INTRODUCTION

As the shortage of traditional energy and increasingly serious environmental problems, the development of clean, pollution-free, and renewable energy is imminent. Also, solar energy has a broad prospect for utilization as it is inexhaustible and friendly to the environment. 1,2

Solar power generation mainly includes solar photovoltaic generation and solar thermal power generation. 3 Both technologies use the direct normal irradiation of the sun as a main source to generate electrical power. The solar photovoltaic generation 4 converts solar radiation to electrical energy directly by means of photovoltaic cells. However, the manufacturing processes of photovoltaic cells, which are made of semiconducting materials, run counter to the principle of environmental protection. By contrast, solar thermal power generation is getting more and more attention because of the less pollution during its product life cycle.

At present, there are two major categories of solar thermal power generation systems: concentrating and nonconcentrating. The concentrating solar thermal power (CSTP) system has attracted more attention owing to its high conversion efficiency. The International Energy Agency has set an electricity generation target of 630 GW for CSTP

Conceptual design of a new thermal-electric conversion device in lightweight concentrating solar thermal power system

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Abstract

Concentrating solar thermal power (CSTP) technology has become a hot issue in the present research because of its high conversion efficiency. However, traditional schemes cannot be widely applied due to their complex structure, low power density, and high cost. In view of this, based on a new type of lightweight cable mesh reflector structure and the Rankine cycle, a new conceptual design of thermal-electric conversion device in CSTP system is proposed. The process of thermal-electric conversion is actualized through smart and lightweight structural design; the performance estimation of the device is carried out through focusing performance analysis and thermodynamic analysis; the cycling feasibility is verified through multifield coupling simulation analysis. Compared with the existing thermal-electric conversion devices, the device proposed in this paper has higher power-to-weight ratio. And the power generation system that combines the proposed device with the flexible cable net reflector has the characteristics of lightweight, portability, and small land area, which has a great significance for the popularization and application of distributed layout CSTP system.

KEYWORDS
concentrating solar thermal power, cycling feasibility analysis, Rankine cycle, thermal-electric conversion, thermodynamic analysis
technology by 2050.\textsuperscript{5} According to the different kinds of structural form, it can be divided into three techniques: parabolic trough technology, solar tower technology, and dish technology.\textsuperscript{6,7}

The parabolic trough system\textsuperscript{8} consists of a series of linear parabolic reflectors that concentrate sunlight onto the receiver positioned along the reflector's focal line. The reflectors are produced from raw material such as aluminum or low-iron glass to lessen the absorption losses.\textsuperscript{9} After being processed into trough paraboloid, manufacturing processes, such as silver plating or protective coating, will be applied on the surface of the reflector to improve its reflectivity. The receiver usually consists of a glass sleeve tube and a metal tube. The glass sleeve tube is designed to reduce heat losses and protect the absorber. The metal tube, usually a stainless steel tube coated with a selective coating to increase the absorptivity of solar radiation and to decrease the emissivity on the infrared spectrum,\textsuperscript{10} transfers concentrated solar heat to the working fluid.

Different from the structure of parabolic trough system, the solar tower power plant\textsuperscript{11} is composed of a central tower with a receiver mounted on its top. The tower is built on the middle of a field of mirrors called heliostats. The heliostats redirect the solar radiation to the receiver by solar tracking system.

For the above two systems, the Rankine cycle is mostly adopted.\textsuperscript{12} It consists mainly of four components: an evaporator, which evaporates the working fluid by absorbing the concentrated solar flux; a turbine, in which the working fluid expands and generates power; a condenser, in which the fluid condenses at lower pressure; and a pump, which pressurizes the condensed fluid. These two techniques have been developed maturely and realized in large-scale and grid-connected commercial applications.\textsuperscript{13}

