CHARACTERISTICS AND OPTIMIZATION OF SCO_2 BRAYTON CYCLE SYSTEM FOR HIGH POWER SODIUM-COOLED FAST REACTOR ON MARS

by

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Mars is the target of deep space exploration. The first problem of landing on Mars and building a satellite base is the energy source. For more than 50 kW high power demand, space nuclear energy system has the advantages of high output power, large energy density, small area, short working time and so on. Super-critical CO_2 Brayton cycle with sodium cooled fast reactor is the most promising power solution because of the high energy conversion efficiency. The thermodynamic model of super-critical CO_2 Brayton cycle system with sodium cooled fast reactor as the heat source has been established. After the analysis of circulation process, the relationship between temperature, pressure and enthalpy at working point has been discussed, and the relationship of circulation efficiency has been deduced. The real gas model is used to correct the thermophysical properties of super-critical CO_2. The thermal efficiency of the system is analyzed under the typical working condition of Mars surface. What's more, the effects of pressure ratio, compressor inlet temperature, turbine inlet temperature, and the temperature ratio on the cycle efficiency are discussed to get the optimal cycle characteristic and condition parameters.

Key words: *Mars surface, nuclear power, super-critical CO_2 Brayton system, cycle characteristics, optimization*

Introduction

With the development of society and economy, humankind's demand for energy is increasing day by day. Nuclear energy has received more and more attention as a clean, safe and efficient energy source. Space nuclear power is mainly divided into space nuclear reactor power source and radioactive isotope power source [1].

Mars, as a close neighbor of the Earth, has always been the preferred target for deep space exploration and development. Landing on Mars, establishing a base on Mars, and developing and utilizing Mars resources must solve the power supply problem firstly [2]. For electric power requirements of about 50 kW or above it, space nuclear reactor power usually has higher specific power than solar cell-battery power supplies [3]. In addition, it can meet the demand of large power and long-time power supply in the absence of sunlight [4]. The space nuclear power system can operate stably under extreme environments, with high output power, high energy density, small specific area, long working hours, and autonomy and reliability of the system [5].

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For six types of the Generation-IV Reactors (gas-cooled fast reactor, lead-cooled fast reactor, molten salt reactor, molten salt reactor, sodium-cooled fast reactor (SFR), very high temperature reactor, super-critical water-cooled reactor), SFR has been identified as the most matured and hence the most suitable for near-term demonstration [6-8]. Many previous SFR use a steam Rankine cycle as a power conversion system. However, the problem of sodium water reaction has been one of the main design issues of SFR [9]. At the same time, nuclear power systems require smaller size and lighter quality than ordinary ground power systems, which requires higher levels of conversion efficiency at the same power level [10]. Super-critical CO_2 (SCO_2) shows excellent characteristics in terms of physics and thermal engineering*, etc.* Use of closed Brayton cycle has the potential to simplify the design, reduce technical risk, reduce the amount and size of equipment and improve efficiency [11]. The SFR has received great attention in several major countries and has also been applied to planetary surface base reactors [12].

In this paper, the SCO_2 Brayton cycle is used as the thermoelectric conversion system. After establishing the circulation model, the temperature and enthalpy values of each working point are analyzed, and the expression of the cycle efficiency is deduced. The SCO_2 thermodynamic properties are corrected based on the actual gas state equation. The cycle efficiency is calculated and the influence of the influencing factors is analyzed and optimized.

The establishment of Brayton cycle model

Based on a simple Brayton cycle model, the overall model shown in fig. 1. The core of SFR is the heat source of closed Brayton cycle. The exterior is a radiator, and the large radiator panel is cooled by heat radiation. Analyze the thermodynamic processes in each process, establish equations based on the Second law of thermodynamics and the relationship between temperature and enthalpy at each operating point. The temperature and entropy diagram of the cycle are shown in fig. 2.

