Exergoeconomic performance optimization for an endoreversible regenerative gas turbine closed-cycle cogeneration plant

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Finite time exergoeconomic performance of an endoreversible regenerative gas turbine closed-cycle cogeneration plant coupled to constant temperature heat reservoirs is investigated. The analytical formulae about profit rate and exergy efficiency of the cogeneration plant with the heat resistance losses in the hot-, cold- and consumer-side heat exchangers and the regenerator are deduced, respectively. By means of numerical calculations, the heat conductance allocation among the four heat exchangers and pressure ratio of the compressor are optimized by taking the maximum profit rate as the objective. The characteristic of optimal dimensionless profit rate versus corresponding exergy efficiency is investigated and the effects of design parameters on optimal performance of the cogeneration plant are also analyzed. The results show that there exist a sole group of optimal heat conductance allocations among the four heat exchangers and an optimal pressure ratio of the compressor which lead to the maximum dimensionless profit rate, and there exists an optimal consumer-side temperature which leads to double-maximum dimensionless profit rate.

Keywords: Finite time thermodynamics; gas turbine cycle cogeneration plant; exergoeconomic performance; profit rate.

Se investigó el desempeño en tiempo finito de una planta de cogeneración de gas turbina endoreversible regenerativo de ciclo cerrado acoplado con una reserva de temperatura termal constante. Se deduce la formula analítica para la razón de ganancia y eficiencia de energía de la planta de cogeneración con las perdidas de resistencia en calor, frío y lado-consumidor intercambiadores termales y regeneradores, respectivamente. Mediante cálculos numéricos, la asignación de conductor termal entre los cuatro intercambiadores termales y de taza de presión del compresor es optimizada tomando como objetivo la máxima taza de ganancia. Se investiga las características de las dimensiones óptimas de taza de ganancia versus la energía eficiencia correspondiente, así como los efectos de los parámetros sobre el desempeño óptimo de la planta de cogeneración. Los resultados demuestran que existe un solo grupo de óptimo de conductores termales entre los cuatro intercambiadores y una de taza de presión del compresor optima que resultan en las dimensiones máximas de la taza de ganancia, y que existe una temperatura optima de lado-consumidor que resulta con una taza de ganancia de dimensiones doble-máxima.

Descriptores: Termodinámica de tiempo finito; desempeño exorgoeconómico.

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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 optimal design parameters of the heat and power cogeneration plants. Finite-time thermodynamics [2-13] is a powerful tool for analyzing and optimizing various thermodynamic cycles and devices. Some authors have analyzed and optimized the performance for various cogeneration plants with different optimization objectives by using finite-time thermodynamics. Bojic [14] analyzed the annual worth of an endoreversible Carnot cycle cogeneration plant. Sahin *et al.* [15] optimized exergy output rate of an endoreversible Carnot cycle cogeneration plant. Erdil *et al.* [16] investigated the optimal exergetic performance parameters of an irreversible combined Carnot cycle cogeneration plant. Yilmaz [17] optimized the exergy output rate of an endoreversible simple gas turbine closed-cycle cogeneration plant. Hao *et al.* [18] optimized the total useful-energy rate (including power and useful heat rate) of an endoreversible simple gas turbine closed-cycle cogeneration plant. Ust *et al.* [19,20] optimized ecological coefficient of performance (ECOP) of an irreversible regenerative gas turbine closed-cycle cogeneration plant [19] and an irreversible Dual cycle cogeneration plant [20].

Exergoeconomic (or thermoeconomic) analysis and optimization [21,22], combining exergy with conventional concepts from long-run engineering economic optimization, is a relatively new method to evaluate and optimize the performance of energy systems. Salamon and Nitzan [23] combined the endoreversible model [2-13] with exergoeconomic analysis [21,22] for endoreversible Carnot heat engine. It was termed as finite time exergoeconomic analysis [24-30] to distinguish it from the endoreversible analysis with pure thermodynamic objectives and the exergoeconomic analysis with long-run economic optimization. Similarly, the performance bound at maximum profit rate was termed as finite time exergoeconomic performance bound to distinguish it from the finite time thermodynamic performance bound at maximum thermodynamic output (maximum power output for heat engines, maximum cooling load for refrigerators, and maximum heating load for heat pumps) and the reversible thermodynamic performance bound at reversible operation point. The finite time exergoeconomic performance analysis and optimization has been performed for quantum heat engine [31], generalized irreversible Carnot heat engine [32] and refrigerator [33], universal heat engine [34], three-heat-reservoir refrigerator [35] and heat pump [36]. A similar idea was provided by Ibrahim *et al.* [37], De Vos [38], Bejan [39], and Sahin *et al.* [40].

A finite time exergoeconomic performance optimization for gas turbine cycle cogeneration plants does not appear to have been published. The pure thermodynamic index was taken as optimization objective for the gas turbine closed-cycle cogeneration plants in the previous works [17-19]. Therefore, this paper introduces finite time exergoeconomic analysis and optimization to an endoreversible regenerative gas turbine closed-cycle cogeneration plant with the heat resistance losses in the hot-, cold- and consumer-side heat exchangers and the regenerator coupled to constant temperature heat reservoirs. The analytical formulae about profit rate and exergy efficiency of the cogeneration plant are deduced. By means of numerical calculations, the heat conductance allocation and pressure ratio are optimized by taking the maximum profit rate as the objective. The characteristic of optimal dimensionless profit rate and corresponding exergy efficiency is obtained and the effects of design parameters on optimal performance of the cogeneration plant are also analyzed.

