Optimization of Power and Thermal Management System of Hypersonic Vehicle with Finite Heat Sink of Fuel

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Abstract: The scramjet of hypersonic vehicles faces severe high-temperature challenges, but the heat sink available for scramjet cooling is extremely finite. It is necessary to optimize its power and thermal management system (PTMS) with a finite heat sink of hydrocarbon fuel. This paper proposes a two-level optimization method for the PTMS of hypersonic vehicles at Mach 6. The PTMS is based on a supercritical carbon dioxide (SCO₂) closed Brayton cycle, and its heat sink is airborne hydrocarbon fuel. System-level optimization aims to obtain the optimal system parameters for the PTMS. The minimum fuel weight penalty and the minimum heat sink consumption of fuel are the optimization objectives. The segmental (SEG) method is used to analyze the internal temperature distribution of fuel–SCO₂ heat exchangers in the system-level optimal solution set. This ensures the selected optimal solutions meet the requirement of a pinch temperature difference greater than or equal to 10 °C. Further, the component-level optimization for the fuel–SCO₂ heat exchanger is carried out based on the selected optimal solutions. The lightest weight of the heat exchanger and the minimum entropy production are the optimization objectives in this step. Finally, the optimal system parameters and the optimal key component parameters can be searched using this presented two-level optimization method.

Keywords: power and thermal management system; finite heat sink; multi-objective optimization; fuel weight penalty; entropy production

1. Introduction

One of the key technologies of hypersonic flight is the cooling of the scramjet [1–3]. The combustion chamber of the scramjet is in a harsh thermal environment, which even the most advanced composite materials cannot withstand [4]. Traditional aircraft can use cooling air as a heat sink [5]. For hypersonic vehicles, airflow cannot be used as a heat sink due to the excessive temperature, so hydrocarbon fuel becomes the main heat sink [6]. Therefore, the hypersonic vehicle faces the tough problem of the finite heat sink. The design of the power and thermal management system (PTMS) should be paid more attention to.

The application of regenerative cooling technology in X-51A flight tests successfully verified the feasibility of scramjet cooling with endothermic hydrocarbon fuel [7]. It made full use of the chemical heat sink of fuel through the thermal cracking reaction in the cooling channels of the scramjet [8]. The PTMS based on fuel vapor turbine was proposed for the first time in the Hy-Tech program of the United States. In this system, the fuel pyrolysis vapor expands to drive the fuel pump [9]. Zhang et al. [10] evaluated the working capacity performance of the fuel vapor turbine and found that the fuel vapor turbine has enough power to drive a generator, in addition to a fuel pump. The closed Brayton cycle based on supercritical carbon dioxide (SCO₂) utilizes SCO₂ to exchange the high-temperature heat from the scramjet. This cycle avoids the direct heat exchange between the scramjet
and fuel, which is easy to coke, and blocks the cooling channels when the incomplete pyrolysis reaction occurs [11–13]. Compared with hydrocarbon fuel, \( \text{SCO}_2 \) is more reliable and stable [6]. It converts part of the high-temperature waste heat of the scramjet into electricity, and the fuel only needs to cool the residual heat. It can improve the cooling capacity of the fuel heat sink. Guo et al. [14] summarized the advantages and disadvantages of the above schemes and proposed a new PTMS based on a \( \text{SCO}_2 \) cycle and a fuel vapor turbine. This PTMS met the cooling and high-power electricity demands at the same time for long-endurance hypersonic vehicles.

Many scholars conducted optimization research on the PTMS of hypersonic vehicles. Cheng et al. [6] optimized and compared the electric power performance between the simple recuperated and recompressing \( \text{SCO}_2 \) cycles. They found that the former utilizes more cooling capacity of fuel. Miao et al. [12] compared the performance of three typical closed Brayton cycle layouts, namely simple, recuperated and \( \text{SCO}_2 \) cycles. They proposed an improved scheme of the \( \text{SCO}_2 \) cycle with lower cooling fuel consumption. Marchionni et al. [15] studied eight different \( \text{SCO}_2 \) cycles and revealed that the complex \( \text{SCO}_2 \) cycle configurations led to higher efficiency and costs. The existing studies on the PTMS of hypersonic vehicles mainly focused on system scheme design and thermodynamic characteristics optimization. So far, few studies considered the design of system-level and component-level optimization at the same time.

The total mass of the take-off period method is often used to evaluate the performance of airborne equipment [16,17]. It converts various losses caused by the analyzed system into the mass of fuel consumed to transport these masses and generate the required power [18]. Total fuel weight penalty can evaluate and compare different schemes of PTMSs.

Considering fuel weight penalty and power-to-weight Ratio (PWR), the closed Brayton cycle based on \( \text{SCO}_2 \) for hypersonic vehicles usually adopts a printed circuit heat exchanger (PCHE) [13,19–23]. PCHE is a high-temperature heat exchanger that exchanges the heat from \( \text{SCO}_2 \) to fuel. Because the physical properties of \( \text{SCO}_2 \) and fuel vary greatly with temperature and pressure [12], it is necessary to adopt the segmental (SEG) method. It divides PCHE into several sub-heat exchangers, and then the detailed temperature of each sub-heat exchanger can be calculated [24]. The SEG method can be used to solve the question of the unreasonable pinch temperature difference and analyze the feasibility of heat transfer. Li et al. [24] proved that the maximum deviation between the heat exchange rate predicted by the SEG method and the experimental data was less than 5%. Entropy generation should also be considered while minimizing the weight of the heat exchanger [25]. Bejan stated that entropy generation should play a central role in heat transfer analysis [26]. Guo et al. proved that the heat transfer entropy generation is far larger than the frictional entropy generation by the SEG method [27].

