Optimizing the power of a variable-temperature heat reservoir Brayton cycle for a nuclear power plant in space

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ABSTRACT
This study establishes a variable-temperature heat reservoir endoreversible simple closed Brayton cycle model for a nuclear power plant in space and derives its thermal efficiency (TEF) and power output (POW). The maximum POW ($P_{\text{max}}$) for a fixed total heat transfer area of a radiator panel and two heat exchangers (HEXs) is obtained by optimizing the area distributions ($f_H$, $f_L$, and $f_R$) among the two HEXs and radiator panel, the double maximum POW ($P_{\text{max},2}$) is obtained by optimizing the inlet temperature ($T_{\text{lin}}$) of the cooling fluid in the low temperature heat sink, and the triple maximum POW ($P_{\text{max},3}$) is further obtained by optimizing the thermal capacity rate matching ($C_{Wf}/C_f$) between the heat reservoir and working fluid. When $f_H$, $f_L$, and $f_R$ are optimized, $P_{\text{max}}$ increases by 4.33% compared to the initial POW ($P$); when $T_{\text{lin}}$ is furtherly optimized, $P_{\text{max},2}$ increases by 6.33% compared to $P$ and increases by 1.86% compared to $P_{\text{max}}$, with $P_{\text{max},3}$ increasing by 11.76%, 7.13%, and 5.17% compared to $P$, $P_{\text{max}}$, and $P_{\text{max},2}$, respectively.

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1. INTRODUCTION
In the face of the requirements of deep space exploration missions, establishing a high-conversion efficiency, reliable, and compact space-based power plant (SPP) that can respond to challenges has become necessary in recent years. Three ways presently exist for an SPP to provide energy: chemical, solar, or nuclear energy. Among these options, nuclear energy appears as a possible alternative for generating large amounts of energy long term and reducing fuel mass. Furthermore, the SPP would require the maximum power-to-mass ratio for space propulsion purposes. Therefore, a practical energy conversion system must strike a compromise between high conversion efficiency and compactness. This relationship must be balanced during the design process. The main components of SPP are divided into three parts: the reactor, the energy conversion device, and the radiator. The space energy conversion system can be separated into static (thermoelectric converter and thermionic conversion) and dynamic (Stirling, Brayton, and Rankine heat engines) components.

Due to the high power density, high conversion efficiency, stability, and reliability, the closed Brayton cycle and its combined cycles have been used in aircraft, the marine industry, power plants, and space-based power plants. Some scholars (Gonca & Sahin, 2016; Gonca, 2017a, 2017b, 2018; Gonca & Genc, 2019;Gonca & Başhan, 2019;Gonca & Guzel, 2022) have optimized gas turbine cycles.
cycles (Gonca, 2017b; Gonca & Genc, 2019) and gas-steam combined cycles (Gonca & Bashan, 2019; Gonca & Guzel, 2022) with exergetic (Gonca, 2017a, 2017b; Gonca & Guzel, 2022), exer-go-economic (Gonca & Guzel, 2022) and thermo-ecological (Gonca & Sahin, 2016; Gonca, 2017a, 2017b, 2018; Gonca & Genc, 2019; Gonca & Bashan, 2019) performances as the optimization objectives and analyzed the effects of different working fluids, turbine operations, and design parameters on cycle performances.