According to the way the working fluid absorbs heat, the Rankine cycle can be divided into two types: indirect and direct. For the indirect steam generation technology, the heat transfer fluid, thermal oil or heat transfer salt, is needed to absorb the solar heat firstly. Then, the heat transfer fluid would flow through heat exchanges, to heat the working fluid.\textsuperscript{14} For the direct steam generation (DSG) technology, the solar heat is directly transformed to the working fluid, water, after being absorbed by the metal tube. The generated steam can be either saturated or superheated and then is routed to the turbine.\textsuperscript{15} Compared to the indirect steam generation system, the DSG technology can eliminate the heat exchanges between the heat transfer fluid and the working fluid and simplify the structure of power generation system, ultimately reducing the system complexity and the cost. It is reported that the DSG technology can decrease the energy production costs in about 15\% when compared to thermal oil plants.\textsuperscript{16} Moreover, this technology can avoid the environmental risk of oil leakage in solar field, and the working fluid in DSG, water, is more available and cheaper.\textsuperscript{17} Now the DSG is one of the most attractive technologies in CSTP.\textsuperscript{18}

However, the above two techniques, parabolic trough and solar tower, both need the solar concentrator subsystem, heat transfer subsystem, steam turbine unit, working medium pump, cooling subsystem, and other subsystems on the ground to cooperate with each other, which lead to large land area and high construction cost.\textsuperscript{19} Even the trough system is reported to have a smaller land area in both systems, it still occupies a land area of 25 m\(^2\)/kW without heat storage.\textsuperscript{20}

A dish system has several key subcomponents, described here as the reflector, receiver, support structure, tracking system, foundations, and so on.\textsuperscript{21} The reflecting surface of parabolic reflector is made of aluminum or silver on glass or plastic.\textsuperscript{22} The receiver is placed at the focus of the reflector. Its working principle is as follows: The sunlight is concentrated on the heat receiver by reflector to convert solar energy into thermal energy, and then, the thermal-electric conversion process is realized through a heat engine. Many different power conversion cycles can be considered for the receiver, such as Rankine cycle, Stirling cycle, and Brayton cycle.\textsuperscript{23-25} Among them, the Stirling cycle, which is a closed thermodynamic cycle that consists of two isochoric and two isothermal processes, is widely concerned because of its advantages: external combustion, closed cycle, small size, high efficiency, less noise, and so on.\textsuperscript{26,27} According to the arrangement of compression and expansion chambers, the Stirling engine can be divided into three types: alpha, beta, and gamma. According to the way of heating working fluid, the Stirling engine can also be divided into two types: direct and indirect,\textsuperscript{28} which is similar to the Rankine cycle. Of course, the direct type is less complex and more suitable for the popularization of dish system. Because of its own structural characteristics, the dish system can be used not only for centralized layout power generation, but also for distributed layout power generation, which is an important advantage and has attracted wide attention and research.

However, the current Stirling engine and dish system have some shortcomings that limit their wider application. Low power-to-weight ratio is the main disadvantage of Stirling engines. For example, the weight of the SOLO Stirling 161 developed by SPS company is 460 kg with the maximum output power of 9.5 KW;\textsuperscript{29} the weight of a beta Stirling engine developed by Thorsen et al is 100 kg with the output power of 3KW;\textsuperscript{30} the weight of a gamma-type Stirling engine manufactured by EPAS GmbH is 43 kg with the maximum output power of 820 W;\textsuperscript{31} the weight of the Stirling S400 developed by EPAS company is 28 kg with the maximum output power of 400 W;\textsuperscript{32} and the weight of the Stirling engine developed by Microgen company is 49 kg with the maximum electrical output of 1050 W.\textsuperscript{33} Moreover, the regenerator in Stirling engine requires material that can obtain high temperature difference under high pressure, which leads to a high
cost. In addition, the solid paraboloidal concentrator also has the problem of heavy weight. A dish system of 500-m² aperture area was completed on the ANU campus, the total mass of dish is 19.1 tons, and the base and supports weighed 7.3 tons. ToughTrough has developed a steel and glass-faceted, polyurethane-cored sandwich panel, which has been used for heliostats in BioSolar-4SKA dish project, and the specific weight of the mirrors is about 10 kg/m², still high. Thus, due to the above problems, the application of dish system is limited to a certain extent.

In summary, the current CSTP systems are difficult to achieve a wide range of applications owing to their low power density, large land area, or high cost.