For hypersonic vehicles with finite heat sink, a concerning goal is to optimize a PTMS to absorb as much heat and produce as much power as possible. In this paper, a two-level optimization method for the PTMS based on a \( \text{SCO}_2 \) closed Brayton cycle is established by non-dominated sorting genetic algorithms (NSGA-II) [28,29]. Simulation models are validated using published experimental data. The effects of optimization variables on the PTMS are discussed in detail. Furthermore, the optimal parameters of the system and key components are determined, and further analysis is carried out to compare the optimization results.

2. Method

2.1. System Description

For the traditional regenerative cooling scheme for the scramjet, the hydrocarbon fuel is cracked in the cooling channels of the scramjet wall. This easily leads to coke and blocks cooling channels, which causes the scramjet to be scrapped after a long-endurance hypersonic flight. Therefore, the PTMS based on \( \text{SCO}_2 \) closed Brayton cycle has been studied in recent years [6,12–14], as shown in Figure 1. It moves the easily coking process from the cooling channels of the scramjet wall to the fuel–\( \text{SCO}_2 \) heat exchanger. Compared
with the high cost of the scramjet, it is economical to replace the blocked heat exchanger. At the same time, the \( \text{SCO}_2 \) scheme can convert part of scramjet waste heat into electric energy to meet the power demand of hypersonic vehicles. In the \( \text{SCO}_2 \) cycle, PCHE is generally adopted for fuel–\( \text{SCO}_2 \) heat exchangers because of its high heat transfer efficiency and because it is lightweight [13]. Considering the space limitations in the vehicle cabin, it is necessary to comprehensively balance between a thermal design and a geometric design of PCHE.

![Diagram of PTMS based on \( \text{SCO}_2 \) closed Brayton cycle.](image)

**Figure 1.** Scheme of PTMS based on \( \text{SCO}_2 \) closed Brayton cycle.

### 2.2. Two-Level Optimization Method

The two-level optimization method is presented to optimize the PTMS based on the \( \text{SCO}_2 \) closed Brayton cycle with the finite heat sink of hydrocarbon fuel at 6 March. The inputs of the system-level optimization are fuel mass flow, heat production of the scramjet, and \( PWR \) of PCHE.

The multi-objective evolutionary algorithm, NSGA-II, is used to complete two-level optimization. It has been widely used in system optimization and heat exchanger design [29–31]. According to Ref. [29], the population size is set to 100, the number of generations is set to 100, the cross-over probability is set to 0.9, and the mutation probability is set to 0.1.

In the system-level optimization, the objective functions are to minimize the fuel weight penalty and heat sink consumption of fuel, which can be expressed by a vector:

\[
\overrightarrow{f_s}(X_s) = [f_{s,1}(X_s), f_{s,2}(X_s)]
\]

where \( f_{s,1}(X_s) \) and \( f_{s,2}(X_s) \) represent \( \min (M_{\text{total}}) \) and \( \min (Q_{hs}) \), respectively; \( X_s \) represents the system-level optimization variable vector and can be expressed as:

\[
X_s = [x_{s,1}, x_{s,2}, x_{s,3}]
\]

where \( x_{s,1}, x_{s,2} \) and \( x_{s,3} \) represent the compressor inlet pressure \((P_{5,1}, 7.4–11 \text{ MPa})\), heat exchanger efficiency \((\eta_{hx}, 80–95\%)\), and compressor pressure ratio \((\pi_C, 2–5)\), respectively.

The constraints in the system-level optimization are defined as follows:

\[
\begin{align*}
T_{\text{max}} &< 1000 ^\circ \text{C} \\
500 ^\circ \text{C} &< T_{c,0} < 680 ^\circ \text{C}
\end{align*}
\]

where \( T_{\text{max}} \) and \( T_{c,0} \) represent the maximum temperature of the \( \text{SCO}_2 \) cycle and outlet temperature of the cold side of PCHE, respectively.
After the system-level optimization, the SEG method is used to analyze the solution set of system-level optimization, which ensures the selected optimal solutions meet the pinch temperature difference ($\Delta T_{\min}$) greater than or equal to 10 °C. The component-level optimization for the fuel–SCO$_2$ heat exchanger is carried out based on the above-selected solutions.

In component-level optimization, the objective functions can be expressed by a vector:

$$\vec{f}_b(X_b) = [f_{b,1}(X_b), f_{b,2}(X_b)]$$

where $f_{b,1}$ ($X_b$) and $f_{b,2}$ ($X_b$) represent the lightest weight and the minimum entropy production ($S_g$) of PCHE, respectively; $X_b$ represents the component-level optimization variable vector and can be expressed as:

$$X_b = [x_{b,1}, x_{b,2}, x_{b,3}]$$

where $x_{b,1}$, $x_{b,2}$ and $x_{b,3}$ represent the channel width ($w_c$, 1–2 mm), core width ($W$, 0–1 m) and core height ($H$, 0–1 m), respectively.

The constraints in the component-level optimization are defined as follows:

$$\begin{cases}
P_{\text{loss}} < 2% \\
L < 1m \\
PWR > 10
\end{cases}$$

where $P_{\text{loss}}$ and $L$ represent pressure loss in PCHE and channel length, respectively.

It takes the lightest weight of PCHE and the minimum.

The two-level optimization process is shown in Figure 2.