In order to establish a high conversion efficiency, reliable, and compact SPP, some scholars have introduced classical thermodynamics theory into the performance optimization of the closed Brayton cycle for SPPs (El-Genk & Tournier, 2009; Liu et al., 2020; Wang et al., 2021a; Miao et al., 2022; Toro & Lior, 2017). El-Genk and Tournier (2009) established a closed Brayton cycle model for SPP with an inert gas and binary mixture as a coolant and analyzed the influence of the working fluid (WF) on plant performance and turbine size. Their results showed that a cycle with an indirect closed Brayton cycle has higher thermal efficiency (TEF) than one with a single compressor and that the cycle TEF is almost unaffected by the WF’s molecular weight. Liu et al. (2020) took the mass of the HEXs of a closed Brayton cycle for an SPP as the optimization objective, minimizing the total mass of the plant by optimizing the key parameters of the system components with NSGA-II algorithm, thus obtaining a Pareto frontier of key parameters. Wang et al. (2021a) established a closed Brayton cycle model for an SPP with a gas-cooled reactor as the hot side of the heat reservoirs (HRs) and analyzed and optimized the cycle performances. Their results showed the highest temperature that the fuel can reach and how safe the device is under optimal operation. Miao et al. (2022) established a recompressed supercritical N2O-He mixture closed Brayton cycle model for an SPP. They comprehensively studied and optimized important parameters of the plant such as split ratio and pressure ratio and obtained an optimal TEF and closed Brayton cycle rotating unit mass for the cycle. Additionally, Toro and Lior (2017) introduced classical thermodynamics theory into the performance optimization of Stirling cycles for SPPs and analyzed the effects of main cycle parameters on the relationship between TEF and POW for Stirling cycles operating with different WFs.

The theory of finite-time thermodynamics (FTT; Andreason, 1983; Bejan, 1996; Chen et al. 1999; Berry et al., 2020; Andreason & Salamon, 2022) is an innovation of classical thermodynamic theory. FTT can consider the effects of heat transfer loss and the limitations of time and heat exchanger (HEX) area between the heat reservoir (HR) and WF which are largely ignored in classical thermodynamics. Many researchers have introduced FTT into the performance optimizations of thermal cycles and processes, including the optimal performances of the Carnot cycles (Curzon & Ahlborn, 1975; Valencia-Ortega et al., 2021), Stirling engines (Xu et al., 2022), diesel engines (Wu, Feng et al., 2021; Ge et al., 2021), dual cycles (Ge et al., 2022), Kalina cycles (Feng et al., 2020), dual-Miller cycles (Ebrahimi, 2021), organic Rankine cycles (Park & Kim, 2016; Wu, Ge et al., 2020; Feng et al., 2021), combined cycles (Gonca & Guzel, 2022; Wu et al., 2021), thermoelectric devices (Chen et al., 2020a; Chen et al., 2021a; Chen & Lorenzini, 2022a), thermal Brownian cycles (Qi et al., 2021a, 2021b; Chen et al., 2022; Qi et al., 2022a, 2022b), thermorefrigerative devices (Li & Chen, 2021; Zhang, Yang et al., 2021), blue engines (Lin et al., 2022), electron engines (Ding et al., 2021; Qui et al., 2021a), thermonic devices (Qui et al., 2021b), methane reforming (Chen et al., 2022b), chemical engines (Chen & Xia, 2022a, 2023a), chemical pumps (Chen et al., 2023a, 2023b), Brayton cycles (Ibrahim et al., 1991; Ust et al., 2006; Chen et al. 2020b, 2020c; Qui et al., 2022; Jin et al., 2022), and refrigeration cycles (Chen & Lorenzini, 2022b), as well as the optimal configurations of refrigeration cycles (Badescu, 2021; Paul & Hoffman, 2022), heat-transfer systems (Badescu, 2022; Chen & Xia, 2022b), variable-temperature-reservoir heat engines (Li & Chen, 2022; Chen & Xia, 2022c), methanol synthesis (Li et al., 2022), variable-potential-reservoir chemical engines (Chen & Xia, 2022d, 2022e, 2023b, 2023c), and commercial engines (Chen, 2011; Chen & Xia, 2022f). Thermal cycles are divided into two types based on the nature of the cycle: steady flow cycles (Chen et al., 1996; Feidt, 2017) and reciprocating cycles (Curzon & Ahlborn, 1975; Muschik & Hoffman, 2020). For the steady-flow heat engine cycle, considering the variable-temperature HR can have the cycle more closely approach the working state of actual heat engines. Therefore, some scholars have studied the steady-flow cycle under the condition of variable-temperature HRs (Ust et al., 2006; Ibrahim & Boursili, 2021).

