A global optimization method synthesizing heat transfer and thermodynamics for the power generation system with Brayton cycle

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Abstract. Supercritical carbon dioxide operated in a Brayton cycle offers a numerous of potential advantages for a power generation system, and a lot of thermodynamics analyses have been conducted to increase its efficiency. Because there are a lot of heat-absorbing and heat-lossing subprocesses in a practical thermodynamic cycle and they are implemented by heat exchangers, it will increase the gross efficiency of the whole power generation system to optimize the system combining thermodynamics and heat transfer theory. This paper analyzes the influence of the performance of heat exchangers on the actual efficiency of an ideal Brayton cycle with a simple configuration, and proposes a new method to optimize the power generation system, which aims at the minimum energy consumption. Although the method is operated only for the ideal working fluid in this paper, its merits compared to that only with thermodynamic analysis are fully shown.

1. Introduction
Brayton cycle has long been the typical thermodynamic cycle for power generation, and it has an extensive range of application. In order to utilize energy more efficiently, numerous researchers devote themselves to increasing the thermal efficiency of thermodynamic cycle and enhancing the heat transfer capability of the heat exchanger. As there have been numerous thermodynamics studies and heat transfer researches for Brayton cycle respectively, combining these kinds of studies to optimize the power generation system will be an important research direction to reduce the total energy consumption further.

In recent years, most studies about Brayton power cycle use supercritical carbon dioxide (s-CO₂) as its working fluid, as it has many advantages [1,2] such as a potential high efficiency, a high power density and a small size of the power generation system. As the basic research method [3,4] of Brayton power cycle is mature, many researchers analyze the Brayton cycle thermodynamically based on the special thermophysical property of s-CO₂ to pursue breakthroughs in increasing cycle efficiency. The thermal efficiencies of four s-CO₂ Brayton cycle configurations for concentrating solar power system were calculated [5] using engineering equation solver [6] which contains the real-gas CO₂ properties in its program, then the author compared the efficiencies of different cycle configurations and different turbine inlet temperatures. Furthermore, many researchers made a second law analysis of s-CO₂ Brayton cycle [7-9]. They got the exergetic efficiency of the power cycle with
the variation of the component parameters, and then analyzed the irreversibility in the power system, and finally proposed the optimum scheme for power systems with different configurations. As the practical application, Fahad A. Al-Sulaiman et al. [10] compared the performance of different s-CO$_2$ Brayton cycles integrated with the actual condition solar power, and Brian D. Iverson et al. [11] analyzed the influence of the transient nature of the solar resource on the performance parameters of the s-CO$_2$ Brayton power cycle. Besides, for the power generation system utilizing different heat source, many s-CO$_2$ Brayton power cycle configurations were studied for nuclear applications [12-14] and in these studies researchers also made a series of thermodynamic analysis to optimize the cycle efficiency.

On the other hand, for the heat exchangers in the power generation system, the research about its heat transfer capability also achieves certain development. Although the research on heat exchanger and its design criterion have been comparatively mature [15,16], much of the technology appears to enhance the heat-transfer capability and make the heat exchanger adapt to more harsh conditions. In the High Temperature Reactor-E European program, the research group specifically studied the “recuperator” in a power generation system using the Brayton cycle [17]. They investigated most promising technologies for the compact recuperator and selected two of them to make the heat exchanger working under the harsh condition.

In most researches for Brayton cycle, the thermodynamics analysis and the heat transfer analysis work independently. People usually design the process of Brayton cycle first and optimize it to obtain a high cycle efficiency, and then set the heat exchangers which have been optimized by heat transfer analysis into the cycle. Since the overall efficiency of the power generation system is determined by the efficiency of thermal cycle and the heat transfer capability of heat exchangers, combining the thermodynamics and heat transfer analysis to design the whole system is more comprehensive than optimizing it in two stages, and the composite analysis will reduce the energy consumption further. However, the study of this integrated optimization is few because of the complexity of establishing the relationship between the parameters of heat transfer components and the thermal cycle efficiency. In recent years, a new concept for heat exchanger which is named entransy-dissipation-based thermal resistance (EDTR) [18] appeared, and its special mathematical form make it easy to integrate the heat transfer equation into the thermodynamics analysis. This concept has been applied on some refrigeration systems [19,20] to optimize it, and this gives an inspiration to optimize the power generation system with Brayton cycle comprehensively.