![Figure 2. Two-level optimization process.](image)

The basic parameters of optimization are shown in Table 1.

| Parameter                        | Value     |
|----------------------------------|-----------|
| Flight speed (Ma)                | 6         |
| Flight height (H)                | 22 km     |
| Fuel mass flow rate for combustion ($\dot{m}_c$) | 0.5 kg/s  |
| The inlet temperature of the cold side of PCHE ($T_{c,n}$) | 50 °C     |
| The outlet temperature of the cold side of PCHE ($T_{c,0}$) | 500–680 °C |
| Specific speed of compressor and fuel vapor turbine ($N_s$) | 0.4       |
| The efficiency of the compressor ($\eta_C$) | 0.9       |
Table 1. Cont.

| Parameter                                                   | Value     |
|-------------------------------------------------------------|-----------|
| The efficiency of fuel vapor turbine \( \eta_{fT} \)       | 0.75      |
| The efficiency of the turbine in the SCO\(_2\) cycle \( \eta_{ST} \) | 0.9      |
| The efficiency of the fuel pump \( \eta_{P} \)            | 0.7       |
| The efficiency of the generator \( \eta_{G} \)            | 0.9       |
| Pinch temperature difference \( \Delta T_{\text{min}} \)  | \( \geq 10 \degree \text{C} \) |
| The initial power-to-weight ratio of PCHE \( \text{PWR} \) | 10        |
| Maximum temperature of SCO\(_2\) cycle \( T_{\text{max}} \) | < 1000    |
| Relative pressure loss coefficient of PCHE \( \xi_{\text{PCHE}} \) in system-level optimization | 2\%     |
| The maximum relative pressure loss coefficient of PCHE \( \xi_{\text{PCHE}} \) | 2\%     |
| Maximum relative pressure loss coefficient of cooling channels \( \xi_{cc} \) | 2\%     |
| Heat dissipated from the scramjet wall at Mach 6 \( Q_{\text{total}} \) | 1350 kW |
| Lift–drag ratio                                            | 2.95      |
| TSFC \((\text{s}^{-1})\)                                   | 0.001     |
| Inlet pressure of the cold side of PCHE \( P_{c,n} \)     | 5 MPa     |
| Inlet pressure of fuel pump \( P_{\text{p,in}} \)         | 0.1 MPa   |
| Flight duration                                            | 1 h       |
| Pressure drop ratio of fuel vapor turbine \( \pi_{fT} \)  | 3         |
| Compressor inlet pressure \( P_{S1} \)                    | \( \geq 7.4 \) MPa |
| Compressor pressure ratio \( \pi_{c} \)                    | 2~5       |

3. Simulation Model and Its Verification

3.1. Simulation Model

(1) Cooling channel model in the scramjet wall

Referring to X-51A, the fuel mass flow rate is 0.5 kg/s at Mach 6 [7]. According to the classic Lander–Nixon diagram, the demand for fuel heat sink capacity for scramjet cooling is about 1.8~2.7 MJ/kg at Mach 6 [32]. For safety reasons, the maximum value of 2.7 MJ/kg is selected in this paper. Thus, the heat production of the scramjet is 1350 kW. All the scramjet heat is assumed to be taken away by SCO\(_2\) in the cooling channels.

The relative pressure loss coefficient in cooling channels is defined as follows:

\[
\xi_{cc} = \frac{P_{S,2} - P_{S,3}}{P_{S,2}} \tag{7}
\]

where \( P_{S,2} \) and \( P_{S,3} \) are the inlet and outlet pressures of cooling channels, respectively, MPa; \( \xi_{cc} \) is the relative pressure loss coefficient in cooling channels and is set to 2\% according to Ref. [12].

(2) Turbomachinery model

This type of single-stage radial turbomachinery is adopted in this paper because the power output is less than 300 kW [33–35]. For the compressor and turbine, isentropic efficiencies are respectively defined as:

\[
\eta_{C} = \frac{(h_{S,2s} - h_{S,1})}{(h_{S,2} - h_{S,1})} \tag{8}
\]

\[
\eta_{T} = \frac{(h_{S,3} - h_{S,4s})}{(h_{S,3} - h_{S,4s} \text{kJ/kg})} \tag{9}
\]

where \( \eta_{C} \) and \( \eta_{T} \) are the isentropic efficiencies of compressor and turbine, respectively, \%; \( h_{S,1}, h_{S,2} \) and \( h_{S,2s} \) are the inlet specific enthalpy, the outlet specific enthalpy, and the ideal outlet specific enthalpy of the compressor, respectively, kJ/kg; \( h_{S,3}, h_{S,4} \) and \( h_{S,4s} \) are the inlet specific enthalpy, the outlet specific enthalpy and the ideal outlet specific enthalpy of the turbine, respectively, kJ/kg.
Balje’s chart summarized the dimensionless parameters of the total efficiency for the turbine and compressor. The specific speed and specific diameter are calculated as [36]:

\[ N_s = NV^{1/2} / H_{ad}^{3/4} \]  \hspace{1cm} (10)
\[ D_s = DH_{ad}^{1/4} / V^{1/2} \]  \hspace{1cm} (11)

where \( N_s \) is the specific speed; \( D_s \) is the specific diameter; \( D \) is the diameter, m; \( N \) is shaft speed, \text{rad/min}; \( V \) is the volume flow rate, \text{m}^3/\text{s}; \( H_{ad} \) is the adiabatic enthalpy increase, kJ.

For the \( \text{SCO}_2 \) cycle, the following formulas are adopted [37]:

\[ D_{s,\text{comp}} = 2.719N_{s,\text{comp}}^{-1.092} \]  \hspace{1cm} (12)
\[ D_{s,\text{turb}} = 2.056N_{s,\text{turb}}^{-0.812} \]  \hspace{1cm} (13)

The weight of turbomachinery is calculated as [38]:

\[ m = k_0 \times D^2 \]  \hspace{1cm} (14)

where \( m \) is the weight of the turbomachinery, kg; \( k_0 \) is the weight coefficient, and \( k_0 = 180 \) [38,39].

(3) Heat exchanger model

In the \( \text{SCO}_2 \) cycle, the PCHE is selected because of its high-heat transfer efficiency and low weight and volume [13]. The physical properties of \( \text{SCO}_2 \) dramatically change near its critical point. To obtain higher accuracy, the SEG method is used to complete the design of PCHE. The PCHE adopts a semicircular straight channel because of its low pressure loss coefficient. The basic configuration of PCHE is shown in Figure 3 [24].

![Figure 3. PCHE configuration and basic parameters.](image)

In this paper, Incoloy 907 is selected as the material of PCHE, and its maximum allowable temperature is about 1000 °C [40]. The porosity of PCHE is \( \phi = 75.8\% \). The dimensions of the channels are \( d_c / w_c = 0.5, l_p / w_c = 0.51, \) and \( l_f / w_c = 0.015 \).