Some scholars have introduced FTT into the performance optimization of Stirling and Carnot engine cycles for SPPs (de Moura et al., 2022a, 2022b; Wang et al., 2021b, 2022). De Moura et al. (2022a, 2022b) established a Stirling engine model for an SPP, obtained a compact system with optimal temperature conditions for TEF by optimizing the temperature of the HRs of the system, and analyzed different the effects of the Stirling engine structural parameters on final system performance. Wang et al. (2021b, 2022) respectively established endoreversible and irreversible Carnot cycle models for SPPs and obtained the plant double-maximum POW by optimizing the area distributions of the HEXs and temperature of the low temperature heat sink.

Some scholars (Zhang, Liu et al., 2021; Romano & Ribiero, 2021) introduced FTT into the performance optimizations of a closed Brayton cycle for SPPs. Zhang, Liu et al. (2021) established a supercritical CO2 closed Brayton cycle model for an SPP using a sodium-cooled reactor as the hot side of the HR, derived the relationship between TEF and POW, and obtained optimal cycle characteristics and operating parameters. Romano and Ribeiro (2021) established a regenerative closed Brayton cycle model for an SPP, obtained the optimal inlet temperature for the cold side of the HEX and optimal heat source temperature by minimizing specific mass.
Existing FTT research on closed Brayton cycle models for SPPs have not optimized HEX area or HR temperature, nor have they yet obtained the optimal inlet temperature of a cooling fluid in a low temperature heat sink or optimize thermal capacity rate matching between the HR and WF. Applying FTT to SPPs is crucial for establishing a theoretical system, and based on Ust et al. (2006), this paper will establish a variable temperature HR endoreversible closed Brayton cycle model for an SPP. For a fixed total heat transfer area of the radiator panel and two HEXs, the maximum POW of the plant will be obtained by optimizing the area distributions among the radiator panel and two HEXs, the double-maximum POW will be obtained by optimizing the inlet temperature of the cooling fluid in the low temperature heat sink, and the triple-maximum POW will be obtained by optimizing thermal capacity rate matching between the HR and WF. The study will also investigate the impacts the cycle parameters have on the triple-maximum POW.

2. MODEL DESCRIPTION AND PERFORMANCE INDICATORS

Figure 1 shows the diagram of the established model. The model consists of two parts: the first part is the ordinary closed Brayton cycle, which includes a compressor, a turbine, and two HEXs between the HRs and WF. This part has isobaric heat absorption and heat release processes involving the temperature drops \( T_1 \rightarrow T_2 \) accompanying \( Q_H \) and \( T_3 \rightarrow T_4 \) accompanying \( Q_L \) (denoted respectively as the processes 2→3 and 4→1 in Fig. 1). This part also undergoes two isentropic processes (denoted as processes 1→2 and 3→4 in Fig. 1). The second part includes a radiator panel that dissipates heat into space. The HEXs between the HRs and WF are assumed to be counter-current. The inlet/outlet temperatures of the heating and cooling fluids are \( T_{in} \) and \( T_{out} \) respectively.

The constant thermal capacity rate of the WF is \( C_{wf} \), and the thermal capacity rates of the HRs are \( C_L \) and \( C_H \). The heat conductance of the hot and cold sides of the HEX are \( U_H \) and \( U_L \); \( U_H = K \cdot F_H \) and \( U_L = K \cdot F_L \), where \( K \) is the heat transfer coefficient and \( F \) is the HEX area.

According to the properties of the WF and HEX theory, the heat flux between the HRs and WF are respectively:

\[
Q_H = C_{wf} \cdot (T_2 - T_3), \quad Q_L = C_L \cdot (T_4 - T_1)
\]

where the \( E \)s are the effectiveness values of the two HEXs:

\[
E_{HI} = \frac{1 - \exp\left[-N_{HI}(1 - C_{lw} / C_{lw}^{max})\right]}{1 - (C_{lw}^{max} / C_{lw}^{min}) \cdot \exp\left[-N_{HI}(1 - C_{lw}^{max} / C_{lw}^{min})\right]}
\]

\[
E_{LI} = \frac{1 - \exp\left[-N_{LI}(1 - C_{lw}^{max} / C_{lw}^{min})\right]}{1 - (C_{lw}^{max} / C_{lw}^{min}) \cdot \exp\left[-N_{LI}(1 - C_{lw}^{max} / C_{lw}^{min})\right]}
\]

where \( N_{HI} = U_H / C_{lw}^{max} = K_H F_H / C_{lw}^{max} \) and \( N_{LI} = U_L / C_{lw}^{max} = K_L F_L / C_{lw}^{max} \) are the respective number of heat transfer units as defined based on the minimum thermal capacity.