2. Cycle model

The T-s diagram of cogeneration plant composed of an endoreversible regenerative gas turbine closed-cycle is shown in Fig.1. Parameters T_H, T_L and T_K are the hot-, cold- and consumer-side reservoir temperatures, respectively. Processes 1-2 and 4-5 are isentropic adiabatic compression and expansion processes in the compressor and turbine. Process 2-3 is an isobaric absorbed heat process in the regenerator and process 5-6 is an isobaric released heat process in the regenerator. Process 3-4 is an isobaric absorbed heat process in the hot-side heat exchanger and process 7-1 is an isobaric evolved heat process in the cold-side heat exchanger.

It is assumed 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 heat exchangers between the working fluid and the heat reservoirs and the regenerator are counter flow. When the heat transfer obeys Newton's law, according to 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, and the rate (Q_R) of heat regenerated in the regenerator are, respectively:

$$Q_H = C_{wf}(T_4 - T_3) = C_{wf}\varepsilon_H(T_H - T_3) \tag{1}$$

$$Q_L = C_{wf}(T_7 - T_1) = C_{wf}\varepsilon_L(T_7 - T_L) \tag{2}$$

$$Q_K = C_{wf}(T_6 - T_7) = C_{wf}\varepsilon_K(T_6 - T_K) \tag{3}$$

$$\begin{aligned} Q_R &= C_{wf}(T_3 - T_2) = C_{wf}(T_5 - T_6) \\ &= C_{wf}\varepsilon_R(T_5 - T_2) \end{aligned} \tag{4}$$

where ε_H, ε_L, ε_K, and ε_R are the effectivenesses of the hot-side, cold-side, consumer-side heat exchangers and the regenerator, and are defined as:

$$\begin{aligned} &\varepsilon_H = 1 - e^{-N_H}, \quad \varepsilon_L = 1 - e^{-N_L}, \\ &\varepsilon_K = 1 - e^{-N_K}, \quad \varepsilon_R = N_R/(N_R + 1) \end{aligned} \tag{5}$$

where N_H, N_L, N_K, and N_R are the numbers of heat transfer units of the hot-side, cold-side, consumer-side heat exchangers and the regenerator, and are defined as:

$$\begin{aligned} &N_H = U_H/C_{wf}, \quad N_L = U_L/C_{wf}, \\ &N_K = U_K/C_{wf}, \quad N_R = U_R/C_{wf} \end{aligned} \tag{6}$$

where U_H, U_L, U_K and U_R are the heat conductances (heat transfer surface area and heat transfer coefficient product) of the hot-, cold- and consumer-side heat exchangers and the regenerator, respectively.

According to the second law of thermodynamics, one has:

$$T_2/T_1 = T_4/T_5 = x \tag{7}$$

where $x = \pi^{(k-1/k)}$ is the isentropic temperature ratio of the cycle, where π is the pressure ratio of the compressor, and k is the ratio of specific heats.

3. Performance analytical relations

Combining equations (1)-(4) and (7) gives:

$$T_3 = \frac{[x^{-1}\varepsilon_R + (1-\varepsilon_L)(1-\varepsilon_K)(1-2\varepsilon_R)]\varepsilon_H T_H + x(1-\varepsilon_R)[(1-\varepsilon_L)\varepsilon_K T_K + \varepsilon_L T_L]}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R) + x\varepsilon_R] - (1-\varepsilon_H)x^{-1}\varepsilon_R} \tag{8}$$

$$T_7 = \frac{(1-\varepsilon_K)(1-\varepsilon_R)x^{-1}\varepsilon_H T_H + [1-(1-\varepsilon_H)x^{-1}\varepsilon_R]\varepsilon_K T_K + (1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R) + x\varepsilon_R]\varepsilon_L T_L}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R) + x\varepsilon_R] - (1-\varepsilon_H)x^{-1}\varepsilon_R} \tag{9}$$

$$T_6 = \frac{(1-\varepsilon_R)x^{-1}\varepsilon_H T_H + [(1-\varepsilon_H)(1-2\varepsilon_R) + x\varepsilon_R][(1-\varepsilon_L)\varepsilon_K T_K + \varepsilon_L T_L]}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R) + x\varepsilon_R] - (1-\varepsilon_H)x^{-1}\varepsilon_R} \tag{10}$$

$$Q_H = \frac{C_{wf}\varepsilon_H\{[1-(1-\varepsilon_L)(1-\varepsilon_K)(1-2\varepsilon_R + x\varepsilon_R) - x^{-1}\varepsilon_R]T_H - x(1-\varepsilon_R)[(1-\varepsilon_L)\varepsilon_K T_K + \varepsilon_L T_L]\}}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R) + x\varepsilon_R] - x^{-1}\varepsilon_R(1-\varepsilon_H)} \tag{11}$$

$$Q_L = \frac{\begin{array}{l} C_{wf}\varepsilon_L\{(1-\varepsilon_K)(1-\varepsilon_R)x^{-1}\varepsilon_H T_H + [1-(1-\varepsilon_H)x^{-1}\varepsilon_R]\varepsilon_K T_K \\ -\{1-(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R\}T_L\} \end{array}}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R} \tag{12}$$

$$Q_K = \frac{\begin{array}{l} C_{wf}\varepsilon_K\{(1-\varepsilon_R)x^{-1}\varepsilon_H T_H - \{1-(1-\varepsilon_L)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R] \\ -(1-\varepsilon_H)x^{-1}\varepsilon_R\}T_K + [(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]\varepsilon_L T_L\} \end{array}}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R} \tag{13}$$