The heat transfer of PCHE is assumed to be one-dimensional along its flow direction and ignores its heat transfer to the environment [24]. Then, the PCHE is divided into \( n \) sub-heat exchangers with the same heat exchange capability, as shown in Figure 4. So, \( Q_i = Q_{\text{total}} / n \) and \( n = 20 \) in this paper [27]. \( T_{h,i-1} \) and \( T_{c,i} \) represent hot side and cold side inlet temperatures of the \( i \)th sub-heat exchanger, separately. \( P_{h,i-1} \) and \( P_{c,i} \) represent hot side and cold side inlet pressures of the \( i \)th sub-heat exchanger, separately. The size, weight, and other parameters of PCHE can be obtained by the following formulas.
For the sub-heat exchanger $i$, the local energy balance is defined as Equation (15):

$$Q_i = m_h (h_{hi,i-1} - h_{hi,i}) = m_c (h_{ci,i-1} - h_{ci,i})$$  \hspace{1cm} (15)

where $Q_i$ is the heat transfer of each sub-heat exchanger, kW; $m$ is the mass flow rate, kg/s; $h$ is the specific enthalpy, kJ/kg; Subscripts $h$ and $c$ represent the hot side and cold side of the heat exchanger, respectively.

The local heat transfer equation can be given as Equation (16):

$$Q_i = U_i A_{h,i} \left( \frac{T_{hi,i-1} - T_{ci,i-1}}{T_{hi,i-1} - T_{ci,i}} \right)$$  \hspace{1cm} (16)

where $A_{h,i}$ represents the heat exchange area of the hot side, m$^2$; $U_i$ represents the total convective heat transfer coefficient; and kW/(m$^2$·K), and is calculated with Equation (17):

$$\frac{1}{U_i} = \frac{1}{\alpha_{h,i} \eta_{h,i}} + \frac{A_{s,h,i}}{\alpha_{c,i} \eta_{c,i} A_{s,c,i}} + \frac{A_{s,h,i} t_w}{A_{s,w,i} k_w}$$  \hspace{1cm} (17)

where $t_w$ represents wall thickness, mm; $k_w$ represents the thermal conductivity of metal wall, kW/(m·K); $A_{s,w,i}$ represents the heat exchange area of metal wall, m$^2$; $\alpha$ represents the heat transfer coefficient; and kW/(m$^2$·K); $\eta_i$ represents the overall surface efficiency defined in Equation (18):

$$\eta_i = \frac{\eta_f + \eta_f \pi w_c}{w_c + \pi w_c} = \frac{2 + \eta_f \pi}{2 + \pi}$$  \hspace{1cm} (18)

where $w_c$ represents the channel width, m; $\eta_f$ represents the fin efficiency of the semicircular channel, and is defined in Equation (19):

$$\eta_f = \frac{7.495 m^2 + 128 t^* + 63.984 m t^* + 30.996 m + 114.31 t^* + 20}{25 m^2 + 128 t^* + 120 m t^* + 45 m + 112 t^* + 20}$$  \hspace{1cm} (19)

where $m$ and $t^*$ are dimensionless numbers and normalized fin thickness, separately.

$m$ and $t^*$ are calculated with Equations (20) and (21), separately:

$$m = \frac{(\pi + 2) \text{Nu} \lambda}{2 \pi \lambda w_c}$$  \hspace{1cm} (20)

$$t^* = \frac{t_f}{2 w_c}$$  \hspace{1cm} (21)

where $\lambda$ represents the thermal conductivity of the fluid, kW/(m·K); $\text{Nu}$ is the Nusselt number.

The following formulas are used to calculate laminar and turbulent [24], separately.

$$f = \begin{cases} 15.78 / \text{Re} & \text{Re} \leq 2000 \\ 4.089 & \text{Re} > 2000 \end{cases}$$  \hspace{1cm} (22)
\[
\begin{align*}
    f &= (1.82 \log_{10} Re - 1.64)^{-2} / 4 \\
    Nu &= \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)}
\end{align*}
\]  

(23)

where \( f \) is fanning friction factor; \( Re \) and \( Pr \) are the Reynolds number and Prandtl number, separately.

The hydraulic diameter is calculated with Equation (24):

\[
d_{h} = \frac{\pi w_{c}}{\pi + 2}
\]

(24)

The pressure drop is calculated with Equation (25):

\[
\Delta P_{i} = f \frac{4L_{i}G^{2}}{d_{h}^{2} \rho_{i}}
\]

(25)

where \( L_{i} \) is the length of each sub-heat exchanger; \( G \) is the mass flow flux, \( \text{kg}/(\text{m}^{2} \cdot \text{s}) \); \( \rho_{i} \) is the density of the fluid, \( \text{kg}/\text{m}^{3} \).

The weight, efficiency, \( PWR \), and pressure loss of PCHE are calculated with Equations (26)–(29), separately:

\[
M = \pi \rho_{w} NLw_{c}^{2} 1 - \phi \frac{8\phi}{8}
\]

(26)

\[
\eta_{hx} = \frac{Q_{2}}{(m_{p})_{\text{min}}(T_{h,in} - T_{c,in})}
\]

(27)

\[
PWR = \frac{Q_{2}}{M}
\]

(28)

\[
P_{\text{loss}} = \frac{\Delta P}{P_{h}}
\]

(29)

where \( \rho_{w} \) is the density of metal wall, \( \text{kg}/\text{m}^{3} \); \( N \) is the number of channels; \( L \) is channel length, \( \text{m} \); \( \phi \) is porosity; \( \dot{m} \) is the mass flow rate of the fluid, \( \text{kg/s} \); \( P_{\text{loss}} \) is hot side pressure loss of PCHE, \( \% \); \( \Delta P \) is the hot side pressure drop of PCHE, \( \text{MPa} \); \( P_{h} \) is hot side inlet pressure of PCHE, \( \text{MPa} \).