According to the endoreversible condition, the relationship between the four temperatures of the cycle is \( T_1 \equiv T_3 \equiv T_2 \equiv T_4 \). Defining the isentropic temperature ratio of the compressor as \( x \) gives:

\[
x = \left(\frac{T_2}{T_1}\right)^{\frac{1}{k-1}} = \left(\frac{T_4}{T_3}\right)^{\frac{1}{k-1}}
\]

where \( \pi \) is the pressure ratio of the cycle, \( m = (k-1)/k \), and \( k \) is the specific heat ratio.

The steady-state heat transfer from the radiator panel to the external environment is:

\[
Q_S = \sigma \varepsilon F_p \cdot \left(\frac{T_{in}}{T_0}\right)^4 - \left(\frac{T_{out}}{T_0}\right)^4
\]

where \( \eta \) is fin efficiency, \( \sigma \) is the Boltzmann constant, \( T_0 \) is the temperature of the space environment, \( \varepsilon \) is emissivity, and \( F_p \) is the area of the radiator panel.

According to Ust et al. (2006), outlet temperatures \( T_2 \) and \( T_4 \) of the compressor and turbine for a conventional variable temperature HR endoreversible closed Brayton cycle model are obtained as:

\[
T_2 = C_{lw}^{max} E_H (C_{HF} C_{lw}^{min} E_L) T_{in} \cdot x C_{HF} C_{lw}^{max} E_L T_{in} + C_{HF} C_{lw}^{max} E_L + C_{lw}^{max} E_L
\]

\[
T_4 = C_{HF} C_{lw}^{max} E_H (C_{HF} C_{lw}^{max} E_L) T_{in} + x C_{HF} C_{lw}^{max} E_L T_{in} + C_{HF} C_{lw}^{max} E_L + C_{HF} C_{lw}^{max} E_L
\]
From the first law of thermodynamics, one can obtain:

\[ Q_L = Q_0 \] (12)

\[ C_{\text{min}} E_{1i} (T_2 - T_{1i}) = \sigma F_i \eta_i (T_{\text{lin}}^4 - T_i^4) \] (13)

From Eqs. 11 and 13, one gets:

\[ x = \pi \left( \frac{C_{\text{min}} E_{1i} C_{\text{min}} E_{1i}}{C_{\text{min}} E_{1i}} + \frac{C_{\text{min}} E_{1i}}{C_{\text{min}} E_{1i}} \right) T_{\text{lin}} \] (14)

From Eq. 15, one then gets:

\[ \frac{Q_L}{C_{\text{min}} E_{1i}} = \frac{\sigma F_i \eta_i (T_{\text{lin}}^4 - T_i^4)}{C_{\text{min}} E_{1i}} \] (15)

From Eqs. 10 and 16, one gets:

\[ T_{\text{lin}} = \frac{(C_{\text{min}} E_{1i}) (C_{\text{min}} E_{1i}) (C_{\text{min}} E_{1i})}{(C_{\text{min}} E_{1i}) + (C_{\text{min}} E_{1i})} \] (16)

From Eqs. 1, 2, 14, and 17, one then gets:

\[ Q_L = C_{\text{min}} E_{1i} \frac{\sigma F_i \eta_i (T_{\text{lin}}^4 - T_i^4)}{C_{\text{min}} E_{1i}} \] (19)

Thus, the POW and TEF of the plant are respectively:

\[ \alpha E_F \eta_i (T_{\text{lin}}^4 - T_i^4) \]