This contribution intends to propose a global optimization method of the power generation system with Brayton cycle, and the method blends the efficiency calculation with the heat transfer performance of the heat exchangers. We transform the logarithmic mean temperature difference (LMTD) to a simple form which makes it easy to combine the thermodynamic analysis and the heat transfer analysis, and then we optimize the parameters of the components in the system with the objective of minimizing the consumption of the heat transfer fluid. Although the model stated in this paper is operated with ideal working fluid, we definitely elaborate the significance of the synthesis of thermodynamic analysis and heat transfer analysis for a power generation system optimization.

2. Analysis for a closed Brayton power cycle
The configurations of a simple closed Brayton power cycle is shown in figure 1. It consists of a compressor, a turbine and two heat exchangers. The working fluid first flows into the compressor and the compressor puts work $W_1$ into it, and then the working fluid with a higher pressure flows into the heater and absorbs heat $Q_1$ from the heat transfer fluid. After being heated, the working fluid with high temperature and pressure does the work $W_2$ in the turbine and finally releases heat $Q_2$ to the cooling water. In figure 1, $T_{h,i}$ and $T_{h,o}$ separately represent the temperatures of the heat transfer fluid at inlet and outlet of the heater, and $T_{c,i}$ and $T_{c,o}$ separately represent the temperatures of the cooling water at inlet and outlet of the cooler. In order to simplify the analysis and calculation, some assumptions are made in the following analysis:
(1) The fluids in the system is considered as ideal fluid and its constant pressure specific heat capacity is constant
(2) The pressure drop in the heat exchangers is ignored, and the thermodynamic process in the compressor and the turbine are isentropic
(3) The heater and cooler are counter-flow heat exchangers
(4) The heat transfer coefficient of the heat exchangers is constant

![Figure 1](image1.png)

**Figure 1.** The sketch of a Brayton power cycle system.

2.1. *Thermodynamic analysis for the system*
Because of the simplification for the model, the thermal process of this system becomes very simple, and the T-s diagram for Brayton cycle is shown in figure 2. Brayton cycle consists of four physical processes: a) isentropic compression (1→2) in the compressor; b) isobaric heat supply (2→3) in the heater; c) isentropic expansion (3→4) in the turbine; d) isobaric heat removal (4→1) in the cooler. In figure 2, $T$ and $p$ separately represent the temperature and pressure of the working fluid, and the

![Figure 2](image2.png)

**Figure 2.** $T$-$s$ diagram for a Brayton power cycle.
subscripts 1,2,3,4 separately denote the four positions in figure 1.

Based on the assumptions as previously mentioned, the temperature and the pressure of the working fluid have the relation below [3,4]:

\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \pi^{\frac{\gamma-1}{\gamma}}, \tag{1a} \]

\[ \frac{T_3}{T_4} = \left( \frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \pi^{\frac{\gamma-1}{\gamma}}, \tag{1b} \]

where \( \gamma \) is the specific heat ratio of working fluid, and the pressure variations in two heat exchangers are ignored, i.e. \( p_2=p_3, p_1=p_4 \), \( \pi \) is the pressure ratio of the compressor. As the specific heat of the working fluid is assumed to be a constant, the cycle efficiency is written as [3,4]

\[ \eta = 1 - \frac{1}{\pi^{\frac{\gamma-1}{\gamma}}}, \tag{2} \]

and \( Q_1 \) and \( Q_2 \) can be expressed as

\[ Q_1 = \frac{W_{net}}{\eta}, \tag{3} \]

\[ Q_2 = \frac{1-\eta}{\eta} W_{net}, \tag{4} \]

where \( W_{net} \) is the network of the cycle and is equal to \( W_2 \) minus \( W_1 \).

2.2. Analysis for the heat exchangers

As the important components in this system, heat exchangers, i.e. the heater and the cooler, should be analyzed to establish the relation between its design parameters and the thermodynamic parameters of this system.

Figure 3. The sketch of a heat exchanger.