The initial \( PWR \) is estimated to be 10 when the channel width is 1–2 mm. When the fluid inlet parameters on both sides of PCHE are determined, the scheme process of the SEG method is shown in Table 2 [24]:

Table 2. Scheme process of the SEG method.

| Step | Process |
|------|---------|
| 1    | Initialize parameters |
| 2    | Let \( i = 1 \) and assume \( P_{c,0} \) |
| 3    | Calculate the average temperature and pressure of the segment |
| 4    | Calculate the thermodynamic properties of \( \text{SCO}_{2} \) [41] and fuel [11] |
| 5    | Calculate \( Re_{i}, f_{i}, Nu_{i}, L_{i}, P_{c,i}, T_{c,i}, P_{h,i}, \) and \( T_{h,i} \) |
| 6    | Let \( i = i + 1 \). If \( i \leq n \), then return to Step 3, otherwise, proceed to the next step |
| 7    | Calculate \( P_{\text{loss}} \). If \( P_{\text{loss}} \geq 2\% \), then return to Step 2, otherwise proceed to the next step |
| 8    | Calculate \( T_{h,i}, T_{c,i}, L, M, PWR, \eta_{hx}, S_{g} \), and end the process |

(4) Fuel vapor turbine model

The specific power generation of the fuel vapor turbine can be calculated with Equation (30) [11]:

\[
\dot{w}_{f,m} = C_{p}(T_{f,in} - T_{f,out}) = \eta_{f}C_{p}T_{f,in} \left[ 1 - \pi_{f}^{(1-k)/k} \right]
\]

(30)
where $w_{p,m}$ is the specific power generation of fuel vapor turbine, kJ/kg; $C(\cdot)_p$ is the average specific heat of fuel, kJ/(kg·K); $\pi_T$ is pressure drop ratio of fuel vapor turbine; $k$ is the specific heat ratio of fuel.

Then, the total power generation of the fuel vapor turbine, $W_f$, can be calculated.

(5) Fuel weight penalty calculation

In this paper, the total mass of the take-off period method is employed [16–18]. The mass and power consumption of the PTMS can be converted into fuel consumption to evaluate their impact on the performance of the vehicle. Ignoring the weight of the fixed pipeline and power consumption of the fuel pump, the total fuel weight penalty of the PTMS is calculated with Equations (31) and (32) [16]:

$$M_{\text{total}} = M_{hx} + M_C + M_T + M_{fT} = \left( \exp \left( \frac{C_e \tau_0 g}{K} \right) - 1 \right) \times (m_{hx} + m_C + m_T + m_{fT}) \quad (31)$$

$$C_e = \frac{TSFC}{g} \quad (32)$$

where $M_{\text{total}}$ is the total fuel weight penalty of the PTMS, kg; $M_{hx}, M_C, M_T$, and $M_{fT}$ are the fuel weight penalty values of PCHE, compressor, turbine in the SCO2 cycle, and fuel vapor turbine, kg; $m$ indicates weight, kg; $C_e$ is the specific fuel consumption, kg/(N·s); $\tau_0$ is the endurance time, s; $g$ is gravitational acceleration, m/s²; $K$ is the lift–drag ratio of the aircraft, and is 2.95 at 6 Mach [42]; TSFC is the thrust-specific fuel consumption, s⁻¹, and is 0.001 at 6 Mach [43].

(6) System scheme design

The system scheme design process is shown in Figure 5. Firstly, the initial parameters, $T_{c,in}, \eta_C, \eta_{S,T}, \eta_P, \eta_{hx}, \eta_{cc}, \xi, \tau_F$, are given. Based on the criterion of $\Delta T_{\text{min}} \geq 10$ K, the compressor inlet temperature, $T_{S,1}$, is assumed. Then, the parameters of the SCO2 cycle and fuel can be obtained under the constraint of $T_{c,0} < T_{S,4}$. Finally, the optimized parameters are obtained.

**Figure 5.** System scheme design process.
3.2. Model Validation

For the cooling channel model, the heat exchange capacity is assumed to be 1350 kW according to the Lander–Nixon diagram [32]. For the fuel vapor turbine model, its parameters to calculate power generation refer to the experimental results in Ref. [11].

The turbomachinery model is verified using published experimental data. The comparison between simulation and experimental results of impeller diameter is shown in Table 3. The experimental data for compressor and turbine are taken from Refs. [44,45], respectively. The differences are 2.7% and 4.7% respectively, and both are less than 5%.

Table 3. Verification of turbomachinery model.

| Component | Rotation Speed (rad/min) | \( m_0 \) (kg/s) | Inlet Temperature (K) | Inlet Pressure (MPa) | \( \pi_c/\pi_T \) | Impeller Diameter (mm) |
|-----------|--------------------------|------------------|-----------------------|---------------------|-----------------|-------------------------|
| Compressor | 75,000                   | 3.5              | 305                   | 7.7                 | 1.8             | 18.7                    |
| Turbine   | 45,000                   | 8                | 392                   | 13.5                | 1.8             | 73                      |

The heat exchanger model is verified using published experimental data [46]. Experimental parameters of PCHE are shown in Table 4. The comparisons of \( \eta_{hx} \) at different \( T_{h,0} \) between experiment and simulation results are shown in Table 5. Their differences are less than 3%.