According to the first law, the power output of cogeneration cycle is:

$$P = Q_H - Q_L - Q_K \tag{14}$$

and the exergy flow rate of power output is:

$$E_P = P \tag{15}$$

The net thermal exergy flow to the cogeneration plant is:

$$E_H = Q_H(1 - T_0/T_H) - Q_L(1 - T_0/T_L) \tag{16}$$

where T_0 is the temperature of environment.

The exergy flow rate into a control volume minus the exergy flow rate out of the volume is equal to the rate of destruction of exergy inside the volume. One has:

$$E_H = E_P + E_K + T_0\sigma \tag{17}$$

where E_K is thermal exergy output rate, *i.e.* the exergy output rate of process heat, and $\sigma = Q_L/T_L + Q_K/T_K - Q_H/T_H$ is the entropy generation rate of the cycle.

The thermal exergy output rate can be written from Eqs. (16) and (17) as:

$$E_K = Q_K(1 - T_0/T_K) \tag{18}$$

In the previous papers [15-17,19,20], the thermal exergy output rate is defined as:

$$E_K = Q_K(1 - T_0/T_a) \tag{19}$$

where $T_a = (T_6 - T_7)/(\ln T_6/T_7)$ is the average temperature in process 6-7.

Comparing Eq. (18) with (19), one can find that the exergy loss rate of the process heat is added to the exergy output rate of process heat in Eq. (19). The question of whether Eq. (18) or (19) is the correct one to apply depends on the way in which the control volume is defined. If the control volume includes the heat transfer process, then Eq. (18) applies, if the control volume does not include the heat transfer process then Eq. (19) applies. In the discussed problem, heat transfer process is included. Therefore, the thermal exergy output rate should be Eq. (18).

Assuming that the prices of exergy input rate, thermal exergy output rate and power output be ϕ_H, ϕ_K and ϕ_P, respectively, the profit rate of cogeneration cycle is:

$$\Pi = \phi_P E_P + \phi_K E_K - \phi_H E_H \tag{20}$$

Combining Eqs. (11)-(16), (18) and (20), Π can be nondimensionalized by using $\phi_H C_{wf} T_0$:

$$\overline{\Pi} = \frac{\begin{array}{l} a(x-1)\{[1-(1-\varepsilon_L)(1-\varepsilon_K)(1-\varepsilon_R+x\varepsilon_R)]x^{-1}\varepsilon_H\tau_1 + x^{-1}[\varepsilon_R(1-\varepsilon_H)(1-x) \\ -x\varepsilon_H][(1-\varepsilon_L)\varepsilon_K\tau_2+\varepsilon_L\tau_3]\} + b\varepsilon_K(1-\tau_2^{-1})\{(1-\varepsilon_R)x^{-1}\varepsilon_H\tau_1 - \{1-(1-\varepsilon_L)[(1 \\ -\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R\}\tau_2 + [(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]\varepsilon_L\tau_3\} - \varepsilon_H \\ (1-\tau_1^{-1})\{[1-(1-\varepsilon_L)(1-\varepsilon_K)(1-2\varepsilon_R+x\varepsilon_R)-x^{-1}\varepsilon_R]\tau_1 - x(1-\varepsilon_R)[(1-\varepsilon_L)\varepsilon_K \\ \tau_2+\varepsilon_L\tau_3]\} + \varepsilon_L(1-\tau_3^{-1})\{(1-\varepsilon_K)(1-\varepsilon_R)x^{-1}\varepsilon_H\tau_1 + [1-(1-\varepsilon_H)x^{-1}\varepsilon_R]\varepsilon_K\tau_2 \\ -\{1-(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R\}\tau_3\} \end{array}}{1-(1-\varepsilon_L)(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R} \tag{21}$$

where $a = \phi_P/\phi_H$ and $b = \phi_K/\phi_H$ are price ratios, $\tau_1 = T_H/T_0$, $\tau_2 = T_K/T_0$ and $\tau_3 = T_L/T_0$ are temperature ratios.

