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Parametirc Analysis And Optimum Performance Of Irreversible Closed Brayton Cycle Combined Cooling, Heating and Power

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Abstract: A combined cooling, heating and power (CCHP) plant model composed of an irreversible closed Brayton cycle and an endoreversible four-heat-reservoir absorption refrigeration cycle is established. The irreversibilities considered in the CCHP plant include heat-resistance losses in the hot-, cold-, thermal consumer-,generator-, absorber-, condenser- and evaporator-side heat exchangers as well as nonisentropic losses in the compression and expansion processes. At first, Equations of exergy efficiency and profit rate of the CCHP plant are derived.The effects of some design parameters, including compressor and gas turbine efficiencies, ratio of heat demanded by the thermal consumer to power output,optimal compressor pressure ratio, maximum profit rate and finite time exergoeconomic performance bound of the CCHP plant are discussed by numerical examples. The results obtained may provide some theoretical guidelines for the designs and operations of the practical CCHP plants.

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Keywords: CCHP**,** Profit rate **,** Endoreversible cycle, Closed Brayton cycle, Four-heat-reservior absorbtion refrigeration

1. Introduction

Nowadays, energy and environmental problems have been widely paid attention. To improve energy utilization efficiency and reduce emissions of harmful pollutants are the main measures for the solution of energy and environmental problems.Much work concentrates on the improvements of thermodynamic system efficiencies, but waste heats of thermodynamic systems aren't made full use of. Compared with thermodynamic systems with only work or heat output, the combined heating and power (CHP) plants make full use of waste heats and combine the work and heat output organically, which makeenergy utilization efficiencies of CHP plants improved greatly. Compared with simple thermodynamics systems and CHP plants, the combined cooling, heating and power (CCHP) plant is an integrated system that makes best use of different forms of energy (including cooling, heating and power) on the basis of energy cascade utilization principle [1]. CCHP plant has anexcellent cycle performance, a higher energy integrated utilization efficiency and a lower emissions of harmful gases. It has a broad prospect of development, and people begin to pay increasingly attention on it.

Finite time thermodynamics (FTT) [2–10] have been applied to analyze and optimize the performances of various thermodynamic systems. For this theme, performance analyses and optimizations on CHP plants have made great progress

The endoreversible [11–13] and irreversible [14–16] Carnot cycle cogeneration plants have been analyzed and optimized by taking annual worth [11], exergy output rate [12–15], exergy efficiency [15], exergy density [16] as optimization objectives.However, the Carnot cycle cogeneration plant is hard to achieve in a real cogeneration plant. Based on Carnot cyclecogeneration plant, the performances of various Brayton cycle cogeneration plant have been analyzed and optimized [17–23]. Yilmaz [17] and Hao and Zhang [18,19] optimized the performances of the endoreversible Brayton cycle cogenerationplants by taking exergy output rate [17,18] and useful energy rate [18,19] as objectives. Ust et al. [20,21] investigated the optimal exergetic performance coefficient of an irreversible regenerative gas turbine cycle cogeneration plant [20] and an irreversible Dual cycle cogeneration plant [21] based on ecological performance analysis. Tao et al. [22,23] analyzed the optimal exergoeconomic performances of the endoreversible simple and regenerative closed Brayton cycle cogeneration plant by taking profit rate as optimization objective, and found that there exited an optimal heat consumer-side temperature.Studies on absorption refrigeration cycle by using the theory of finite time thermodynamics have been made great progresses [24–40]. The three-heat-reservoir absorption refrigeration cycle models [24–31] have been established by some authors. On the basis of the three-heat-reservoir absorption refrigeration cycle model, the four-heat-reservoir absorption refrigeration cycle models were further established. Chen [32] and Shi and Chen [33] investigated the optimal specific cooling load and coefficient of performance (COP) relations of a four-heat-reservoir absorption refrigeration cycle with heat resistance and internal irreversibility. Zheng et al. [34], Chen et al. [35,36] and Zheng et al. [37] investigated the optimal cooling load and COP relations of the endoreversible and irreversible four-heat-reservoir absorption refrigeration cycles with Newton's and linear phenomenological heat transfer laws. Qin et al. [38,39] analyzed and optimized the thermoeconomic performances of constant- and variable-temperature four-heat-reservoir absorption refrigeration cycles.Tao et al. [40] investigated the optimal ecological function of an endoreversible four-heat-reservoir absorption refrigeration cycle.