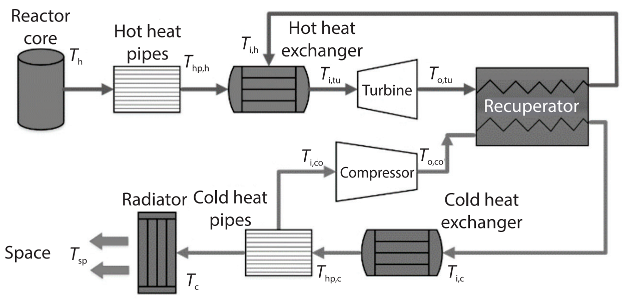

Figure 1. The SCO_2 Brayton cycle model diagram

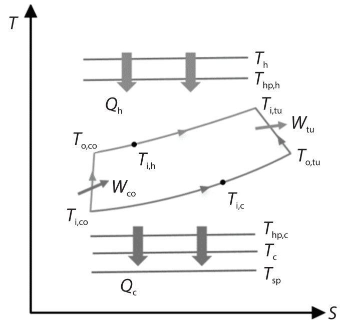

Figure 2. Temperature-entropy diagram of the cycle

The heat in the core is transferred to the heat pipes through convective heat transfer, and the heat transfer to the cold heat pipes and radiator through convection heat transfer:

$$Q_h = UA_{hp,h}\left(T_h - T_{hp,h}\right),\; Q_c = UA_{hp,c}\left(T_{hp,c} - T_c\right) \tag{1}$$

where Q_h is the core heat power, $UA_{hp,h}$ – the thermal conductivity of hot heat pipes at unit temperature, T_h – the core temperature, $T_{hp,h}$ – the hot heat-pipe temperature, Q_c – the cooling power, $UA_{hp,c}$ – the thermal conductivity of cold heat pipes at unit temperature, $T_{hp,c}$ – the cold heat pipe temperature, and T_c – the radiator temperature.

The heat transfer process between the heat pipes and the heat exchanger can be given [13]:

$$Q_\mathrm{h} = \dot{m}c_{p,\mathrm{h}}\left(T_\mathrm{hp,h} - T_\mathrm{i,h}\right)\left(1 - \mathrm{e}^{UA_\mathrm{h}/\dot{m}c_{p,\mathrm{h}}}\right),\; Q_\mathrm{c} = \dot{m}c_{p,\mathrm{c}}\left(T_\mathrm{i,c} - h_\mathrm{hp,c}\right)\left(1 - \mathrm{e}^{UA_\mathrm{c}/\dot{m}c_{p,\mathrm{c}}}\right) \tag{2}$$

where $\dot{m}$ is the mass-flow, $c_{p,\mathrm{h}}$ – the specific heat at constant pressure of hot heat exchanger, $T_\mathrm{i,h}$ – the of hot heat exchanger temperature, UA_h – the thermal conductivity of the hot heat exchanger at unit temperature, $c_{p,\mathrm{c}}$ – the specific heat at constant pressure of cold heat exchanger, $T_\mathrm{i,c}$– the cold heat exchanger temperature, and UA_c – the thermal conductivity of the cold heat exchanger at unit temperature.

According to the Second law of thermodynamics, we have:

$$Q_\mathrm{h} = \dot{m}\left(h_\mathrm{i,tu} - h_\mathrm{i,h}\right),\; Q_\mathrm{c} = \dot{m}\left(h_\mathrm{i,c} - h_\mathrm{i,co}\right) \tag{3}$$

where $h_\mathrm{i,\,tu}$ is the enthalpy of turbine, $h_\mathrm{i,\,h}$ – the enthalpy of hot heat exchanger unit, $h_\mathrm{i,\,c}$ – the enthalpy of cold heat exchanger unit, and $h_\mathrm{i,\,co}$ – the enthalpy of compressor.

There are:

$$\frac{T_\mathrm{i,tu}}{T_\mathrm{o,tu}} = \pi^{\left(\frac{\gamma_\mathrm{tu}-1}{\gamma_\mathrm{tu}}\right)\eta_\mathrm{tu}},\; \frac{T_\mathrm{o,co}}{T_\mathrm{i,co}} = \pi^{\left(\frac{\gamma_\mathrm{co}-1}{\gamma_\mathrm{co}}\right)\eta_\mathrm{co}} \tag{4}$$

where $T_\mathrm{o,tu}$ is the turbine outlet temperature, γ_tu – the isentropic process polytropic index of turbine, η_tu – the turbine work efficiency, $T_\mathrm{o,co}$ – the compressor outlet temperature, $T_\mathrm{i,co}$ – the compressor inlet temperature, γ_co – the isentropic process polytropic index of compressor, and η_co – the compressor work efficiency.

The cycle efficiency, denoted as η_sys, reads:

$$\eta_\mathrm{sys} = \frac{\eta_\mathrm{a}\left[\dot{m}\left(h_\mathrm{i,tu} - h_\mathrm{o,tu}\right) - \dot{m}\left(h_\mathrm{o,co} - h_\mathrm{i,co}\right)\right]}{Q_\mathrm{h}} \tag{5}$$

where η_a is the actual work efficiency.