In general, for a counter-flow heat exchanger shown in figure 3, there are three constraint equations which involve the design parameters and the temperatures at each inlet and outlet, and the heat transfer characteristic equation of heat exchanger can be expressed with logarithmic mean temperature difference (LMTD)

\[ Q = kA \frac{\Delta T_{max} - \Delta T_{min}}{\ln \left( \frac{\Delta T_{max}}{\Delta T_{min}} \right)}, \tag{5} \]
where $Q$ is the heat transfer rate, $k$ is the overall heat transfer coefficient of the heat exchanger, $A$ is the heat transfer area of the heat exchanger, while $\Delta T_{\text{max}}$ and $\Delta T_{\text{min}}$ stand for the maximum and the minimum temperature differences between the hot and the cold fluid on the same side of the heat exchanger, respectively, and the energy conservation equations for the hot fluid and the cold fluid

$$Q = m_a c_{p,a} (T_{h,a} - T_{h,b}), \quad (6a)$$

$$Q = m_b c_{p,b} (T_{c,c} - T_{c,d}), \quad (6b)$$

where $T_{h,a}$, $T_{h,b}$, $T_{c,c}$ and $T_{c,d}$ separately represent the temperatures of the hot fluid at inlet and outlet and that of the cold fluid at inlet and outlet, $m_a$ and $m_b$ are the mass flow rates of hot and cold fluids, respectively, and $c_{p,a}$ and $c_{p,b}$ are the constant pressure specific heat capacities of hot and cold fluids, respectively.

In the power generation system as figure 1 shown, the working fluid is the cold fluid for the heater but the hot fluid for the cooler, and the temperatures of working fluid at different position in the cycle are the intermediate variables and are determined by the parameters of each component. To establish the relationship between the thermodynamic analysis and the heat transfer analysis, the intermediate variables should be eliminated so that the number of constraint equations and the complexity will decrease. However, for the form of equation (5) that the temperatures are taken the natural logarithm, it is difficult to combine heat transfer equations and thermodynamic equations such as equations (1a), (1b), and it is very convenience to combine these equations if the temperatures in the heat transfer characteristic equation of heat exchangers have a linear forms. From this point of view, as a mathematical deformation of equation (5), the heat transfer characteristic equation with linear temperature variables can be expressed as

$$Q = \frac{T_{h,a} + T_{h,b} - T_{c,c} + T_{c,d}}{\frac{2}{R_h}}, \quad (7)$$

where $R_h$ is defined as the thermal resistance of heat exchanger. Plugging equations (5), (6a), (6b) into equation (7), the thermal resistance for a counter-flow heat exchanger is decided as

$$R_h = \xi \frac{\exp(kA\xi) + 1}{2 \exp(kA\xi) - 1}, \quad (8)$$

and $\xi$ is written as $\frac{1}{m_a c_{p,a}} - \frac{1}{m_b c_{p,b}}$. Equation (5) is fully equivalent to equation (7), and the only difference is that the logarithmic term of temperatures in equation (5) is converted into the exponential term in $R_h$ as shown in equation (8), and each of the two equations can completely describe the performance of a heat exchanger, but equation (7) is far better than equation (5) to combine with the thermodynamics equations due to the arithmetic mean temperature difference.

2.3. Physical model of the pumps

Power generation system usually utilizes oil, salt, or steam as heat transfer fluids to transfer energy from the heat source to the power block, and the power block release heat to the cooling water. Both the heat transfer fluid and the cooling water are driven by pumps which consume large amounts of power. For a variable speed pump (VSP), the mass flow rate $m$, the head loss $H$, and the rotation frequency $\omega$ obey the following formula [21,22].

$$H = a_1 \omega^3 + a_2 \frac{m}{\rho} + a_3 \frac{m^2}{\rho^2}, \quad (9)$$
where $a_0$, $a_1$, $a_2$ are the characteristic parameters of VSP, and $\rho$ is the fluid density, and the power consumption of the pump is represented as

$$P = mgH,$$  \hspace{1cm} (10)

Although the model of this system is the most simplified and idealized, the new analysis method for this kind of problems can reflect its superiority compared to the traditional methods. For the more practical problem, the more complex model can be established and the equations which truly describe the physical processes can be plugged into the analysis, and this optimization method is also applied.

3. Optimization model for the closed Brayton power cycle

For a closed Brayton power cycle system shown in figure 1 with a prescribed system power output and the given inlet temperatures $T_{h,i}$ and $T_{c,i}$, the optimization goal is to decreased the flow rate of the heat transfer fluid which stands for the primary energy consumption of the system, and meanwhile the heat transfer area of the heater and the cooler must be restricted to make sure the dimension of the system not too large.