3. POWER MAXIMIZATION

3.1. Initial design

In accordance with Ust et al. (2006), Wang et al. (2021b, 2022), and Romano & Ribiero (2021), an initial design has been performed with \( P_{\text{pow}} = 6.7 \times 10^4 \text{kW/(m}^2\text{K}) \), \( \eta = 0.6, e = 0.9, \ C_1 = 1.5 \text{kW/K}, \ C_2 = 1.2 \text{kW/K}, \ K_1 = 0.2 \text{kW/(m}^2\text{K}), \ T_{\text{lin}} = 1150 \text{K}, \ T_{\text{lin}} = 400 \text{K}, \ T = 200 \text{K}, \ F_1 = 12.24 \text{m}^2, \ F_2 = 12.24 \text{m}^2, \) and \( F_3 = 122.24 \text{m}^2 \). The POW of the initial design is \( P = 122.94 \text{kW} \).

3.2. The maximum and double-maximum POWs

A change in the area of the HEXs will also change and, thus, enabling the POW to be maximized. Assuming that the sum of the area of the three HEXs is constant:

\[ F_i = F_i, f_i = F_i \] (22)

and defining the three area distribution ratios as:

\[ \sum_{i=1}^{3} f_i = 1, 0 < f_i < 1 \text{ (i=H, L, R)} \] (23)

one can perform the POW maximization with respect to the area ratios. The obtained maximum POW is \( P_{\text{max}} \).

Figure 2 reflects \( P \) versus \( f_t \) and \( f_t \) with \( F_t = 153.8 \text{ m}^2, \ C_1 = 1.5 \text{kW/K}, \ K_1 = 0.2 \text{kW/(m}^2\text{K}), \ T_{\text{lin}} = 400 \text{K}, \) \( \eta_{\text{opt}} = 122.94 \text{kW} \), with \( P_{\text{max}} \) (the peak of the curve in Fig. 3) is obtained with HEX's area allocations \( f_i= f_H, f_L \) and \( f_R \) with \( f_H \) and \( f_R \) as optimization variables and the fixed \( f_L \), with the corresponding optimal area allocations of the HEXS and radiator panel being \( f_{H_{\text{opt}}}, f_{L_{\text{opt}}} \) and \( f_{R_{\text{opt}}} \), respectively. \( P_{\text{max}} \) is 128.26 kW, with \( P_{\text{max}} \) increasing by about 4.33% compared to the POW (P) of the initial design.

The double-maximum POW (\( P_{\text{max,2}} \)) is further obtained by optimizing the inlet temperature of the cooling fluid \( (T_{\text{lin}}) \) based on \( f_{H_{\text{opt}}, f_{L_{\text{opt}}} \text{, and } f_{R_{\text{opt}}} \text{. Figure 3 reflects the maximum POW (\( P_{\text{max,2}} \)) versus } T_{\text{lin}} \text{. In the calculation, almost all of the parameters are the same as those for Figure 2 except } T_{\text{lin}}, \text{ which is variable. } P_{\text{max,2}} \text{ (the peak of the curve in Fig. 3) is 130.65; with } P_{\text{max,2}} \text{ increasing by about 1.86% compared to maximum POW (\( P_{\text{max}} \)), and } P_{\text{max,2}} \text{ increasing by about 6.27% compared to the POW (P) of the initial design.}

Here the study will perform some parameter analyses regarding the maximum POW and double-maximum POW. Figures 4 and 5 reflect the effects of the thermal capacity rate \( C_1 \text{, the WF and heat transfer coefficients of the HEXs (} K_1 \text{, and } K_2 \text{) on the relationship between } P_{\text{max}}, \text{ and the corresponding TEF (} \eta_{\text{opt}} \text{, the relationship between } P_{\text{max}}, \text{ and the inlet temperature of the cooling fluid (} T_{\text{lin}} \text{, the relationship between } P_{\text{max}}, \text{ and } f_{H_{\text{opt}}, f_{L_{\text{opt}}} \text{, and } f_{R_{\text{opt}}} \text{, as well as the relationship between } P_{\text{max}}, \text{ and the corresponding pressure ratio (} \eta \text{).}