Radiator transfer sheat to space through radiation:

$$Q_\mathrm{c} = \sigma\varepsilon_\mathrm{r}A\eta_\mathrm{f}\left(T_\mathrm{c}^{\,4} - T_\mathrm{sp}^{\,4}\right) \tag{6}$$

where σ is the black body radiation constant, take $5.67 \cdot 10^{-8}$ W/m^{2}K^4, η_f – the fin efficiency, ε_r – the radiator efficiency, A – the area of fin, T_c – the temperature of cold heat exchanger unit, and T_sp – the temperature of space.

Calculation of cycle efficiency

Based on the actual gas-state equations, the Redlich-Kwong (RK) equation of state is well fitted in the super-critical state, the error of the Soave RK (SRK) equation and Peng-Robinson (PR equations) is large, the error of the RK equation that modify factor are finded by experiment (EXP-RK) and PR equation that modify factor are finded by experiment (EXP-PR) is small, but the equation is more complex than the RK equation [14]. This article uses the RK equation as the equation of state for CO_2:

$$p = \frac{\mathrm{R_g}T}{v-b} - \frac{a}{T^{0.5}v(v+b)} \tag{7}$$

where P is the pressure of CO_2, $\mathrm{R_g}$ – the gas constant, take 8.314 J/molK, v – the molar volume of CO_2, b – the constant to modify the volume of gas, T – the temperature of CO_2, and a – the constant to modify the inter-molecular attraction.

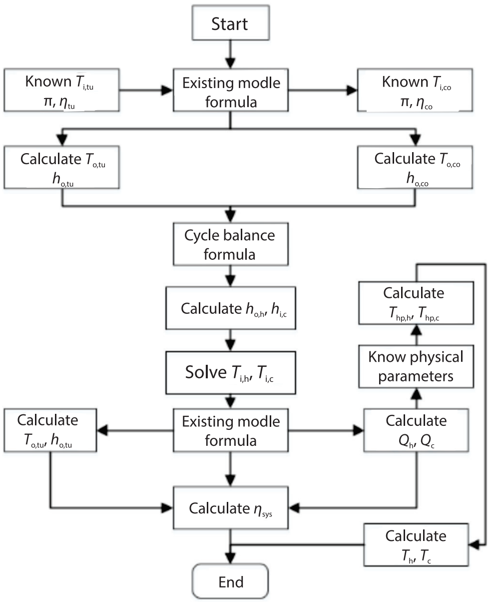

Figure 3. The calculation flow chart of the model

According to the property of SCO_2, set the compressor inlet temperature to 304 K, the pressure to 7. 4MPa, the specific volume of 0.0022 m^3/kg, set the turbine inlet temperature to 912 K. Based on the given working parameters and the process temperature-entropy map, the preliminary calculations can be determined as shown in fig. 3.

Cycle efficiency analysis

In the initial working condition, the pressure ratio, π, is set to 2. The turbine and compressor inlet temperatures are not changed, the value of π is gradually changed. The influence of pressure ratio, π, on efficiency is shown in fig. 4. From the aforementioned analysis, it can be seen that in the range of π about 1.5~3.4, the efficiency increases gradually with the increase of π. The compressor inlet temperature is 304 K and the turbine inlet temperature is 912 K. The pressure ratio of 2.7~3.0 can achieve higher efficiency.

Without changing the turbine inlet temperature and π, the values of $T_{i,co}$ are gradually changed, and the influence of compressor inlet temperature $T_{i,co}$ on efficiency is shown in fig. 5. As $T_{i,co}$ increases and the inlet temperature of the turbine remains constant, the efficiency decreases approximately linearly. This is because the density of CO_2 is higher at the critical temperature point, the density decreases when it deviates from the critical point, the power consumption of the compressor increases, and the cycle thermal efficiency decreases. In order to prevent phase change, the compressor inlet temperature must not be lower than the critical temperature of CO_2, 304 K. When $T_{i,co}$ is 304 K, the cycle efficiency is the highest. In order to achieve higher efficiency, $T_{i,co}$ can be set between 304~310 K.