In this optimization problem, the constraint equations are all the physical equations mentioned in section 2, which are equations (6a), (6b), (7) applied to the heater and the cooler respectively, combining with equations (1)-(4) and the heat transfer area constraint

$$kA_h + kA_c = kA_{out},$$  \hspace{1cm} (11)

where $kA_h$ and $kA_c$ are the thermal conductance of the heater and the cooler, respectively, and $A$ is a constant. Considering all the power consumption components and the power generation components, the energy conservation equation for the system is expressed as

$$W_{sys} - P_h - P_c = W_{sys},$$  \hspace{1cm} (12)

where $W_{sys}$ is the output work of the power generation system, and $P_h$ and $P_c$ are the power consumption of the heat transfer fluid pump and the cooling water pump, respectively.

In all the twelve constraint equations, the design variables are only five which are $kA_h$, $kA_c$, $m_h$ which is the mass flow rate of the heat transfer fluid in the heater, $m_c$ which is the mass flow rate of the cooling water in the cooler and $m_0$ which is the mass flow rate of the working fluid, and the eight unknown variables which are $Q_1$, $Q_2$, $T_1$, $T_2$, $T_3$, $T_4$, $T_{h,o}$, and $T_{c,o}$ are intermediate variables. The numerous intermediate variables associated with a series of equations are useless for the optimization design and increase the computation amount greatly. Therefore, eliminating the intermediate variables and the corresponding constraint equations will simplify the optimization. For this system, $Q_1$ and $Q_2$ are completely determined by $\pi$ and $W_{net}$, so equations (2)-(4) are plugged into the others, and equation (6a) applied for the cooler and equation (6b) applied for the heater are linear dependent. On the other hand, owing to the linear temperature variables in equation (7), the six temperature parameters, i.e. $T_1$, $T_2$, $T_3$, $T_4$, $T_{h,o}$ and $T_{c,o}$, are easily eliminated. Finally, all the constraint equations remain three without intermediate variables which are equations (11), (12) and

$$T_{k,i} - \pi \tau \ T_{c,j} = \pi \tau \left(1 - \eta \right) \left(W_{sys} + P_h + P_c\right) + \frac{W_{sys} + P_h + P_c}{2 \eta m_h c_{p,h}} + \frac{\xi_h}{\eta} \left(\frac{\exp(kA_h \xi_h) + 1}{\exp(kA_h \xi_h) - 1}\right) W_{sys} + P_h + P_c,$$  \hspace{1cm} (13)

where $\xi_h = \frac{1}{m_h c_{p,h}}$, $\xi_c = \frac{1}{m_c c_{p,c}}$, $c_{p,h}$, $c_{p,c}$ and $c_{p,0}$ are the constant pressure specific heat capacities of the hot fluid, the cold fluid and the working fluid.
Based on the simplification stated above, this optimization problem comes down to an conditional extremum problem, and the constraints are equations (11)-(13) and the objective is to find the values of $kA_h$, $kA_c$, $m_h$, $m_c$, and $m_0$ that make the consumption of heat transfer fluid take its minimum.

In order to solve the conditional extremum problem, we use the Optimization Toolbox in MATLAB to calculate the optimal solution for this problem. One convenient feature of Optimization Toolbox is that it provides simple functions for finding parameters that minimize the objective while satisfying nonlinear constraints.

4. Optimization results and discussion

4.1. The optimization result and its verification

As an application of this method, a optimization problem is solved for a Brayton power cycle system which is working in the following fictitious situation: $T_{h,i} = 1700$ K, $T_{c,i} = 300$ K, $W_{sys} = 1$ MW, $kA_{total} = 50000$ W/K, $\pi = 4$. The physical property of the fluid in the system is assumed as some approximate values according to its actual values [23,24]. The constant pressure specific heat capacities of the hot fluid, the cold fluid and the working fluid are assumed as constant, and the values used in this case are: $c_{p,h} = 1500$ JK$^{-1}$ kg$^{-1}$, $c_{p,c} = 4200$ JK$^{-1}$ kg$^{-1}$, $c_{p,0} = 2000$ JK$^{-1}$ kg$^{-1}$, and the specific heat ratio of working fluid equals to 1.30. The characteristic parameters of the heat transfer fluid pump, $a_{h,0}$, $a_{h,1}$, $a_{h,2}$ and the characteristic parameters of the cooling water pump, $a_{h,0}$, $a_{h,1}$, $a_{h,2}$ are considered to be the same and they are shown in table 1, and the rotation frequency is set as $\omega_h = \omega_c = 50$Hz, and the density of the heat transfer fluid and the cooling water takes $\rho_h = 800$kgm$^{-3}$ and $\rho_c = 1000$kgm$^{-3}$. With all the known parameters and equations (11)-(13), the Optimization Toolbox gives the values of design variables when $m_h$ takes its minimum, and the result is shown in table 2.