Figure 4 reflects \( P_{\text{max,2}} \text{, } T_{\text{lin}}, \text{ and } f_{H_{\text{opt}}, f_{L_{\text{opt}}} \text{, and } f_{R_{\text{opt}}} \text{ at double-maximum } P_{\text{max,2}} \text{, while decreasing } P_{\text{max,2}}, \text{ the hot side of the HEX area allocation } \} f_{H_{\text{opt}}, f_{L_{\text{opt}}} \text{, and the inlet temperature of the cooling fluid (} T_{\text{lin}} \text{, the relation between } P_{\text{max,2}}, \text{ and } f_{H_{\text{opt}}, f_{L_{\text{opt}}} \text{, and } f_{R_{\text{opt}}} \text{, as well as the relationship between } P_{\text{max,2}}, \text{ and the corresponding pressure ratio (} \eta \text{).}
kW; respective \( T_{\text{Lin}} \) \( P_{\text{max}} \) values of 428 K, 419 K, and 413 K; respective values of 0.468, 0.473, and 0.476; respective \( \eta_{\text{opt}} \) \( P_{\text{max}} \) values of 0.171, 0.156, and 0.151; and respective \( f_{H_{\text{opt}}} \) \( P_{\text{max}} \) values of 0.658, 0.68, and 0.698. Increasing \( C_{\text{wf}} \) from 1.5 to 2.5 decreases \( P_{\text{max}} \) by about 5.82% and \( T_{\text{Lin}} \) \( P_{\text{max}} \) by about 3.50%, increases \( \eta_{\text{opt}} \) \( P_{\text{max}} \) by about 1.71%, decreases \( f_{H_{\text{opt}}} \) \( P_{\text{max}} \) by about 11.70%, and increases \( f_{R_{\text{opt}}} \) \( P_{\text{max}} \) by about 5.71% as well as \( \pi_{\text{opt}} \) \( P_{\text{max}} \) by about 5.49%.

The optimal pressure ratio \( (\pi_{\text{opt}}) P_{\text{max}} \) corresponding to \( P_{\text{max}} \) should be pointed out to have one-to-one correspondence to the optimal inlet temperature of the cooling fluid \( (T_{\text{Lin}}) P_{\text{max}} \); namely, only one independent variable is found between the inlet temperature of the cooling fluid \( (T_{\text{Lin}}) \) and the pressure ratio \( (\pi) \).

Figure 2. The relations of \( P \) versus \( f_{H} \) and \( f_{L} \).

Figure 3. The relations of \( P_{\text{max}} \) versus \( T_{\text{Lin}} \).

Figure 5 reflects the \( P_{\text{max}} \) \( T_{\text{Lin}} \), \( P_{\text{max}} \) \( \eta_{\text{opt}} \), \( P_{\text{max}} \) \( f_{H_{\text{opt}}} \), \( P_{\text{max}} \) \( f_{R_{\text{opt}}} \), and \( P_{\text{max}} \) \( \pi \) values under different \( K_{1}=K_{2} \) and. When \( C_{\text{wf}} \)=1.5, increases to \( K_{1} \) and \( K_{2} \) will increase \( P_{\text{max}} \), \( T_{\text{Lin}} \), and \( f_{R_{\text{opt}}} \) \( P_{\text{max}} \) while decreasing \( f_{H_{\text{opt}}} \) \( P_{\text{max}} \) and \( \eta_{\text{opt}} \) \( P_{\text{max}} \). When \( K_{1} \) and \( K_{2} \) are 0.1, 0.2, and 0.3, \( P_{\text{max}} \) has respective values of 111.94kW, 130.65kW, and 139.75kW; \( (T_{\text{Lin}}) P_{\text{max}} \) has respective values of 426 K, 428 K, and 429.8 K; \( (\eta_{\text{opt}}) P_{\text{max}} \) has respective values of 0.469, 0.468, and 0.467; \( (f_{H_{\text{opt}}}) P_{\text{max}} \) has respective values of 9.15, 9.10, and 9.08; \( (f_{R_{\text{opt}}}) P_{\text{max}} \) has respective values of 0.213, 0.171, and 0.147; and \( (\pi_{\text{opt}}) P_{\text{max}} \) has respective values of 0.573, 0.658, and 0.706. Increasing \( K_{1} \) and \( K_{2} \) from 0.1 to 0.3 increases \( P_{\text{max}} \) by about 24.84% and \( T_{\text{Lin}} \) \( P_{\text{max}} \) by about 0.89%, decreases \( \eta_{\text{opt}} \) \( P_{\text{max}} \) by about 0.426% and \( f_{H_{\text{opt}}} \) \( P_{\text{max}} \) by about 30.99%, increases \( f_{R_{\text{opt}}} \) \( P_{\text{max}} \) by about 23.21%, and decreases \( \pi_{\text{opt}} \) \( P_{\text{max}} \) by about 0.765%.