CCHP plants, based on temperature matched and energy cascade utilization principle, have been paid attention by many authors. Performance analyses and optimizations of CCHP plants have been carried out by using classical thermodynamic theory. Kong et al. [41] performed the energy efficiency evaluation and economic feasibility analysis of a small scale trigeneration system with an available Stirling engine. It showed that this system saved fuel resources and had the assurance of economic benefits compared with conventional independent CCHP plant. Huangfu et al. [42] performed the economic evaluation and analysis of the micro-scale CCHP system, and gave an ideal pay back period with the consideration of a good economic efficiency of the micro-scale CCHP system. Mago and Chamra [43] provided the optimal operation strategy of CCHP plant with considerations of primary energy consumption, operating costs and carbon dioxide emissions, and made a comparison between general performance and optimal performance of the CCHP plant. Khaliq and Kumar [44] discussed the effects of overall pressure ratio, turbine inlet temperature, pressure drop in combustor and evaporator temperature on the fuel utilization efficiency, electrical to thermal energy ratio and second law efficiency of the gas turbine cycle, cogeneration and CCHP plant based on the first law as well as the second law of thermodynamics. Kavvadias and Maroulis [45] developed a multi-objective optimization method on consideration of technical, economical, energetic and environmental performance indicators for the design of trigeneration plants, and optimized both construction and discrete operational variables based on realistic conditions.

Exergoeconomic (or thermoeconomic) analysis and optimization [46,47], as a relatively new method, 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's work [48] combined the endoreversible model [2–10] with exergoeconomic analysis. It was termed as finite time exergoeconomic analysis [22,23,49–60] 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. This ideal has been extended to quantum heat engine [49], generalized irreversible Carnot heat engines [50], refrigerators [51] and heat pumps [52], universal heat engine [53], refrigerator [54] and heat pumps [55,56], three-heat-reservoir refrigeration and heat pump cycles [57,58], endoreversible and irreversible four-heat-reservoir absorption heat pumps and heat transformers[59], generalized irreversible combined Carnot refrigeration cycle [60] as well as endoreversible closed-cycle simple and regenerative gas turbine cogeneration plants [22,23].

Zhang and Yang [61] optimized the performance of a combined heating and power plant as well as a combined cooling and power plant by taking exergy output rate as objective, and obtained characteristics of optimal matching between heating (cooling) and power generation. It is a good attempt to introduce the finite time thermodynamic theory into the analysis and optimization of CCHP plant. However, this work cannot reflect the overall performance of the CCHP plant with cooling, heating and power supplied simultaneously, and it cannot provide overall theoretical guidelines for the designs and operations of CCHP plants. In this paper, a CCHP plant model composed of an irreversible closed Brayton cycle and an endoreversible four-heat-reservoir absorption refrigeration cycle will be established by using finite time thermodynamics. This CCHP plant model will consider the available cooling, heating and power simultaneously. Based on the finite time exergoeconomic analysis method, profit rate optimization of the CCHP plant will be carried out by searching the compressor optimal pressure ratio and the optimal hot-, cold, thermal consumer-, generator-, absorber-, condenser- and evaporator-side heat exchangers for a fixed total heat exchanger inventory. The effects of design parameters on the optimal performances of the CCHP plant will be discussed by numerical examples.