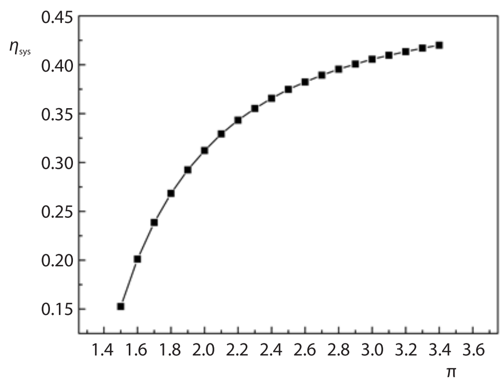

Figure 4. Influence of π on efficiency

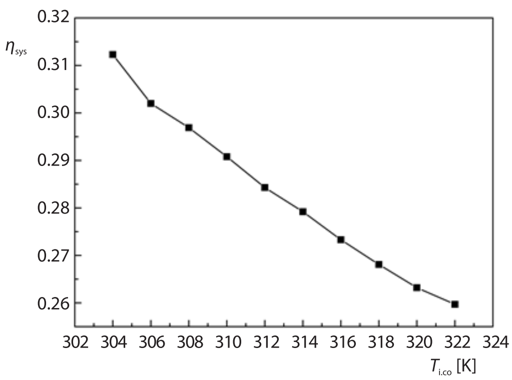

Figure 5. Influence of $T_{i,co}$ on efficiency

Turbine inlet temperature is set to 912 K in the initial conditions. The $T_{i,co}$ and π are not changed, and the $T_{i,tu}$ values is changed gradually. The influence of turbine inlet temperature

$T_{i,tu}$ on efficiency is shown in and fig. 6. There is a temperature ratio, τ, which is the ratio of $T_{i,tu}$ to $T_{i,co}$ also makes adifference on cycle efficiency. In the initial conditions, $T_{i,co}$ is set to 304 K, $T_{i,tu}$ is set to 912 K, $\tau = 3$, the value of τ is gradually changed. The influence of temperature ratio τ on efficiency is shown in fig. 7

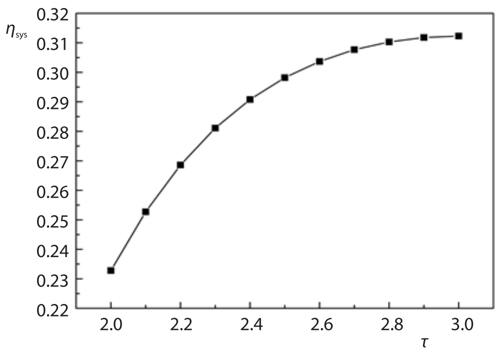

Figure 6. Influence of τ on efficiency

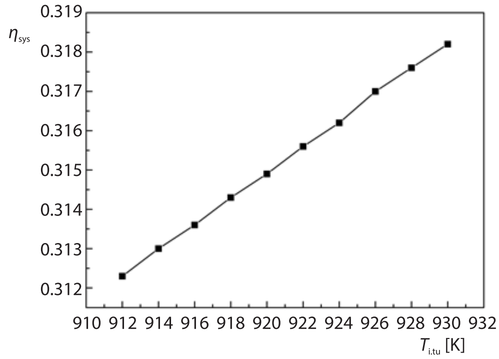

Figure 7. Influence of $T_{i,tu}$ on efficiency

When $T_{i,tu}$ increases and $T_{i,co}$ does not change, the total heat transfer during the reheating process increases, the power of the turbine increases, and the efficiency increases approximately linearly with the inlet temperature of the turbine. Under the condition that π is constant, when τ is 2.0~3.0, the endothermic average temperature increases and the average exothermic temperature does not change with the increase of τ. What's more, the efficiency also increases, but the efficiency increase gradually decreases. After τ = 2. 6, the efficiency increase tends to be flat, and τ should be reasonably selected according to the constraints of the material. Under the setting conditions, it is more appropriate to choose τ of 2.6~3.0.

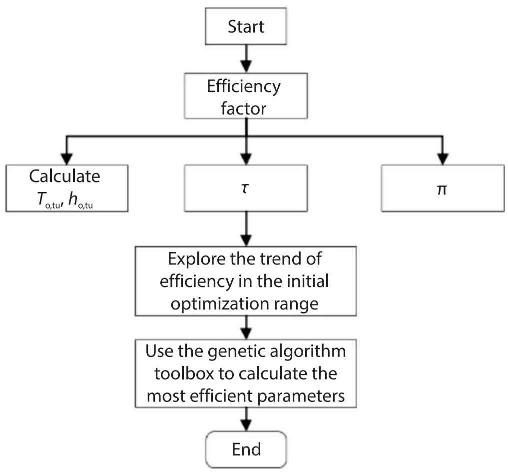

Figure 8. The efficiency optimization flow chart based on genetic algorithm

System optimization based on genetic algorithm

The genetic algorithm is based on evolutionary theory, which simulates the global random search and optimization method of natural biological evolution mechanism, fig. 8. Using the MATLAB genetic algorithm toolbox, the optimal solution can be obtained in a short time.