Figure 4. Curves for \( P_{\text{max}} \) versus (a) \( T_{\text{Lin}} \); (b) \( \eta_{\text{opt}} \); (c) \( f_{H_{\text{opt}}} \); (d) \( f_{R_{\text{opt}}} \); and (e) \( \pi \) for various \( C_{\text{wf}} \) values.
3.3. The triple-maximum POW

The triple maximum POW ($P_{\text{max},3}$) has been further obtained by optimizing the thermal capacity rate matching ($C_{wf}/CH$) between the HRs and WF based on $f_{H\text{opt}}, f_{L\text{opt}}, f_{R\text{opt}}$, and $(T_{L\text{in}})P_{\text{max},2}$. When almost all of the parameters are the same as those for Figure 3 except for $C_{wf}/CH$, which is variable, the optimization shows the triple maximum POW to be $P_{\text{max},3}=137.40$ kW, as shown by the peak of Curve 2 in Figure 6. $P_{\text{max},3}$ increases by about 5.17% compared to the double-maximum POW ($P_{\text{max},2}$), by about 7.13% compared to the maximum POW ($P_{\text{max}}$), and by about 11.76% compared to the POW ($P_{\text{max}}$) of the initial design.

Now we further study the effects of the thermal capacity ratio ($CL/CH$) of the HRs and $K_1$ on the triple-maximum POW ($P_{\text{max},3}$). Figures 6 and 7 reflect $P_{\text{max},2}-C_{wf}/CH$ under different $CL/CH$ and $K_1$ values, respectively. One can see that as $C_{wf}/CH$ increases, $P_{\text{max},2}-C_{wf}/CH$ reflects a stable parabolic-like change, and an optimal $C_{wf}/CH$ ($[C_{wf}/CH]_{opt}$) is found for the cycle to reach triple-maximum POW ($P_{\text{max},3}$; i.e., the peaks of the curves).

Figure 6 reflects $P_{\text{max},2}-C_{wf}/CH$ under different $C_L/CH$. When $K_1=K_2=0.2$ kW/(m$^2$K), increasing $CL/CH$ increases both $P_{\text{max},3}$ and ($C_{wf}/CH)_{opt}$. $C_{wf}/CH$ values of 0.6, 1.0, and 1.6 result in respective $P_{\text{max},3}$ values of 116.57, 137.40, and 148.31 kW and corresponding ($C_{wf}/CH)_{opt}$ values of 0.75, 1.0, and 1.23. When $C_{wf}/CH$ increases from 0.6 to 1.6, $P_{\text{max},3}$ increases by about 27.23%, while ($C_{wf}/CH)_{opt}$ increases by about 64%.

Figure 7 reflects the $P_{\text{max},2}-C_{wf}/CH$ curve for different $K_1$ values. When $K_1=K_2=0.2$ kW/(m$^2$K) and $C_{wf}/CH=1.6$, increasing $K_1$ results in $P_{\text{max},3}$ increasing and ($C_{wf}/CH)_{opt}$ decreasing. For $K_1$ values of 0.2, 0.3, and 0.4, $P_{\text{max},3}$ has respective values of 148.31, 153.95, and 157.38 kW and corresponding ($C_{wf}/CH)_{opt}$ values of 1.23, 1.20, and 1.18. When $K_1$ increases from 0.2 to 0.4, $P_{\text{max},3}$ increases by about 6.12%, while ($C_{wf}/CH)_{opt}$ decreases by about 4.07%. 