Table 4. Experimental parameters of PCHE.

| Parameter     | \( L \) (mm) | \( w_c \) (mm) | \( d_c \) (mm) | \( t_p \) (mm) | \( t_f \) (mm) | \( T_{c,n} \) (K) | \( P_{c,n} \) (MPa) | \( P_{h,0} \) (MPa) | \( m_c/m_h \) (kg/s) | Number of Channels, \( N \) |
|---------------|--------------|----------------|----------------|----------------|---------------|------------------|-------------------|-------------------|------------------------|--------------------------|
| Value         | 150          | 0.4            | 0.225          | 0.48           | 0.2           | 295              | 3                 | 2.6               | 0.01                   | 2400                     |

Table 5. Verification of heat exchanger model.

| \( T_{h,0} \) (K) | \( \eta_{hx} \) (%) | Experiment | Simulation | Difference (%) |
|------------------|---------------------|------------|------------|----------------|
| 400              | 88.2                | 86.7       | 1.7        |
| 412              | 89.4                | 88.0       | 1.6        |
| 432              | 90.0                | 87.6       | 2.7        |
| 452              | 90.0                | 89.2       | 0.9        |

4. Results and Analysis

4.1. System-Level Optimization

It is assumed that the heat production of the scramjet is 1350 kW at Mach 6. The objective of the system-level optimization is to minimize the consumption of fuel heat sink and the fuel weight penalty. The preliminary optimization solution set is shown in Figure 6, in which the abscissa and the ordinate represent the consumption of the fuel heat sink and the fuel weight penalty, separately. In Figure 6, a point is highlighted with a triangle as an example point. It can be seen that \( Q_{hs} \) is less than the heat production of the scramjet (1350 kW). This is because SCO\(_2\) converts part of the high-temperature waste heat of the scramjet into electricity, and the fuel only needs to cool the residual heat.

However, the pinch point may be located inside the fuel–SCO\(_2\) heat exchanger, because the physical properties of SCO\(_2\) and fuel vary greatly with the change of temperature and pressure [12,24,47]. The internal temperature distribution of PCHEs in the solution set will be further analyzed using the SEG method to ensure the selected solutions meet \( \Delta T_{\min} \geq 10 \) °C.
Figure 6. Preliminary optimization solution set and the example point of PTMS.

Taking the example point in Figure 6 as an example, the inlet pressure, temperature, and mass flow rate of the PCHE hot side are 7.5 MPa, 773 °C, and 1.54 kg/s, respectively. The corresponding values on the cold side are 4 MPa, 50 °C, and 0.5 kg/s, respectively. The amount of heat exchange is 1231 kW. Based on these parameters, the variation curve of the water equivalent \((c_p \times \dot{m})\) with temperature is shown in Figure 7a. The square point represents the point with the lowest temperature difference. It can be seen that the temperature difference between \(\text{SCO}_2\) and fuel is the lowest near 377 °C. Figure 7b shows the temperature change curve in the \(i\)th \((i = 1 \sim 20)\) sub-heat exchanger with the SEG method. The arrow indicates the solution direction. The fluid temperature difference on both sides gradually becomes smaller, and it is less than 10 K after the 10th sub-heat exchanger. In the 11th sub-heat exchanger, the fluid temperatures on both sides are crossed, but this heat reversal is impossible. This illustrates that the heat exchange process cannot occur under the given conditions. Therefore, these unfeasible solutions should be removed from the preliminary optimization set.

Figure 7. Heat exchange process of example point. (a) Variation curve of water equivalent with the temperature. (b) Temperature variation curve of sub-heat exchanger.

In Figure 8, the selected solutions are marked in black squares to distinguish them from the preliminary solutions in the yellow circle. Circular points represent the infeasible solution, and square points represent the feasible solution.
Figure 9 shows the influence of hot side temperature and efficiency on the heat sink consumption of fuel. To ensure the heat exchange process occurring in PCHE, namely $\Delta T_{\text{min}} \geq 10 ^\circ C$, the hot side inlet temperature of the first sub-heat exchanger ($T_{h,0}$) cannot be too low. The green line in Figure 9a shows the boundary of $T_{h,0}$, which is 770–800 $^\circ C$. Similarly, the heat exchanger efficiency cannot be too high. The green line in Figure 9b shows the boundary of $\eta_{hx}$, which is 90–91%. $T_{h,0}$ and $\eta_{hx}$ are consistent with the experimental results [47].

Pareto points (A–F) and the Pareto front of the above feasible solutions are shown in Figure 10. The minimum consumption of fuel heat sink corresponds to the abscissa, and the minimum fuel weight penalty corresponds to the ordinate. The square points in black represent the non-dominated solutions, the triangle points in red represent the dominated solutions, and the solid line is the Pareto optimal front. A point is highlighted with a yellow circle as a reference system. The system parameters of Pareto points are shown in Table 6.

Figure 8. System-level optimization results after feasible selection with SEG method.

| Pareto Point | $\eta_{hx}$ (%) | $P_1$ (MPa) | $\pi_C$ | $m_h$ (Kg/s) | $P_h$ (MPa) | $T_{h,0}$ (°C) | $T_{C,0}$ (°C) | $Q_{hs}$ (kW) | $M_{total}$ (kg) |
|--------------|-----------------|-------------|---------|--------------|-------------|----------------|----------------|--------------|----------------|
| A            | 89.9            | 7.5         | 3.4     | 1.52         | 7.6         | 813.8          | 616.8          | 1243.6       | 130.3          |
| B            | 90.0            | 7.6         | 3.3     | 1.53         | 7.7         | 812.5          | 616.9          | 1243.8       | 130.2          |
| C            | 90.0            | 7.7         | 3.2     | 1.55         | 7.8         | 801.2          | 617.1          | 1244.6       | 130.0          |
| D            | 90.0            | 8.0         | 3.2     | 1.56         | 8.1         | 797.3          | 617.3          | 1246.5       | 129.9          |
| E            | 89.9            | 8.5         | 3.0     | 1.54         | 8.6         | 799.2          | 617.8          | 1249.8       | 129.7          |
| F            | 89.9            | 8.8         | 2.9     | 1.54         | 8.9         | 801.3          | 618.0          | 1251.6       | 129.6          |
Figure 10. System-level optimization results.

