure 5 shows the effect of τ_1 on the characteristic of the optimal intercooling pressure ratio ($(\pi_1)_{\overline{\Pi}_{opt}}$) versus π. It indicates that $(\pi_1)_{\overline{\Pi}_{opt}}$ increases with the increase of π, and approximately stays constant for different τ_1.

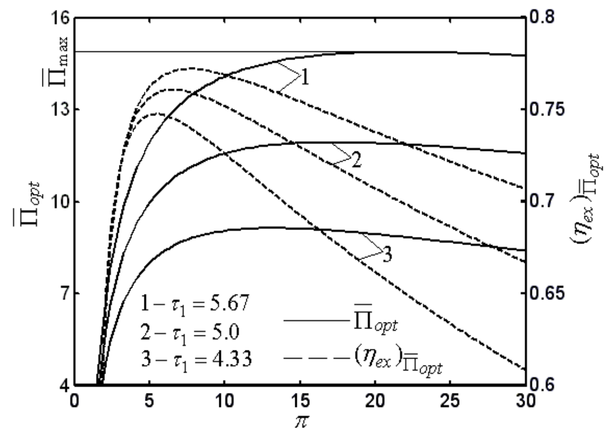

Figure 4. Effects of τ_1 on the characteristics of $\overline{\Pi}_{opt}$ and $(\eta_{ex})_{\overline{\Pi}_{opt}}$ versus π

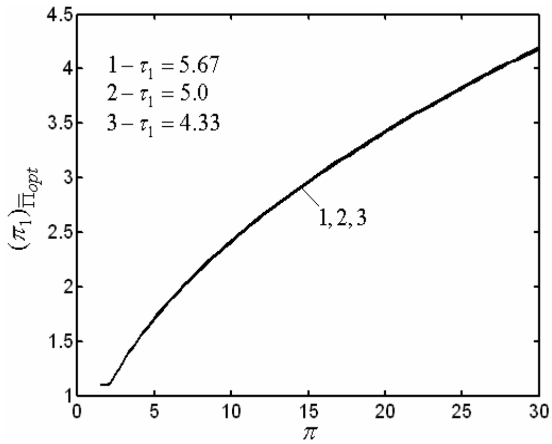

Figure 5. Effect of τ_1 on the characteristic of $(\pi_1)_{\overline{\Pi}_{opt}}$ versus π

Figure 6. shows the effects of E_R on the characteristics of $\overline{\Pi}_{opt}$ and $(\eta_{ex})_{\overline{\Pi}_{opt}}$ versus π. It can be seen that there exists a critical total pressure ratio (π_c). When $\pi < \pi_c$, $\overline{\Pi}_{opt}$ increases with the increase of E_R. When $\pi > \pi_c$, the calculation illustrates that the outlet temperature of turbine is lower than the outlet temperature of high-pressure compressor, i.e. $T_6 < T_4$, and the regenerative process will lead to heat loss in this case, $\overline{\Pi}_{opt}$ decreases with the increase of E_R. The effect of E_R on $(\eta_{ex})_{\overline{\Pi}_{opt}}$ is similar to that of E_R on $\overline{\Pi}_{opt}$. Figure 7 shows the effect of E_R on the characteristic of $(\pi_1)_{\overline{\Pi}_{opt}}$ versus π. It indicates that $(\pi_1)_{\overline{\Pi}_{opt}}$ increases with the increase of E_R.

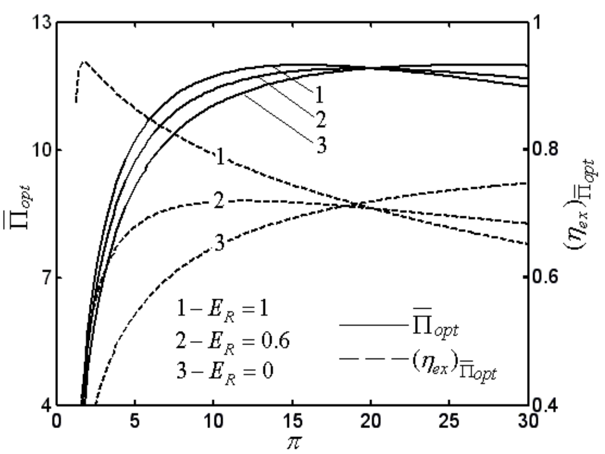

Figure 6. Effects of E_R on the characteristics of $\overline{\Pi}_{opt}$ and $(\eta_{ex})_{\overline{\Pi}_{opt}}$ versus π