2. Cycle model of CCHP plant

An overall schematic diagram and T-s diagram of heating and power subsystems of CCHP plant are shown in Figs. 1 and 2,respectively. The cycle model operates between temperatures of four-heat reservoirs TH, TL, T0, and Th. In the closed Brayton cycle, processes 1→2 and 3→4 are the irreversible adiabatic compression and expansion processes in the compressor and gas turbine, respectively (processes 1→2s and 3→4s are the isentropic adiabatic compression and expansion processes corresponding to processes 1→2 and 3→4); QH is the heat transfer rate from the heat source at temperature TH to the working fluid in process 2→3; QKc is the heat transfer rate from the working fluid to the generator of the four-heat-reservoir absorption refrigerator at temperature T0 in process 4→5; QKh is the heat transfer rate from the working fluid to the thermal consumer at temperature Th in process 5→6; QL is the heat transfer rate from the working fluid to the heat sink at temperature TL in process 6→1; P is the power output of CCHP plant.In the closed Brayton cycle, when the heat transfer obeys Newton's law, according to the properties of the working fluid and the theory of heat exchangers, one can obtain:

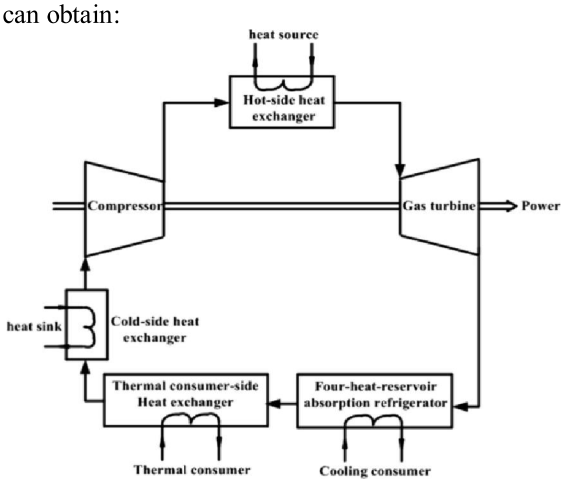

Fig. 1. Overall flow diagram of CCHP plant.

$Q_H=C_{wf}(T_3-T_2)= C_{wf}\, E_H(T_H-T_2)$,

$Q_L=C_{wf}(T_6-T_1)= C_{wf}\, E_L(T_6-T_L)$,

$Q_g=C_{wf}(T_4-T_5)= C_{wf}\, E_g(T_4-T_g)$,

$Q_h=C_{wf}(T_5-T_6)= C_{wf}\, E_h(T_5-T_h)$,

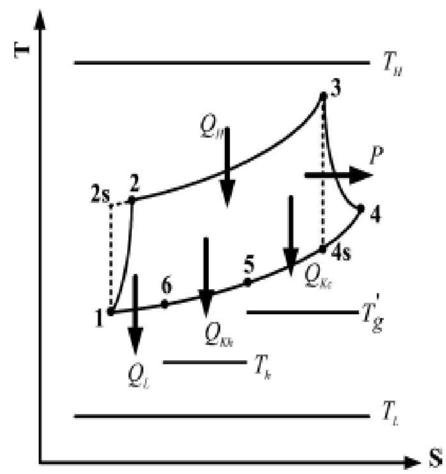

Fig. 2. T–S diagram of heating and power subsystems of the irreversible closed Brayton cycle CCHP plant

In the closed Brayton cycle 1–2s–3–4s–1, the endoreversible condition requires:
$T_{2s}/T_1=x$ and $\eta_c=(T_{2s}-T_1)/(T_2-T_1)$.
An endoreversible four-heat-reservoir absorption refrigeration cycle model, which consists of a generator, an absorber, a condenser and an evaporator, is shown in Fig. 3. Assuming that the flow of the working fluid in the absorption refrigeration cycle is stable, Qa is the heat transfer rate from the working fluid of absorber at temperature T_a' to the heat sink at temperatureTa, Qc is the heat transfer rate from the working fluid of condenser at temperature T_c' to the heat sink at temperature Tc, and Qe is the heat transfer rate from the heat source at temperature Te to the working fluid of evaporator at temperature T_e'.
In the endoreversible four-heat-reservoir absorption refrigeration cycle, according to the Newton's heat transfer law, heat transfer rates (Qe) are, respectively, given by:
$R=Q_e= U_e(T_a-T_a')$,

where R is the cooling load of the absorption refrigeration cycle, and Ue, Uc and Ua are the heat conductances of the evaporator-,condenser- and absorber-side heat exchangers, respectively. Power input required by the solution pump and heat loss rate caused by flowing in the absorption refrigeration system is negligible relative to the energy input to the generator and is often neglected for the purpose of analysis [32–40]. Therefore, the distribution of the total heat rejection quality between the absorber and condenser (n) and COP can be, respectively, defined as:

$n = Q_a / Q_c$ $\varepsilon = Q_e / Q_{Kc}$,

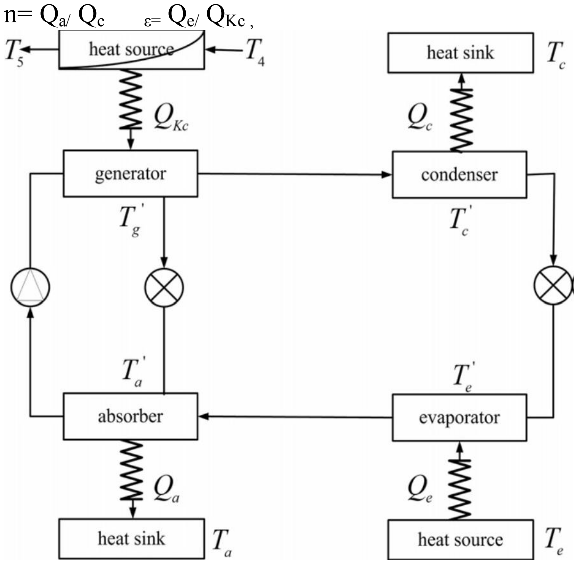

Fig. 3. Endoreversible four-heat-reservoir absorption refrigeration cycle model.

Applying the first and second law of the thermodynamics For the given heat exchanger heat conductances the general relation between the cooling load (R) of four-heat-reservoir absorption refrigeration cycle and the heat transfer rate of the generator(Q_{Kc}) is given by [32]:

$$U_c(Q_{Kc}+R)/[Q_{Kc}+R+U_cT_c(n+1)]+nU_a(Q_{Kc}+R),$$
$$=U_eR/(U_eT_e-R)+C_{wf}Q_{Kc}E_g/(C_{wf}E_gT_4-Q_{Kc}).$$

3.Profit rate and exergy efficiency of CCHP plant

The power output of CCHP plant is:

$P = Q_H - Q_L - Q_{Kh} - Q_{Kc}$.

The ratio of heat demanded by thermal consumer to power output of the CCHP plant is defined as [12]:

$\omega_h = Q_{Kh}/P$.

The exergy output rate of the power output is:

$EX_W = P$.

According to the method of calculating thermal exergy output rate in Ref. [22], the thermal exergy output rate (EX_{Kh}) released to the thermal consumer is given by:

$EX_{kh} = Q_{kh}(1 - T_0/T_h)$,

where T0 is environment temperature.

The cooling exergy output rate of endoreversible four-heat-reservoir absorption refrigeration cycle is:

$EX_{Kc} = R(T_0/T_e - 1)$,

where the cooling load of endoreversible four-heat-reservoir absorption refrigeration cycle is determined. The total exergy input rate (EXI) and exergy output rate (EX) of CCHP plant are, respectively, given by:

$EX_I = Q_H(1 - T_0/T_h) - Q_L(1 - T_0/T_L)$,

$EX = EX_W + EX_{Kh} + EX_{Kc}$

The exergy efficiency of CCHP plant is:

$\eta = EX/ EX_I$

Assuming that prices of power output, cooling exergy output rate of four-heat-reservoir absorption refrigeration cycle, thermalexergy output rate and exergy input rate be φ_p , φ_c , φ_I and φ_I , respectively, the profit rate of CCHP plant is defined as:

$\Pi = \varphi_p EX_W + \varphi_I EX_{Kh} + \varphi_c EX_{Kc} - \varphi_I EX_I$.