The exergy efficiency is defined as the ratio of total exergy output rate to total exergy input rate:

$$\eta = (E_P + E_K)/E_H \tag{22}$$

Substituting Eqs. (11)-(16) and (18) into Eq. (22) yields:

$$\eta = \frac{\begin{array}{l} (x-1)\{[1-(1-\varepsilon_L)(1-\varepsilon_K)(1-\varepsilon_R+x\varepsilon_R)]x^{-1}\varepsilon_H\tau_1 + x^{-1}[\varepsilon_R(1-\varepsilon_H)(1-x) \\ -x\varepsilon_H][(1-\varepsilon_L)\varepsilon_K\tau_2+\varepsilon_L\tau_3]\} + \varepsilon_K(1-\tau_2^{-1})\{(1-\varepsilon_R)x^{-1}\varepsilon_H\tau_1 - \{1-(1-\varepsilon_L)[(1-\varepsilon_H) \\ (1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R\}\tau_2 + [(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]\varepsilon_L\tau_3\} \end{array}}{\begin{array}{l} \varepsilon_H(1-\tau_1^{-1})\{[1-(1-\varepsilon_L)(1-\varepsilon_K)(1-2\varepsilon_R+x\varepsilon_R)-x^{-1}\varepsilon_R]\tau_1 - x(1-\varepsilon_R)[(1-\varepsilon_L)\varepsilon_K\tau_2+\varepsilon_L\tau_3]\} \\ -\varepsilon_L(1-\tau_3^{-1})\{(1-\varepsilon_K)(1-\varepsilon_R)x^{-1}\varepsilon_H\tau_1 + [1-(1-\varepsilon_H)x^{-1}\varepsilon_R]\varepsilon_K\tau_2 \\ -\{1-(1-\varepsilon_K)[(1-\varepsilon_H)(1-2\varepsilon_R)+x\varepsilon_R]-(1-\varepsilon_H)x^{-1}\varepsilon_R\}\tau_3\} \end{array}} \tag{23}$$

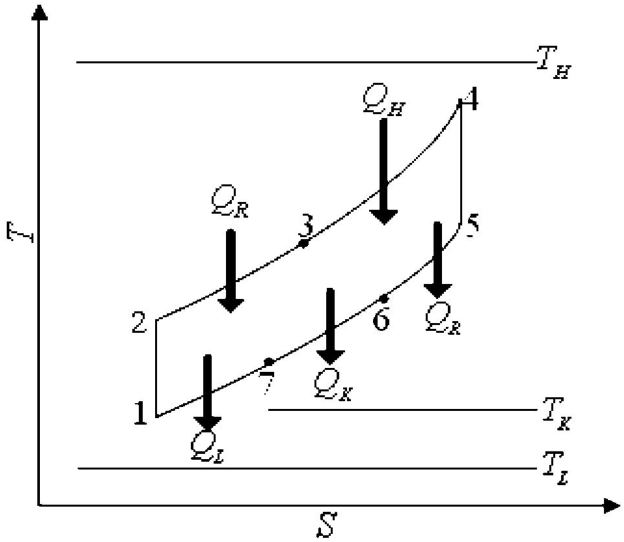

FIGURE 1. The cogeneration plant model composed of an endoreversible regenerative gas turbine closed-cycle.

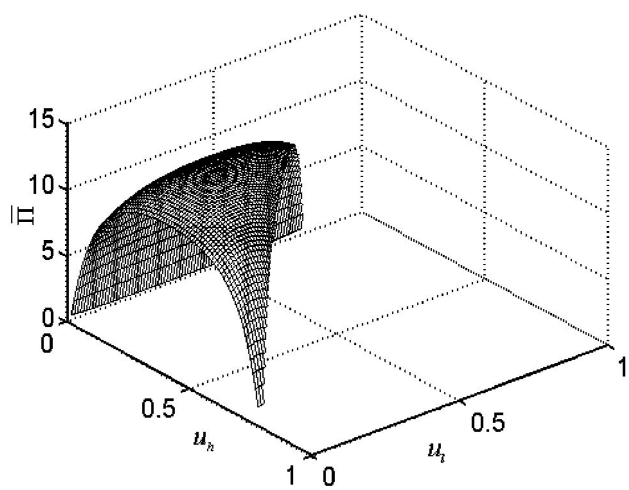

FIGURE 2. The three-dimensional characteristic of $\overline{\Pi}$ versus (u_h, u_l).

In order that the cogeneration plant is guaranteed to be profitable, the price per unit of power must be greater or equal to the price per unit of input heat exergy and that the price per unit of output heat exergy must be greater than the price per unit of input heat exergy, $\phi_P \geq \phi_H$ and $\phi_K \geq \phi_H$ hold.

When $\varepsilon_R = 0$, Eqs. (21) and (23) become the profit rate and exergy efficiency of an endoreversible simple gas turbine cycle cogeneration plant. Moreover, Eq. (23) is different from that of Ref. 17 owing to the modified calculation of thermal exergy output rate.

When $\phi_P = \phi_K = \phi_H$, equation (20) becomes:

$$\Pi = \phi_P(E_P + E_K - E_H) = -\phi_P T_0 \sigma \tag{24}$$

The maximum profit rate objective is equivalent to a minimum entropy generation rate objective in this case.

When $\phi_P = \phi_K$ and $\phi_P/\phi_H \to \infty$, Eq. (20) becomes:

$$\Pi = \phi_P(E_P + E_K) \tag{25}$$

The maximum profit rate objective is equivalent to a maximum total exergy production objective in this case.