![Figure 5 Curves for $P_{\text{max}}$ versus (a) $T_{L\text{in}}$; (b) $\eta_{opt}$; (c) $f_{H\text{opt}}$; (d) $f_{R\text{opt}}$; and (e) $\pi$ for different $K_1$ and $K_2$ values.](image)

![Figure 6 $P_{\text{max},2}$ versus $C_{wf}/CH$ for different $C_L/CH$ values.](image)

![Figure 7 $P_{\text{max},2}$ versus $C_{wf}/CH$ for different $K_1$ values.](image)
4. CONCLUSION

Based on Ust et al. (2006), this study has established a variable temperature HR endoreversible closed Brayton cycle for an SPP and derived the relationships between POW (P) and inlet temperature of the cooling fluid in a low temperature heat sink, as well as between the TEF and inlet temperature of the cooling fluid. For a fixed total heat transfer area of two HEXs and one radiator panel, the maximum POW (Pmax) of the plant is obtained by optimizing the area distributions (PHeX and PfR). The double-maximum POW (Pmax2) is obtained by optimizing the inlet temperature of the cooling fluid (Tlin), and the triple-maximum POW (Pmax3) is further obtained by optimizing the thermal capacity rate matching (Cwf/CHeX) between the HRs and WF. The optimization effects are obvious. This study has researched the impacts of plant parameters on optimal performance, and the main conclusions are as follow:

1. Optimal PHeX, Pft, and PfR values exist for having the cycle reach Pmax1. Optimal Tlin and optimal PHeX, Pft, and PfR values exist for having the cycle reach Pmax2. The curve Pmax2/CHeX/CfR reflects a stable parabolic-like change with an (CH/CfR)opt value that enables the cycle to reach Pmax2.

2. When PHeX and PfR are optimized, Tlin = 400 K, the POW of the initial design plan is P = 122.94. When PHeX, PfR, and Tlin are optimized and Tlin = 400 L, the maximum POW is Pmax = 128.26 kW, with Pmax increasing by about 4.33% compared to P. When further optimizing Tlin, the double-maximum POW becomes Pmax2 = 130.65 kW, with Pmax2 increasing by about 1.86% compared to Pmax and by about 6.27% compared to P. When further optimizing Cwf/CfR, the triple-maximum POW becomes Pmax3 = 137.40 kW, with Pmax3 increasing by about 5.17% compared to Pmax2, by about 7.13% compared to Pmax, and by about 11.76% compared to P.

3. When PHeX, PfR, and Tlin are optimized, increasing Cwf from 1.5 to 2.5 decreases Pmax by about 5.82% and (Tlin)P by about 3.50%, increases (ηopt/Pmax) by about 1.71%, decreases (fHeX/Pmax) by about 11.70%, and increases (fR/Pmax) by about 5.49%. Increasing Kf and Kr from 0.1 to 0.3 increases Pmax by about 24.83% and (Tlin)P by about 0.89%, decreases (ηopt/Pmax) by about 0.426% and (fHeX/Pmax) by about 30.99%, increases (fR/Pmax) by about 23.21%, and decreases (ηopt/Pmax) by about 0.765%.

4. When PHeX, PfR, and Tlin are optimized, CHeX/CfR is optimized, increasing CfR/CHeX from 0.6 to 1.6 increases Pmax by about 27.23% and (CH/CfR)opt by about 64%. Increasing Kr from 0.2 to 0.4 increases Pmax by about 6.12% and decreases (CH/CfR)opt by about 4.07%.

5. Using FTT to optimize the closed Brayton cycle for an SPP has obtained the optimal area distribution and optimal inlet temperature of the cooling fluid. The optimization results provide a theoretical basis for the design of a heat exchanger structure and for the selection of its temperature in a space-based power plant. Therefore, FTT is shown to be an important tool for studying SPPs.

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DATA AVAILABILITY STATEMENT

The published publication includes all graphics and data collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

FINANCIAL DISCLOSURE

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