Submitting The previous equation yields the exergy efficiency and dimensionless profit rate $\overline{\Pi} = \Pi /(\varphi I C_{wf} T_0)$ respectively,

$$\eta = \frac{b_5C_{wf}^{-1}T_0^{-1}(\tau_c^{-1}-1)R + E_h(1+\omega_h-\tau_h^{-1}\omega_h)(x-1)[\eta_c\eta_t E_H\tau_H + b_4(\tau_h + E_L\tau_L - E_L\tau_h)]}{E_L(\tau_L^{-1}-1)[xE_h\eta_c(\tau_h-\tau_L)+\omega_h(x-1)(1-E_h)(\eta_c\eta_t E_H\tau_H + b_4\tau_L)] + E_H(1-\tau_H^{-1})}$$
$$\times [x\eta_c E_h\tau_H + b_2xE_h(E_L\tau_h - E_L\tau_L - \tau_h) - \omega_h\tau_H(x-1)(1-E_h)(1-E_L)(b_2\eta_t - x)],$$

$$\Pi = b_5^{-1}E_h(x-1)[\psi_p + \psi_h\omega_h(1-\tau_h^{-1})][b_4(\tau_h + E_L\tau_L - E_L\tau_h) + \eta_c\eta_t E_H\tau_H] + b_5^{-1}E_L(1-\tau_L^{-1})[x\eta_c E_h(\tau_h - \tau_L)$$
$$+\omega_h(x-1)(1-E_h)(\eta_c\eta_t E_H\tau_H + b_4\tau_L)] + b_5^{-1}E_H(\tau_H^{-1}-1)[x\eta_c E_h\tau_H - \omega_h\tau_H(x-1)(1-E_h)(1-E_L)(b_2\eta_t - x)]$$
$$+ b_2xE_h(E_L\tau_h - E_L\tau_L - \tau_h)] + \psi_c C_{wf}^{-1}T_0^{-1}(\tau_c^{-1}-1)R,$$

4. Performance optimization of CCHP plant

the dimensionless profit rate of CCHP plant are functions of heat conductances of hot-, cold-, thermal consumer-, generator-, absorber-, condenser- and evaporator-side heat exchangers (UH, UL, Uh, Ug, Ua, Uc and Ue) as well as the temperature ratio (x, i.e., the compressor pressure ratio p) for the fixed price ratios (ψ_p, ψ_h andψ_c), heat reservoir temperature ratios as well as efficiencies of compressor and gas turbine (gc and gt). For the fixedtotal heat exchanger inventory UT (UH + UL + Uh + Ug + Ua + Uc + Ue = UT), heat conductance distributions of hot-, cold-, thermal consumer-, generator-, absorber-, condenser- and evaporator-side heat exchangers can be, respectively, defined as follows:

uH=UH/UT, uL=UL/UT,uh=Uh/UT,ug=Ug/UT,
ua=Ua/UT, uc=Uc/UT, ue=Ue/UT.

Obviously, heat conductance distributions of the seven heat exchangers should satisfy the following constraint:

0 < uH < 1, 0 < uL< 1, 0 < uh < 1, 0 < ug < 1,
0 < ua < 1,0 < uc < 1,0 < ue < 1,
uH+ uL+ uh+ ug+ ua+ uc+ ue=1.

To make sure that the CCHP plant runs at a significant work condition, temperatures of CCHP plant should satisfy the following constraints:

TL < T1, T1 < T2, T2 < T3, T3 < TH,
T1 < T6, Th < T6,T6 < T5, T5 < T4, T4 < T3,
T1 + T3 + T2 + T4 > 0.

Because it is hard to get the analytical formulae about cooling load of the four-heat-reservoir absorption refrigeration cycle, the optimal heat conductance distributions and compressor optimal pressure ratio can be obtained by using Powell arithmetic when carrying out profit rate optimization.