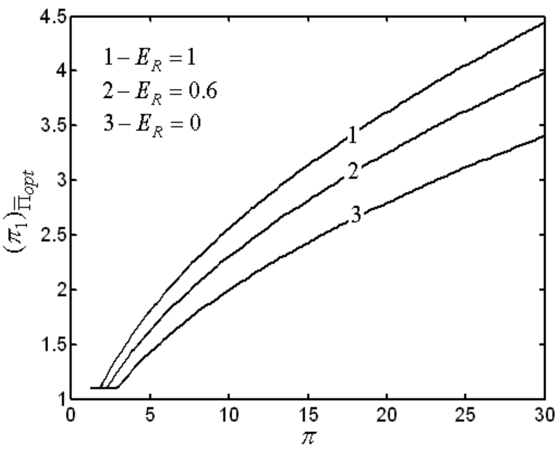

Figure 7. Effect of E_R on the characteristic of $(\pi_1)_{\overline{\Pi}_{opt}}$ versus π

4.3 Dimensionless profit rate versus exergetic efficiency characteristic

Figure 8 shows the characteristic of $\overline{\Pi}_{opt}$ versus $(\eta_{ex})_{\overline{\Pi}_{opt}}$ with $E_H = E_L = E_I = E_R = E_K = 0.8$ and $\tau_1 = 5.0$. One can find that the characteristic is loop-shaped. There exist a maximum dimensionless profit rate ($\overline{\Pi}_{\max}$) and the corresponding exergetic efficiency ($(\eta_{ex})_{\overline{\Pi}_{\max}}$), and $(\eta_{ex})_{\overline{\Pi}_{\max}}$ is termed as the finite time exergoeconomic performance limit to distinguish it from the finite time thermodynamic performance limit at maximum thermodynamic output. The calculation illustrates that the curve is always not closed.

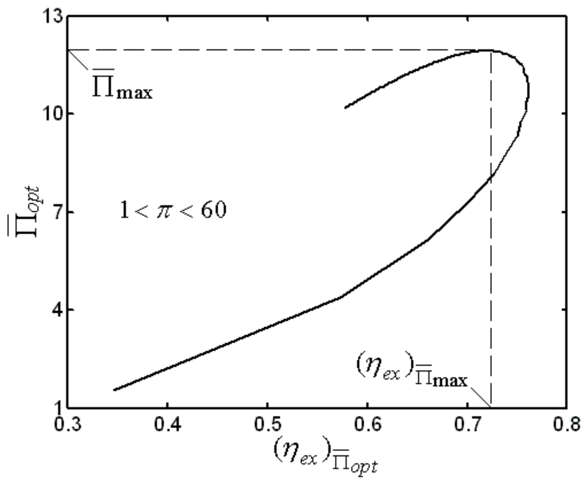

Figure 8. Characteristic of $\overline{\Pi}_{opt}$ versus $(\eta_{ex})_{\overline{\Pi}_{opt}}$

4.4 The effect of consumer-side temperature

It can be seen from equation (19) that the effect of consumer-side temperature (τ_4) on exergoeconomic performance of the cogeneration plant is complex. Figures 9 and 10 show the characteristics of the maximum dimensionless profit rate ($\overline{\Pi}_{\max}$), the corresponding exergetic efficiency ($(\eta_{ex})_{\overline{\Pi}_{\max}}$), the optimal total pressure ratio ($\pi_{\overline{\Pi}_{\max}}$) and the optimal intercooling pressure ratio ($(\pi_1)_{\overline{\Pi}_{\max}}$) versus τ_4 with $E_H = E_L = E_I = E_R = E_K = 0.8$ and $\tau_1 = 5.0$. It can be seen from Figure 9 that there exists an optimal value of consumer-side temperature which corresponds to a double-maximum value of dimensionless profit rate. $(\eta_{ex})_{\overline{\Pi}_{\max}}$ also exists a extremum with respect to τ_4. It can be seen from Figure 10 that with the increase of τ_4, $(\pi_1)_{\overline{\Pi}_{\max}}$ decreases, and $\pi_{\overline{\Pi}_{\max}}$ increases first, and then decreases, but the value of $\pi_{\overline{\Pi}_{\max}}$ changes slightly.