![Figure 4. The comparison diagram between the five cases and the optimization design scheme.](image)

| $a_{h,0}$, $a_{c,0}$ | $a_{h,1}$, $a_{c,1}$ | $a_{h,2}$, $a_{c,2}$ |
|---------------------|---------------------|---------------------|
| 0.01                | 2                   | -1000               |

Table 1. The characteristic parameters of the two pumps.

| $kA_h$(W/K$^{-1}$) | $kA_c$(W/K$^{-1}$) | $m_0$(kgs$^{-1}$) | $m_c$(kgs$^{-1}$) | $m_h$(kgs$^{-1}$) |
|-------------------|-------------------|------------------|------------------|------------------|

Table 2. The optimized result with the total thermal conductance of 50000W/K.
Table 3. Six different groups of the design parameters.

| Parameter | Case 1 | Case 2 | Case 3 | Case 4 | Case 5 | Case 6 |
|-----------|--------|--------|--------|--------|--------|--------|
| $kA_h$ (WK$^{-1}$) | 25000  | 25000  | 25000  | 20000  | 20000  | 30000  |
| $kA_c$ (WK$^{-1}$) | 25000  | 25000  | 25000  | 30000  | 30000  | 20000  |
| $m_0$ (kgs$^{-1}$) | 2      | 2.5    | 3      | 2.5    | 2      | 2.5    |
| $m_c$ (kgs$^{-1}$) | 10     | 15     | 5      | 10     | 5      | 10     |

In order to validate the optimization result shown in table 2 and because of the infinitely many of combinations of the design parameters, table 3 gives six groups of design parameters which are arbitrary given, and with these parameters and solving the constraint equations the mass flow rates of heat transfer fluid in each case can be determined. The comparison diagram between the six cases and the optimization result is depicted in figure 4, and it is obvious that the mass flow rates of the hot fluid in all the six cases are larger than the optimization result, which verifies the optimum parameters somehow.

On the other hand, keeping the value of $m_c$ as shown in table 2 a univariate analysis is made and the variation of $m_h$ with the changing of $m_0$ and $kA_h$ is shown in figure 5. Figure 5 indicates that $m_h$ initially decreases and then increases with the increasing of $m_0$, and $m_h$ decreases with the increasing of $kA_h$ when $m_0$ is small and increases with the increasing of $kA_h$ when $m_0$ is relatively large. In the course of calculation, the pressure ratio remains the same which means that the thermal efficiency of the ideal Brayton cycle remains the same and is 0.274 in this case, and only the variation of the parameters of the heater changes the overall energy consumption of the power generation system. All the values of $m_h$ shown in figure 5 are greater than the optimization value and that further validates the optimization result shown in table 2. However, figure 5 shows a relatively large variation range of the mass flow rate of the working fluid $m_0$ whereas $m_h$ changes a little, and this is because the initial temperature of the heat transfer fluid remains 1700 K and the variation of $m_0$ mainly changes the temperature of the working fluid in the heater, and figure 6 shows the variation of the working fluid temperature at the inlet of the turbine with the changing of $m_0$ and $kA_h$. According to figure 6, $T_3$ decreases significantly with the increasing of $m_0$ and that is because the increasing of mass flow rate makes the working fluid flow through the heater faster and reduces its heat absorbing time. Therefore, the variation of $m_0$ mainly changes the mean heat absorbing temperature of the working fluid, not the consumption of heat transfer fluid.