5. Results

To analyze the optimal performance and the effects of design parameters on the optimal performance of the CCHP plant, itis set that UT = 20 kW/K, Cwf = 1.0 kW/K, k = 1.4, ηc = ηt = 0.85, τH = 5, τh = 1.2, τL = 1, τh = 1.0, n = 1, Ta = 303 K, Tc = 303 K,Te = 280 K, and T0 = 303 K. It is set that the price ratios of the power output to exergy input rate, cooling exergy output rateof four-heat-reservoir absorption refrigeration cycle to exergy input rate and thermal exergy output rate to exergy input rateare ψP = 10, ψc = 8 and ψh = 5, respectively, which are in reasonable ranges according to the Refs.[22,23,62,63]. Moreover, theeffects of the price ratios on the optimal performance of the CCHP plant will be discussed later. In the analyses, if there is nospecial explanation, the parameters mentioned above will keep constant.

Fig.4 shows the characteristics of the optimal dimensionless profit rate ($\overline{\Pi}$opt) and the corresponding seven optimal heat conductance distributions versus compressor pressure ratio (π). It indicates that there exists a sole group of optimal heat conductance distributions ((ui)opt(i = g, a, c, e, H, h, L)) corresponding to the optimal dimensionless profit rate ($\overline{\Pi}$opt) for a fixed compressor pressure ratio. Meanwhile, the curve of $\overline{\Pi}$opt versus π is a parabolic-like one, i.e., there exist a sole compressor optimal pressure ratio ($\pi_{\overline{\Pi}}$) and the corresponding seven optimal heat conductance distributions ,which lead to the maximum dimensionless profit rate ($\overline{\Pi}$max).

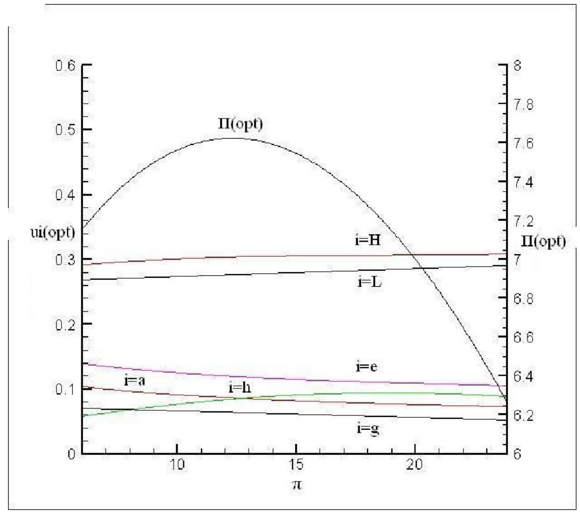

Fig.4. (ui)opt(i = g, a, c, e, H, h, L) and $\overline{\Pi}$opt versus π

Fig.5 shows the effects of the compressor and gas turbine efficiencies (ηc = ηt) on the characteristic of the optimal dimensionless profit rate (Πopt) versus the corresponding exergy efficiency (ηΠopt). The curve of Πopt versus ηΠopt is loopshaped one, i.e., there exist a maximum exergy efficiency (ηmax) with the corresponding dimensionless profit rate (Πη) as well as a maximum dimensionless profit rate (Πmax) with the corresponding exergy efficiency (ηΠ), which is the finite time exergoeconomic performance bound. Moreover, Πmax and ηmax increase with the increases of ηc and ηt; when ηc = 1 and ηt = 1, Πmax and ηΠ are the maximum dimensionless profit rate and finite time exergoeconomic performance bound of the endoreversible closed Brayton cycle CCHP plant.