The efficiency optimization is abstracted as a mathematical expression:

$$\begin{gathered} f(\pi, T_{i,co}, \tau) = \max \eta_{sys}(\pi, T_{i,co}, \tau) \\ 2.7 \le \pi \le 3.3 \\ 304\ \text{K} \le T_{i,co} \le 310\ \text{K} \\ 2.6 \le \tau \le 3.0 \end{gathered} \tag{8}$$

Basedon the existing research, it is found that in the case of $\pi = 2$, $T_{i,tu} = 912$ K, $T_{i,co} = 304$ K, the efficiency reaches the maximum value. The efficiency varies with $T_{i,co}$ at different pressure ratios are shown in fig. 9. The efficiency varies with $T_{i,co}$ atdifferent temperature ratios are shown in fig. 10. When π is 2.7~3.0, the cycle efficiency reaches a maximum when $T_{i,co}$ is 304 K. When τ is 2.6~3.0, the cycle efficiency reaches the maximum when $T_{i,co}$ is 304 K. When π is 2.7~3.0 and τ is 2.6~3.0, the compressor inlet temperature at the maximum cycle efficiency is 304 K, which can be directly determined.

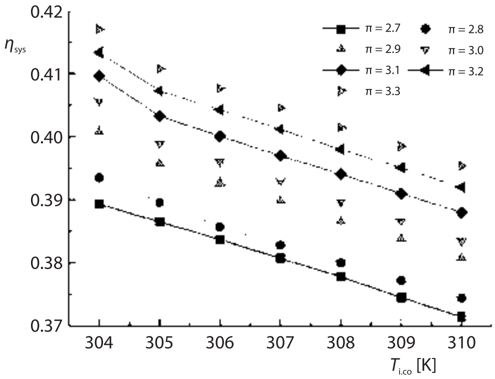

Figure 9. The efficiency varies with $T_{i,co}$ at different pressure ratios

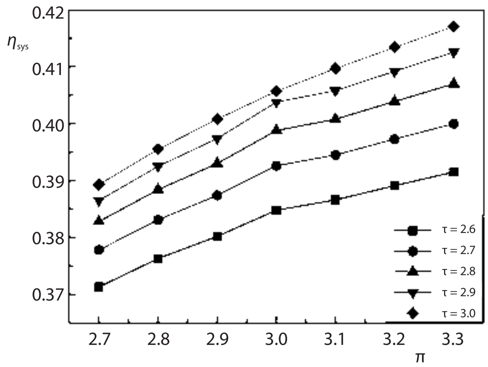

Figure 10. The efficiency varies with $T_{i,co}$ at different temperature ratios

The problem is further simplified, mainly to discuss the effect of τ and π on cycle efficiency. The efficiency varies with τ at different pressure ratios are shown in fig. 11. The efficiency varies with π at different temperature ratios are shown in fig. 12.