The following conclusions can be drawn from Figure 10 and Table 6:

1. Fuel weight penalty mainly depends on the weight of PCHE, compressor, and turbines. The weight of PCHE is much greater than that of other components. The initial PWR of PCHE is fixed at 10, so the weight of PCHE depends on the heat exchange capability. Therefore, in the direction shown by Trend 1 in Figure 10, the greater \( Q_{hs} \), the greater \( M_{total} \).

2. For Pareto points of A~F in the direction shown by Trend 2, there is a conclusion opposite to Trend 1. When \( Q_{hs} \) is the maximum value of 1251.6 kW, \( M_{total} \) is the minimum of 129.6 kg. When \( Q_{hs} \) is the minimum value of 1243.6 kW, \( M_{total} \) is the maximum of 130.3 kg. This is because the weight of turbomachinery decreases significantly with the increase in \( Q_{hs} \). \( Q_{hs} \) and \( M_{total} \) become two competitive optimization variables. When \( Q_{hs} \) decreases, \( M_{total} \) increases accordingly, and vice versa. The Pareto points are located in the regions close to the coordinate axes.

3. The slope of the Pareto front changes significantly near point C (1244.6 kW, 130.0 kg). When \( Q_{hs} \) is less than 1244.6 kW, \( M_{total} \) increases significantly with the decrease in \( Q_{hs} \). When \( Q_{hs} \) is greater than 1244.6 kW, the change rate of \( M_{total} \) decreases with the increase in \( Q_{hs} \). Thus, point C can be considered as a compromise between \( M_{total} \) and \( Q_{hs} \).

The relationships between optimization variables and objectives are shown in Figure 11. The following conclusions can be drawn from Figure 11 and Table 6:

1. On the whole, increasing \( \pi_C \) or reducing \( P_1 \) not only reduces \( Q_{hs} \), but also increases \( M_{total} \). When \( \eta_{hx} \) is less than 91%, increasing \( \eta_{hx} \) can reduce \( Q_{hs} \) and \( M_{total} \) at the same time. Pareto points of A~F have approximately the same \( \eta_{hx} \).

2. \( T_{c,n} \) and \( \dot{m}_c \) are constants, so \( Q_{hs} \) depends on \( P_h \), \( T_{h,0} \), \( m_h \), and \( \eta_{hx} \) from Section 3. Among the above factors, only \( P_h \) and \( T_{h,0} \) change significantly, as shown in Table 6. However, \( Q_{hs} \), decreases with the increase in \( T_h \), and the decrease in \( P_h \), which indicates that \( P_h \), is the main influencing factor of \( Q_{hs} \).

3. For Pareto points, \( M_{total} \) mainly depends on the weight of turbomachinery. Increasing \( \pi_C \) or decreasing \( P_1 \) leads to a decrease in \( H_{sat} \), which in turn leads to an increase in the weight of the turbomachinery from Equations (10)–(14).

4.2. Component-Level Optimization

The component-level optimization is carried out based on the optimal parameters of point C. The lightest weight of PCHE and the minimum entropy production are the optimization objectives. The optimization results are shown in Figure 12. C1~C8 are Pareto points, and the curve is the Pareto front. Entropy production includes heat transfer entropy production (\( S_{gT} \)) and pressure entropy production (\( S_{gp} \)), which affect the efficiency and pressure loss of PCHE, separately. The relationship curve between \( S_{gT} \) and \( S_{gp} \) is shown in Figure 13. The detailed parameters of C1~C8 are shown in Table 7.
Figure 11. Relationships between optimization variables and optimization objectives. (a) $\pi_C$ and $Q_{hs}$ (b) $\pi_C$ and $M_{total}$ (c) $P_1$ and $Q_{hs}$ (d) $P_1$ and $M_{total}$ (e) $\eta_{hx}$ and $Q_{hs}$ (f) $\eta_{hx}$ and $M_{total}$.

Figure 12. Component-level optimization results.
As can be seen from Figure 12, the weight of PCHE decreases from 57.2 kg to 22.2 kg, and $S_g$ increases from 46.3 J/kg·K to 50.4 J/kg·K. Pareto optimal solutions compete with each other, which proves that the weight is reduced at the cost of increasing irreversibility in PCHE. C1–C3 are almost perpendicular to the ordinate, reflecting significant weight reduction. C5–C8 are almost perpendicular to the abscissa, reflecting that the significant increase in $S_g$. $L$ of C5 is more than 1 m, so C3 and C4 can be considered as better choices.

As can be seen from Figure 13, $S_{ST}$ decreases only slightly, so the efficiency of PCHE is almost constant and stable at 90%. With the increase in $S_{SP}$, $P_{loss}$ increases from 0.14% to 0.49%. The pressure losses of C3 and C4 are less than 1% and meet the design requirements [20]. C4 is selected, considering its lighter weight and smaller volume. Thus, the optimal efficiency (89.84%) and PWR (38.0) of PCHE are obtained, and the optimal total fuel weight penalty can be calculated as 48.6 kg. According to Equations (2)–(14), the weight and dimensional design of the compressor, turbine in the SCO$_2$ cycle and fuel vapor turbine are calculated as shown in Table 8.

Table 8. Weight and dimensional design of turbomachinery.

| Parameter                  | Compressor | Turbine in SCO$_2$ Cycle | Fuel Vapor Turbine |
|----------------------------|------------|--------------------------|--------------------|
| $N_s$                      | 0.4        | 0.51                     | 0.4                |
| Rotation speed in optimal working condition (rad/min) | 10,778     | 10,778                   | 5761               |
| Impeller diameter (mm)     | 226        | 173                      | 197                |
| Weight (kg)                | 9.3        | 5.4                      | 7.1                |

The fuel weight penalty of each component is shown in Figure 14. The compressor and turbine are compact components, and the maximum weight is no more than 10 kg, which is consistent with Ref. [34]. The total fuel weight penalty mainly depends on the weight of PCHE, which is greater than the sum of other components.
To compare the optimization effect, the reference system and optimization system of C and C4 are used to compare the parameters before and after the two-level optimization. The comparison of optimization results is shown in Figure 15.