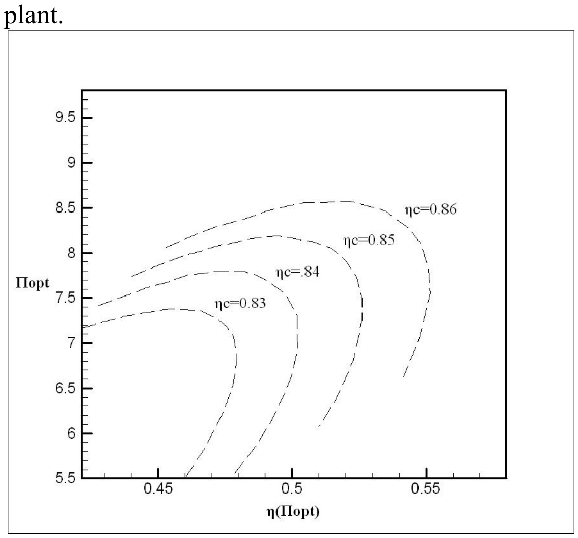

Fig. 5. Effects of ηc and ηt on the characteristic of Πopt versus $\eta_{\Pi\text{opt}}$.

The characteristics of (ui)opt(i = g, a, c, e, H, h, L),$\pi_{\overline{\Pi}}$, $\overline{\Pi}$max and versus ωh and τhare shown in Figs. 6-9 respectively.

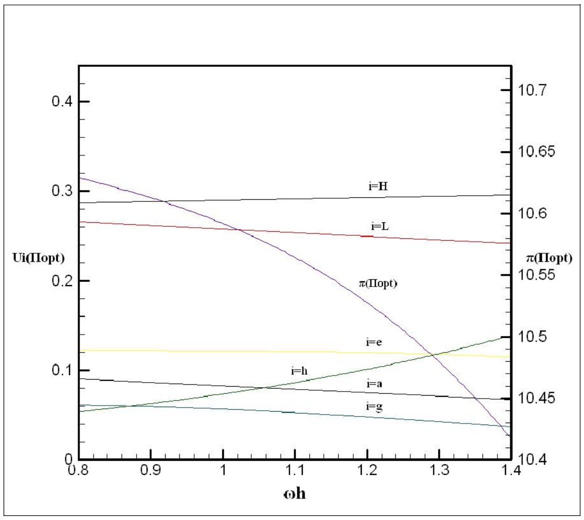

Fig. 6. (ui)opt(i = g, a, c, e, H, h, L) and $\overline{\pi}$opt versus ωh.

In general, the optimal hot- and cold-side heat conductance distributions are larger than the other optimal heat conductance distributions. Because the absorber and condenser of four-heat-reservoir absorption refrigerator have the same heat reservoir temperatures and n = 1, the optimal heat conductance distributions $(ua)_\pi$ and $(uc)_\pi$ identically equal to each other. The optimal heat conductance distributions have the same change tendency with the change of each parameter. Moreover, the effects of τh on $(ua)_\pi$, ψ_c on Πmax, UT on $(uh)_\pi$ well as ψP, ψh and ψc on (ui)opt(i = g, a, c, e, H, h, L) and η_Π are not obvious. These are the main conclusion obtained by analyzing the effects of design parameters on the optimal performance of the CCHP plant.

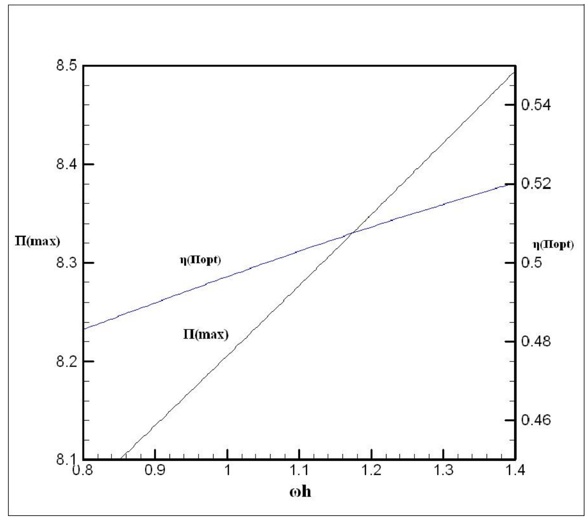

Fig. 7 Characteristics of Πmax and $\eta_{\Pi\text{opt}}$ versus ωh.