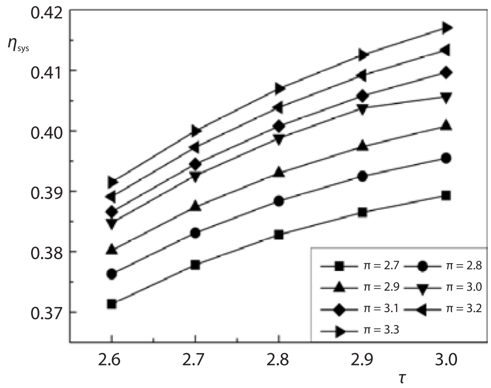

Figure 11. The efficiency varies with τ at different pressure ratios

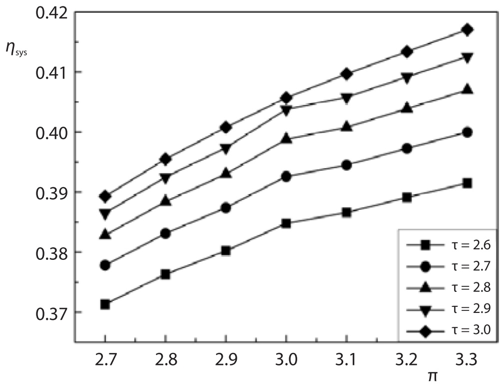

Figure 12. The efficiency varies with π at different temperature ratios

When π is between 2.7 and 3.0, the cycle efficiency increases gradually with the increase of τ, but the increase gradually decreases. Using the genetic algorithm toolbox, when π is 2.0, the efficiency is at a maximum of 0.3126 when τ is 3.002. When π is 3.0, the efficiency reaches a maximum value of 0.4061 when τ is 3.005. The efficiency reaches a maximum when τ is approximately around 3.0. According to the constraint, τ is chosen to be 3.0 and $T_{i,tu}$ is 912 K.

In the case where $T_{i,co}$ = 304 K, τ = 3.0 has been determined, the efficiency varies with π is shown in fig. 13.

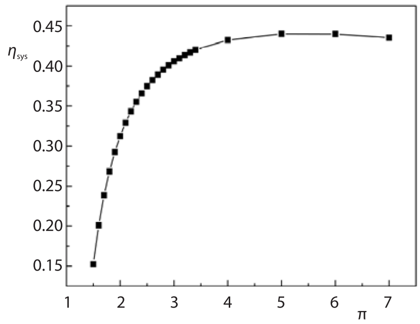

Figure 13. The efficiency varies with π

The data is fitted to obtain the relationship between efficiency and pressure ratio as follows. Using the genetic algorithm toolbox, the iteration is around 50 generations, and the efficiency reaches a maximum value of 0.4517 when π is 5. 666. However, the turbine inlet pressure is as high as 42 MPa, considering the constraint, π is 3.0. Then, the effectiveness η_{sys} reads:

$$\eta_{sys} = 0.00024852\pi^7 - 0.0074126\pi^6 + 0.092584\pi^5 - \\ -0.62873\pi^4 + 2.5163\pi^3 - 5.9795\pi^2 + 7.9307\pi - 4.2209 \tag{9}$$

Conclusion

This paper chooses SFR as the heat source, SCO_2 as the coolant, and Brayton cycle as the thermoelectric conversion method. Analyze the specific influence process of the influence factors of the cycle on the efficiency to get the optimized working conditions. Increasing the pressure ratio within a certain range can increase the cycle efficiency, but taking into account the limitations of the material and the effect of excessively high pressure on the regenerator, the suitable range for the pressure ratio is 2.7~3.0. Reducing the compressor inlet temperature and increasing the turbine inlet temperature can improve the cycle efficiency. With the increase of the temperature ratio, the cycle efficiency also increases, but the increase rate gradually decreases, and the suitable range of temperature ratio is 2.6~3.0. After further optimizing the efficiency by genetic algorithm, the optimized working condition can be obtained as: compressor inlet temperature is 304 K, turbine inlet temperature is 912 K, temperature ratio is 3.0, and pressure ratio is 3.0. The optimized system efficiency is 0.4057, which is nearly 30% higher than the previous efficiency of 0.3123.

Acknowledgment

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Nomenclature

A – radiator area, [m^2]
$\dot{m}$ – mass-fiow, [kgs^{-1}]
Q_c – thermal power, [kW]
Q_h – core heat power, [kW]
T_c – radiator temperature, [K]
T_h – core temperature, [K]
$T_{hp,c}$ – cold heat pipe temperature, [K]
$T_{hp,h}$ – hot heat pipe temperature, [K]
$T_{i,c}$ – cold heat exchanger temperature, [K]
$T_{i,co}$ – compressor inlet temperature, [K]
$T_{i,h}$ – hot heat exchanger temperature, [K]
$T_{i,tu}$ – turbine inlet temperature, [K]
$T_{o,co}$ – compressor outlet temperature, [K]
$T_{o,tu}$ – turbine outlet temperature, [K]
T_{sp} – temperature of space
UA – thermal conduuctivity, [kW(mK)$^{-1}$]
v – compressor inlet volume, [m/s]
W_{co} – compressor power, [W]
W_{tu} – turbine power, [W]

Greek symbols

ε_r – radiator performance, [–]
η_a – actual work efficiency, [–]
η_{co} – compressor work efficiency, [–]
η_{sys} – effectiveness, [–]
η_{tu} – turbine work efficiency, [–]
π – pressure ratio/compression ratio, [–]

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