It can be seen from Figure 15 that the system-level optimization mainly reduces the heat sink consumption of fuel, and the component-level optimization mainly reduces the fuel weight penalty. The system-level optimization can reduce the heat sink consumption of fuel by 20.2 kW and the fuel weight penalty by 1.9 kg. Based on the optimal solution in the system-level optimization, the weight of PCHE can be reduced from 124.5 kg to 30.9 kg in the component-level optimization. Accordingly, the total fuel weight penalty can be further reduced by 83.3 kg.

5. Conclusions

In this study, a two-level optimization method for the PTMS is proposed by NSGA-II for hypersonic vehicles at Mach 6. The PTMS is based on a SCO2 closed Brayton cycle with finite chemical heat sink of hydrocarbon fuel. The following conclusions can be obtained:

(1) The system-level optimization can obtain the preliminary solution set. To ensure the feasibility of heat exchanger design, the SEG method is employed to analyze the detailed heat transfer process in PCHE. The minimum temperature difference of PCHE is limited to 10 °C, and the unfeasible solutions are removed. The minimum $T_{h,0}$ is 770–800 °C, and the maximum $\eta_{hx}$ is 90% in feasible solutions.

(2) The turbomachinery adopts compact components, and their optimal design parameters are given. The results of the component-level optimization show that the weight of PCHE is greater than the sum of other components, and the total fuel weight penalty mainly depends on the weight of PCHE.

(3) The proposed two-level optimization method gives the optimal system parameters and the optimal size of key components. It can reduce the heat sink consumption of fuel by 20.2 kW and the fuel weight penalty by 85.2 kg compared to the reference system.
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Abbreviations

| Symbol | Denotation |
|--------|------------|
| $\pi_C$ | Compressor pressure ratio |
| $\pi_{fT}$ | Pressure drop ratio of fuel vapor turbine |
| $\pi_T$ | Pressure drop ratio of turbine in $\text{SCO}_2$ cycle |
| $C$ | The efficiency of compressor, % |
| $G$ | The efficiency of the generator, % |
| $P$ | The efficiency of the fuel pump, % |
| $S_{T2}$ | The efficiency of turbine in $\text{SCO}_2$ cycle, % |
| $\eta_{hx}$ | Heat exchanger efficiency, % |
| $\eta_{fT}$ | The efficiency of fuel vapor turbine, % |
| $cc$ | Relative pressure loss coefficient in cooling channels, % |
| $hx$ | Relative pressure loss coefficient of PCHE, % |
| $hx,s$ | Relative pressure loss coefficient of PCHE in system-level optimization, % |
| $\dot{m}_c$ | Fuel mass flow rate, kg/s |
| $\dot{m}_{h_{SCO}}$ | $\text{SCO}_2$ mass flow rate, kg/s |
| $A_s$ | Cross area, m$^2$ |
| $C_f$ | The specific fuel consumption, kg/(N·s) |
| core | Heat exchanger core |
| $d_c$ | Channel depth, mm |
| $d_h$ | Hydraulic diameter, mm |
| $G$ | Mass flow flux, kg/(m$^2$·s) |
| $h_{S2,2}$ | Compressor outlet-specific enthalpy, kJ/kg |
| $h_{S3,3}$ | Turbine inlet-specific enthalpy, kJ/kg |
| $h_{S2,2}$ | Ideal outlet-specific enthalpy of the compressor, kJ/kg |
| $h_{S2,4a}$ | Ideal outlet-specific enthalpy of the turbine, kJ/kg |
| $H$ | Core height of PCHE, m |
| $K$ | Lift–drag ratio |
| $L$ | The channel length of PCHE, m |
| $L_i$ | The channel length of each subheat exchanger, m |
| $M$ | Weight of PCHE, kg |
| $M_{total}$ | Total fuel weight penalty, kg |
| $N_s$ | Specific speed |
| $P_1$ | Compressor inlet pressure |
| POF | Pareto front |
| $P_h$ | Hot side inlet pressure of PCHE, MPa |
| $P_{loss}$ | Pressure loss of PCHE, % |
| $P_{p,in}$ | Inlet pressure of the fuel pump, MPa |
| $P_{p,out}$ | Outlet pressure of the fuel pump, MPa |
| $P_{S1,1}$ | Compressor inlet pressure, MPa |
| $P_{S2,2}$ | Compressor outlet pressure, MPa |
| $P_{S3,3}$ | Cooling channel outlet pressure, MPa |
| $P_{S4,4}$ | Turbine outlet pressure, MPa |
| PWR | Power-to-Weight Ratio of PCHE |
\begin{align*}
Q_{hs} & \text{ Heat sink consumption of fuel, kW} \\
Q_{\text{total}} & \text{ Engine heat production, kW} \\
S_g & \text{ Total entropy production, J/kg-K} \\
S_{\text{p}} & \text{ Pressure entropy production, J/kg-K} \\
S_{\text{ST}} & \text{ Heat transfer entropy production, J/kg K} \\
t_f & \text{ Fin thickness, mm} \\
t_p & \text{ Plate width, mm} \\
t_{sw} & \text{ Wall thickness, mm} \\
T_{c,0} & \text{ The outlet temperature of the cold side of PCHE, °C} \\
T_{c,1} & \text{ The inlet temperature of the cold side of PCHE, °C} \\
T_{h,0} & \text{ The hot side inlet temperature of the first sub-heat exchanger, °C} \\
T_{\text{max}} & \text{ The maximum temperature of the SCO\textsubscript{2} cycle, °C} \\
T_{s,1} & \text{ Compressor inlet temperature} \\
T_{s,4} & \text{ Turbine outlet temperature} \\
TSFC & \text{ Thrust-specific fuel consumption, s\textsuperscript{-1}} \\
w_c & \text{ Channel width, mm} \\
W & \text{ Core width of PCHE, m} \\
\Delta T_{\text{min}} & \text{ Pinch temperature difference}
\end{align*}

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