4. Performance optimization

In the practical design, U_H, U_L, U_K, U_R, and π are changeable. For the fixed pressure ratio (isentropic temperature ratio), there exist a group of optimum distributions among the heat conductance of hot-side, cold-side, consumer-side heat exchangers and the regenerator for the fixed total heat exchanger inventory, which lead to the optimum dimensionless profit rate ($\overline{\Pi}_{opt}$). The optimum pressure ratio and a group of optimum distributions lead to the maximum optimum (double-maximum) dimensionless profit rate ($\overline{\Pi}_{\max}$). They may be determined using numerical calculation. The constraint on total heat exchanger conductance implies that the cost per unit of conductance is the same for each heat exchanger, which is probably not the case in practice, since different materials may be used or because the conductivity of the external fluids may be much higher than that of the working fluid [39]. For simplifying the problem, the constraint on total heat exchanger conductance is used as for regenerated gas turbine closed cycles [41-51]. The similar optimizations were performed for power, efficiency, ecological and power density optimization of regenerated gas turbine closed-cycle power plant [41-51].

According to Eq. (21), the dimensionless profit rate $\overline{\Pi}$ is the function of a, b, τ_1, τ_2, τ_3, U_H, U_L, U_K and U_R. Assuming that the total heat exchange inventory U_T ($U_T = U_H + U_L + U_K + U_R$) is fixed, a group of heat conductance allocations are defined as:

$$u_h = U_H/U_T, \quad u_l = U_L/U_T,$$
$$u_k = U_K/U_T, \quad u_r = U_R/U_T \tag{26}$$

Obviously, u_h, u_l, u_k and u_r should be satisfied:

$$0 < u_h < 1, \quad 0 < u_l < 1, \quad 0 < u_k < 1,$$
$$0 < u_r < 1, \quad u_h + u_l + u_r + u_k = 1 \tag{27}$$

One method is to search the optimal values of u_h, u_l, u_k and u_r for fixed U_T. One can always obtain $u_r = 0$ for the dimensionless profit rate maximization. The reason is that the regeneration makes the dimensionless profit rate decrease.

Another method is adopted herein. The optimization is performed by searching the optimal heat conductance allocations among hot-, cold- and consumer-side heat exchangers for fixed total heat exchange inventory and fixed heat conductance allocation of the regenerator, and by searching the optimal pressure ratio of the compressor. That is, to search the optimal values of u_h, u_l, u_k and π for fixed U_T and u_r. The numerical calculations are performed by using the optimization toolbox of Matlab. In the calculations, $U_T = 10kW/K$, $u_r = 0.1$, $k = 1.4$, $C_{wf} = 1.0kW/K$, $\tau_1 = 5$, $\tau_2 = 1.4$

and $\tau_3 = 1$ are set. According to analysis in Ref. 52, $a = 10$ and $b = 6$ are set. The three-dimensional characteristic of $\overline{\Pi}$ versus $(u_h,\ u_l)$ with pressure ratio $\pi = 8$ is shown in Fig.2. It can be seen from Fig.2 that there exists a sole group of heat conductance allocations among hot-, cold- and consumer-side heat exchangers which leads to an optimal dimensionless profit rate $\overline{\Pi}_{opt}$. The characteristic of $\overline{\Pi}_{opt} - \pi$ is shown in Fig.3. It can be seen from Fig.3 that there exists an optimal pressure ratio $\pi_{\overline{\Pi}}$ which leads to the maximum dimensionless profit rate $\overline{\Pi}_{\max}$. That is, there exist a sole group of optimal heat conductance allocations ($(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$) among hot-, cold- and consumer-side heat exchangers and an optimal pressure ratio $\pi_{\overline{\Pi}}$ which lead to the maximum dimensionless profit rate $\overline{\Pi}_{\max}$.

The characteristics of, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$ and $\pi_{\overline{\Pi}}$ versus τ_1, τ_2, a, b and U_T are shown in Figs.4-8, respectively. It can be seen that the hot-side optimal heat conductance allocation $(u_h)_{\overline{\Pi}}$ does not change obviously with the changes of τ_1, τ_2, a, b and U_T, the cold-side optimal heat conductance allocation

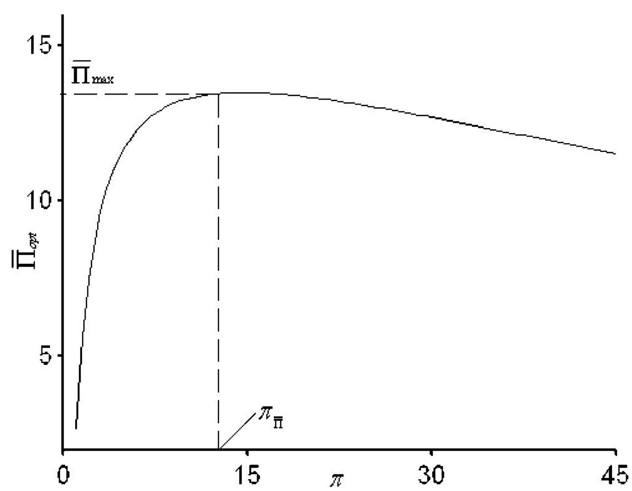

FIGURE 3. The characteristic of $\overline{\Pi}_{opt}$ versus π.