From practical design and production, turbine should not work under extreme high temperature because of the restriction of the material. According to figure 6, the increasing of $m_0$ significantly reduces the inlet temperature of the turbine with a fixed output work, while the variation of the heat transfer area changes the temperature little. Meanwhile, based on figure 5, in many cases the consumption of heat transfer fluid is not much larger than the minimum value, but the work condition for turbine is greatly improved. Although this optimization does not involve the flow resistance of the working fluid and ignores the operational cost of driving the working fluid, figure 5 and figure 6 give a profound guiding significance to improve the work condition of turbine by adjusting the distribution of heat transfer area of the heat exchangers.
4.2. The influence of the compression ratio and the total heat transfer area

The compression ratio of the compressor and the total heat transfer area of the heat exchangers are directly related to the thermal efficiency and the energy consumption of the system. For a power generation system with a certain net work, it is generally accepted that large heat transfer area and high compression ratio lead to low energy consumption, and in order to analyze the influence of these two factors on energy consumption the variations of the optimal value of \( m_h \) with the changing of \( kA_h \) and \( \pi \) are depicted in figure 7 and figure 8, respectively.
Figure 7. The variation of the optimization result with the changing of total thermal conductance.

Figure 8. The variation of the optimization result with the changing of compression ratio of the compressor.

Figure 7 shows the variation of the optimization result with the changing of total thermal conductance of the heater and the cooler and constant compression ratio $\pi = 4$. In figure 7, the minimum of $m_h$ with the optimal design decreases with the increasing of the total thermal conductance. The increasing of $kA_{total}$ enhances the heat transfer capacity of the heater and the cooler, which means a lower consumption of the hot fluid can meet the heat demand in all case, so the optimal value of $m_h$ certainly decreases with the increasing of the heat transfer area of the heat exchangers, therefore the value of $kA_{total}$ should be given as a constant when the optimization proceeds, otherwise $m_h$ will be close to zero with infinite heat transfer area and it is impractical. On the other hand, although enhancing the heat transfer area of the heat exchangers definitely reduces the operation energy consumption of this system, the optimal value of $m_h$ decreases insignificantly when
$A$ is large, and this indicates that an excessive enhancing of heat transfer area is not obvious for reducing energy consumption. In reality, a large heat transfer area leads to a large size of the components and a expensive production costs, so it is advisable to conduct the optimization with synthesizing the manufacture cost and the operating energy consumption.

Figure 8 shows the variation of the optimization result with the changing of compression ratio of the compressor and constant total thermal conductance $kA_{total} = 50000WK^{-1}$. In figure 8, the minimum of $m_h$ with the optimal design decreases with the increasing of the total thermal conductance, and contrasting with figure 7 its decreasing trend is always apparent in the calculation range. For an ideal Brayton cycle, the increasing of $\pi$ enhances the thermal efficiency directly, therefore it reduces the consumption of hot fluid with a constant power generation of system. In actual design process, the pressure ratio of the compressor cannot be infinite, and combining with the actual parameters of the compressor this optimization method gives an optimal design for each situation.

5. Conclusions
A power generation system mainly involved two types of thermal processes which are the heat-work conversion processes and the heat transfer processes, and the combination of the performances of compressor, turbine and heat exchangers judges the overall energy consumption. By changing the form of the heat transfer characteristic equation for heat exchanger and applying the conception of thermal resistance to heat exchanger, we synthesized the thermodynamics analysis and the heat transfer analysis, and proposed an overall optimization method for a power generation system with Brayton cycle.

For the power generation system studied in this paper, based on the new form of the heat transfer characteristic equation we eliminated the physics equations which involves the intermediate variables, e.g. the state parameters of the working fluid, and obtained the constraint equations which only consists of the design variables, e.g. the mass flow rates of heat transfer fluid, working fluid and cooling water and the heat transfer area of heat exchangers. The final set of equation directly reflects the relations of all the design parameters and keeps out the complexity brought by a large number of variables and equations. With the constraint equations and the optimization goal, we took the optimization problem as a conditional extremum problem and used an optimization calculation toolbox of MATLAB to solve this problem. Finally, to get the minimum total energy consumption of the system, we calculated the optimal values of design parameters for each component, and the comparison of different optimization results at different situations reveals the margin of optimization when the cycle efficiency and the total heat transfer area are decided.

The optimization method proposed in this paper indicates that the overall efficiency is determined not only by the thermodynamic cycle process but also by the heat transfer performance of heat exchangers, and it gives the optimal structural and operation parameters after considering the influence of heat transfer capacity of heat exchangers on the isobaric heat absorption process and isobaric heat rejection process in Brayton cycle. Although the newly developed optimization method is just applied to a single cycle, it definitely indicates the advantages of synthesizing thermodynamic and heat transfer analysis.

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