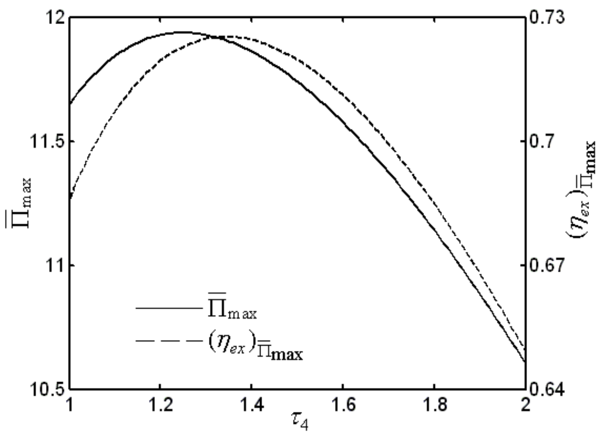

Figure 9. Characteristics of $\overline{\Pi}_{\max}$ and $(\eta_{ex})_{\overline{\Pi}_{\max}}$ versus τ_4

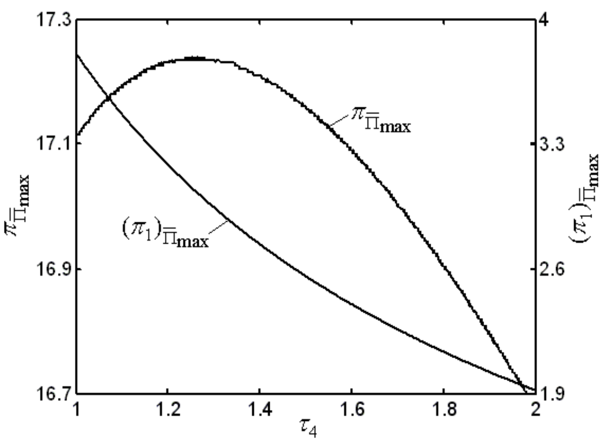

Figure 10. Characteristics of $\pi_{\overline{\Pi}_{\max}}$ and $(\pi_1)_{\overline{\Pi}_{\max}}$ versus τ_4

5. Conclusion

Finite time exergoeconomic analyses is applied to investigate the exergoeconomic performance of an endoreversible intercooled regenerative Brayton cogeneration plant coupled to constant-temperature heat reservoirs. Analytical formulae about dimensionless profit rate and exergetic efficiency are derived. The effects of intercooling and regeneration on the general and optimal exergoeconomic performance of the cogeneration cycle are different with the changes of pressure ratios, and it is found that there exist the critical intercooling pressure ratio and the critical total pressure ratio. Also the optimal intercooling pressure ratio, the optimal total pressure ratio and corresponding exergetic efficiency are obtained. Dimensionless profit rate versus exergetic efficiency characteristic is studied and the characteristic is loop-shaped. At last, the effect of consumer-side temperature on the exergoeconomic performance is analyzed and it is found that there exists an optimal consumer-side temperature which leads to a double-maximum dimensionless profit rate. The results obtained in this paper may provide some guidelines for the optimal design and parameters selection of practical gas turbine cogeneration plant. The dimensionless profit rate of the model cycle will be optimized by optimal allocation of the heat conductance of the heat exchangers in Part 2 of this paper [47].

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Nomenclature

a	price ratio of power output to exergy input rate
b	price ratio of thermal exergy output rate to exergy input rate
C	heat capacity rate (kW/K)
E	effectiveness of the heat exchanger
e	exergy flow rate (kW)
k	ratio of the specific heats
N	number of heat transfer units
P	power output of the cycle (kW)
Q	rate of heat transfer (kW)
s	entropy (kJ/K)
T	temperature (K)
U	heat conductance (kW/K)
x	isentropic temperature ratio for the low-pressure compressor
y	isentropic temperature ratio for the total compression process

Greek symbols

φ	price of exergy flow rate ($dollar/kW$)
η	efficiency
Π	profit rate ($dollar$)
π_1	intercooling pressure ratio
π	total pressure ratio
σ	entropy generation rate of the cycle (kW/K)
τ_1	ratio of the hot-side heat reservoir temperature to environment temperature
τ_2	ratio of the cold-side heat reservoir temperature to environment temperature
τ_3	ratio of the intercooling fluid temperature to environment temperature
τ_4	ratio of the consumer-side temperature to environment temperature

Subscripts

c	critical value
ex	exergy
H	hot-side
I	intercooler

K	consumer-side
L	cold-side
max	maximum
opt	optimal
R	regenerator
wf	working fluid
0	ambient
$1,2,3,4,5,6,7,8,9$	state points of the cycle
$^{-}$	dimensionless

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