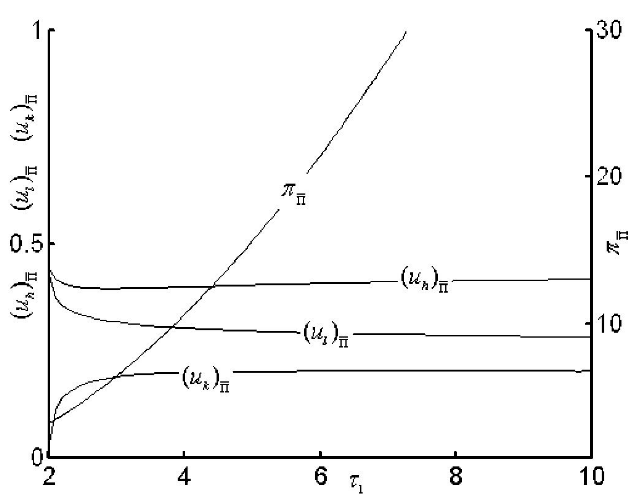

FIGURE 4. The characteristics of $(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$ and $\pi_{\overline{\Pi}}$ versus τ_1.

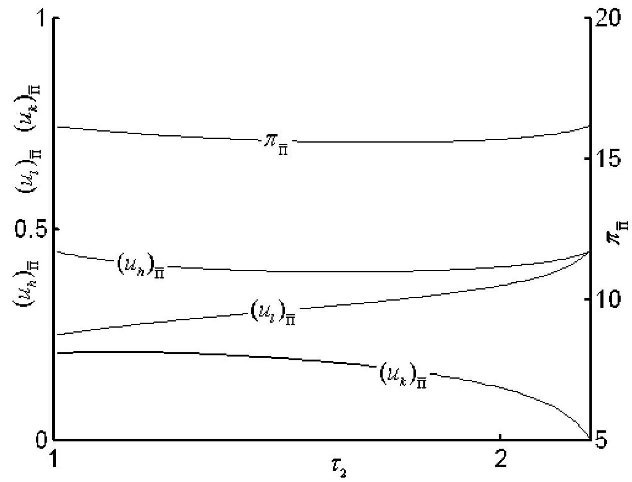

FIGURE 5. The characteristics of $(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$ and $\pi_{\overline{\Pi}}$ versus τ_2.

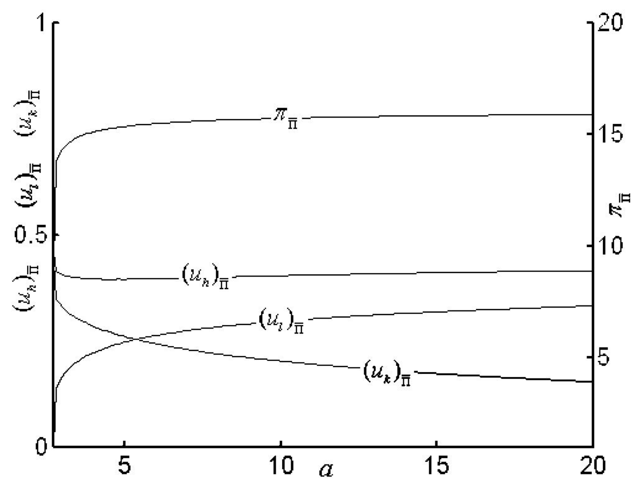

FIGURE 6. The characteristics of $(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$ and $\pi_{\overline{\Pi}}$ versus a.

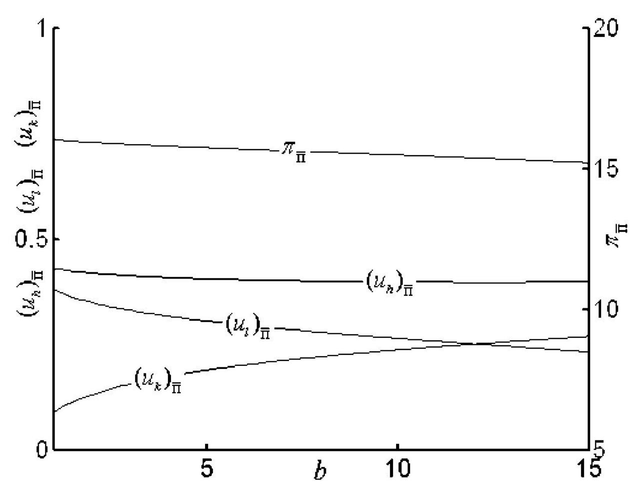

FIGURE 7. The characteristics of $(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$ and $\pi_{\overline{\Pi}}$ versus b.

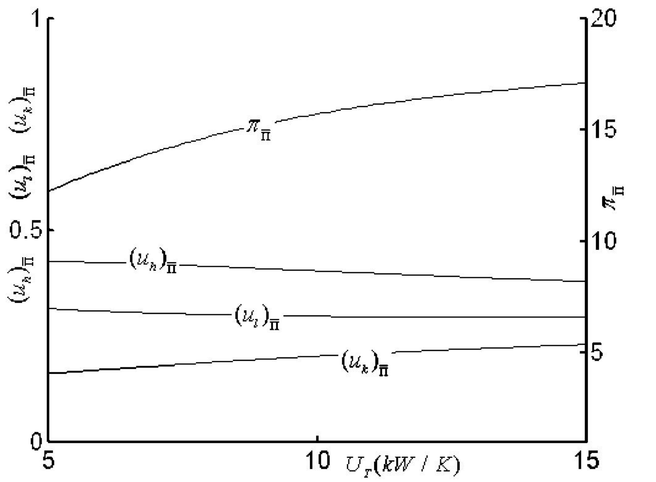

FIGURE 8. The characteristics of $(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$ and $\pi_{\overline{\Pi}}$ versus U_T.