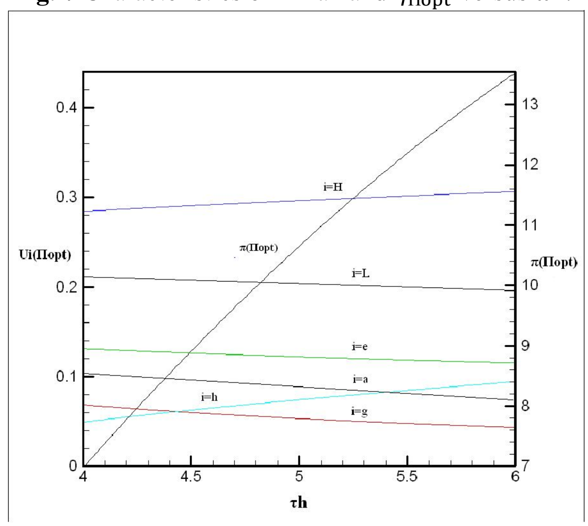

Fig.8 (ui)opt(i = g, a, c, e, H, h, L) and $\overline{\pi}$opt versus τh.

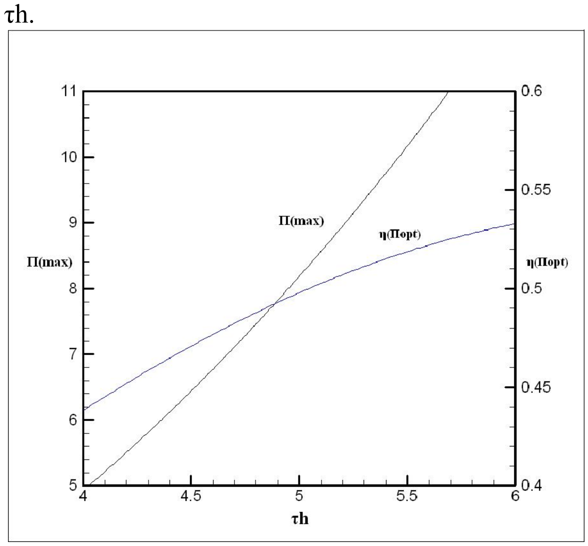

Fig. 9 Characteristics of Πmax and $\eta_{\Pi\text{opt}}$ versus τh

6. Discussions

The combined cooling, heating and power (CCHP) plant model composed of an irreversible closed Brayton cycle and an endoreversible four-heat-reservoir absorption refrigeration cycle is established by using the theory of finite time thermodynamics.Expressions of the profit rate and exergy efficiency of the CCHP plant are derived. The main purpose is to get the optimal heat conductance distributions of the hot-, cold-, thermal consumer-, generator-, absorber-, condenser- and evaporator-side heat exchangers, compressor optimal pressure ratio, maximum profit rate and finite time exergoeconomic performance bound of the CCHP plant for a fixed total heat exchanger inventory based on the finite time exergoeconomicanalysis method. Because it is hard to get the analytical formulae about cooling load of the four-heat-reservoir absorption refrigeration cycle, the Powell arithmetic is introduced when carrying out profit rate optimization. It is proved that there exist a sole group of the seven optimal heat conductance distributions and compressor optimal pressure ratio, which lead to a maximum profit rate. The curve of optimal profit rate versus the corresponding exergy efficiency is loop-shaped one, and there exists a finite time exergoeconomic performance bound corresponding to the maximum dimensionless profit rate. The results obtained can provide some theoretical guidelines for the designs and operations of practical CCHP plants. In fact, practical Brayton cycle is not only simple closedone with constant-temperature heat reservoirs, but regenerated, intercooled, and intercooled and regenerated ones withvariable-temperature heat reservoirs as well as simple and regenerated open ones. Therefore, researches on CCHP plantswith different models and different optimization objectives by using the theory of finite time thermodynamics are extremely significant, which will greatly enrich the theory of finite time thermodynamics and can provide more theoretical guidelines for the designs and operations of various practical CCHP plants.

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