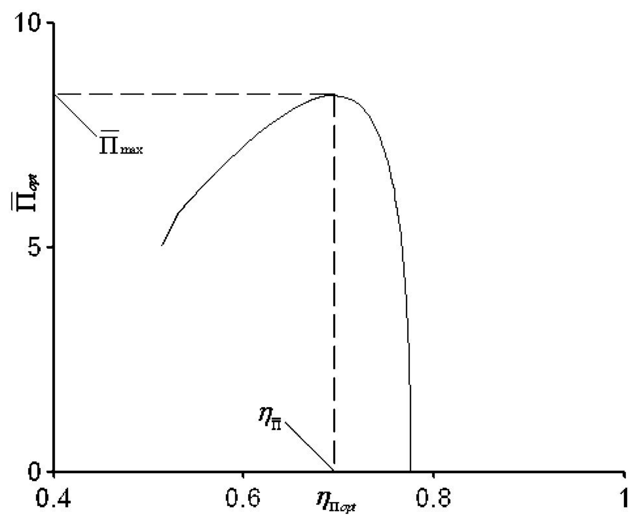

FIGURE 9. The characteristics of $\overline{\Pi}_{opt}$ versus $\eta_{\Pi opt}$.

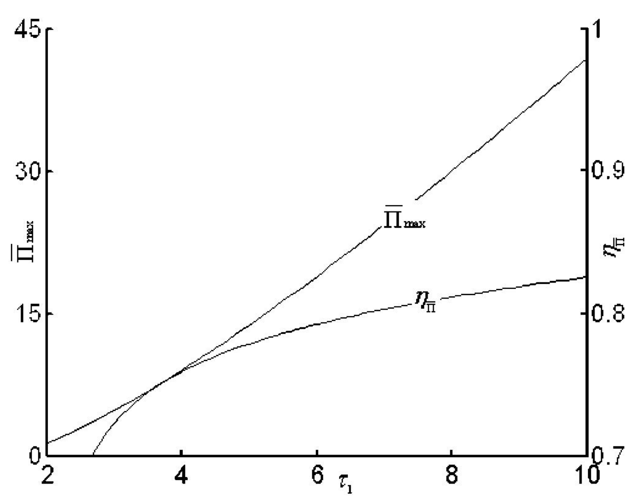

FIGURE 10. The characteristics of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus τ_1.

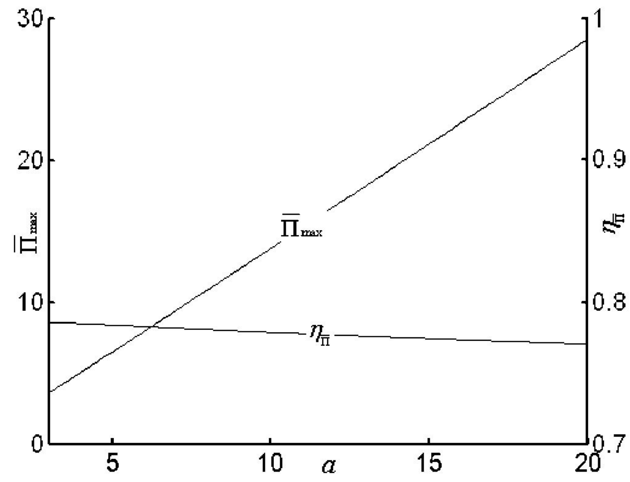

FIGURE 11. The characteristics of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus a.

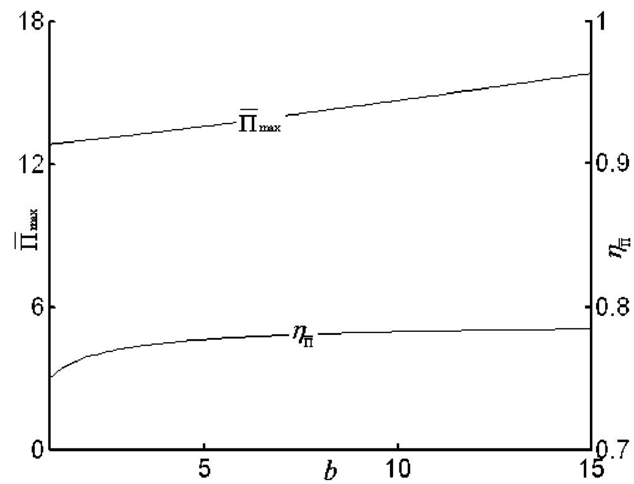

FIGURE 12. The characteristics of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus b.

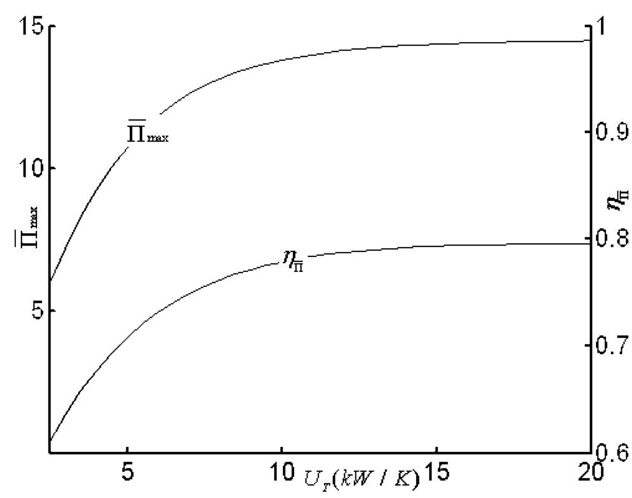

FIGURE 13. The characteristics of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus U_T.

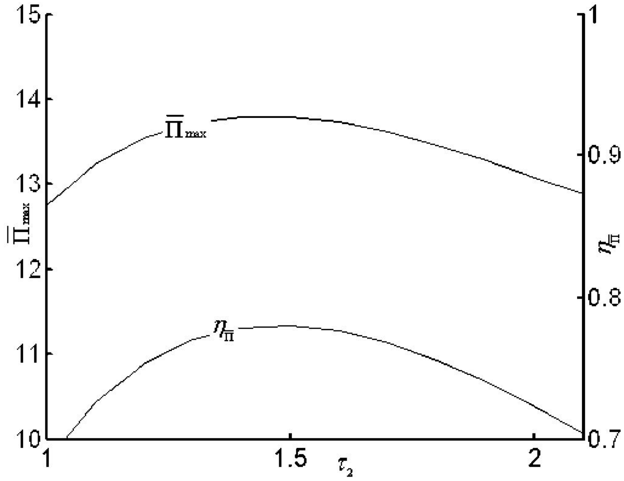

FIGURE 14. The characteristics of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus τ_2.

$(u_l)_{\overline{\Pi}}$ increases and the consumer-side optimal heat conductance allocation $(u_k)_{\overline{\Pi}}$ decreases with the increases of τ_2 and a, and the decreases of b and U_T. The optimal pressure ratio $\pi_{\overline{\Pi}}$ does not change obviously with the changes of τ_2, a and b, and increases with the increases of τ_1 and U_T.

The characteristic of optimal dimensionless profit rate $\overline{\Pi}_{opt}$ versus corresponding exergy efficiency $\eta_{\Pi opt}$ is shown in Fig.9. It can be seen that the $\overline{\Pi}_{opt} - \eta_{\Pi opt}$ characteristic is a parabolic-like curve. There exists a maximum profit rate $\overline{\Pi}_{\max}$ and the corresponding exergy efficiency $\eta_{\overline{\Pi}}$, *i.e.* the finite time exergoeconomic performance bound.

The characteristics of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus τ_1, a, b, U_T and τ_2 are shown in Figs.10-14. Figs.10-13 illustrate that the maximum profit rate $\overline{\Pi}_{\max}$ increases with the increases of τ_1, a, b and U_T. $\overline{\Pi}_{\max}$increases asymptotically with the increase of U_T on. The finite time exergoeconomic performance bound $\eta_{\overline{\Pi}}$ does not change obviously with the changes of a and b, and increases asymptotically with the increases of τ_1 and U_T. It can be seen from Fig.14 that the characteristic of $\overline{\Pi}_{\max}$ and $\eta_{\overline{\Pi}}$ versus τ_2 are parabolic-like curves, *i.e.* there exists an optimal consumer-side temperature. The conclusion can also be obtained by taking the total exergy output rate given in Eq. (25) as the optimization objective. However, the consumer-side temperature should be kept as low as possible according to analysis in the previous works [15-20].

5. Conclusion

Finite time exergoeconomic analysis and optimization is applied to optimize the profit rate of an endoreversible regenerative gas turbine closed-cycle cogeneration plant coupled to constant temperature heat reservoirs. It is shown that there exist a sole group of optimal heat conductance allocations among hot-, cold- and consumer-side heat exchangers and an optimal pressure ratio which lead to the maximum profit rate for fixed total heat exchange inventory and fixed heat conductance allocation of the regenerator. Moreover, the hot-side optimal heat conductance allocation almost remains a constant, the cold-side and consumer-side optimal heat conductance allocations vary with the consumer-side temperature, the total heat exchanger inventory and the price ratios. The characteristic of optimal dimensionless profit rate versus corresponding exergy efficiency is obtained and the effects of various parameters on the maximum profit rate and the finite time exergoeconomic performance bound are also analyzed. It is found that there exists an optimal consumer-side temperature owing to the modified calculation of thermal exergy output rate.

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A Nomenclature

a	ratio of power output to exergy input rate price
b	ratio of thermal exergy output rate to exergy input rate price
C	heat capacity rate (kW/K)
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)
u_h	hot-side heat conductance allocation
u_k	consumer-side heat conductance allocation
u_l	cold-side heat conductance allocation
u_r	heat conductance allocation of the regenerator
x	isentropic temperature ratio of the working fluid

B Greek symbols

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

Subscripts

H	hot-side
K	consumer-side
L	cold-side
R	regenerator
$\max$	maximum
opt	optimal
P	power output
T	total
wf	working fluid
$\overline{\overline{\Pi}}$	maximum dimensionless profit rate
Πopt	optimal profit rate
0	ambient
$1,2,3,4,5,6,7$	state points of the cycle
	dimensionless

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