In recent years, the new type of lightweight cable mesh reflector structure has been paid more and more attention because of its advantages of lightweight, high storage ratio, wide range of applicable diameters, portability, and so on. It uses a fixed truss hoop and a central truss cylinder as the rigid backbone, among which are all flexible cables to form a mesh surface, forming a lightweight rigid-flexible combined structure. Additionally, there is a reflective membrane laid on the cable meshes. The surface density of the lightweight structure applied in space can be realized to be less than 0.28 kg/m². When the reflector structure is applied on the ground, the wind load and other factors need to be considered, which would cause weight gain, even so its surface density will still not exceed 1 kg/m². Moreover, due to the greatly low surface density, the movement of the lightweight reflector would be fairly easy to realize and to maintain by a simple structural scheme. So this new type of lightweight cable mesh reflector structure has an irreplaceable advantage to be applied in the field of distributed layout CSTP.

However, in this situation, due to the requirement on surface precision of the lightweight reflector, the structure and weight of the thermal-electric conversion device are limited to a certain extent. It is obviously inappropriate for existing Stirling engines to be used on such a lightweight reflector structure. Therefore, aiming at the new type of lightweight cable mesh reflector structure, a new conceptual design of thermal-electric conversion device based on the Rankine cycle is put forward in this paper.

Besides, the pump is the only electricity-consuming component in conventional Rankine cycle and needs to be driven by external force. To eliminate the mechanical booster pump, scholars have proposed various schemes, such as the thermosiphon Rankine engine, the gravity-driven Rankine cycle, the pumpless Rankine cycle, and the vapor-liquid ejector. However, the above schemes more or less have the shortcomings of requiring large room, needed to be cooled and heated alternately, two-phase mixing flow, and so on. Based on this, for the proposed thermal-electric conversion device, a new booster cavity is designed to replace the mechanical booster pump. For this proposed structure, the pressurization of low-pressure condensate liquid is achieved by utilizing the internal energy of high-temperature and high-pressure steam, with no need for electric drive. With the use of this structure, the whole cycle process of thermal-electric conversion device becomes independent from electricity, which makes the device more compact and portable.

Furthermore, the possibility of proposed device is verified. Compared with the existing thermal-electric conversion devices, it has higher power-to-weight ratio and lower cost, which has great significance for the popularization and application of distributed layout CSTP system.

The paper is organized as follows: Section 2 describes the conceptual design of such a thermal-electric conversion device. Section 3 gives performance estimation of the device, including focusing performance and thermodynamic performance. Section 4 shows the results obtained by the cycling feasibility analysis. Lastly, Section 5 outlines the conclusions and final remarks.

2 | STRUCTURE CONCEPTUAL DESIGN

2.1 | Scheme design

The CSTP system generally consists of three integral parts: thermal-electric conversion device, solar concentrator, and
solar tracker. For distributed layout CSTP system, it is undoubtedly a good choice to use lightweight cable mesh reflector structure as the solar concentrator, as shown in Figure 1. In addition, the solar tracking can be achieved by different smart structures in combination with tracking algorithm. For the two parts of the solar concentrator and the solar tracker, we have already done some research.\textsuperscript{42,43}

The thermal-electric conversion device is installed at the focus of the solar concentrator. Taking into account that the thermal-electric conversion device would cause deformation of the cable net reflector, and the precision of the cable net reflector would affect the performance of the thermal-electric conversion device in turn, the structure and weight of the thermal-electric conversion device must be carefully considered in the design process. Based on our invention patent,\textsuperscript{44} we propose a new design scheme of lightweight thermal-electric conversion device.

The main components of this device are evaporation cavity, expansion cavity, booster cavity, condensation cavity and generator, as illustrated in Figure 2. Among them, the four cavities are sequentially fixed and connected through pipelines, and the generator is connected with the scroll expander installed in expansion cavity.

2.2 Working principle

During working hours, the solar tracker can keep the solar concentrator always normal to the incoming sunlight according to a tracking algorithm, and the reflective membrane laid on the cable mesh reflector can reflect and concentrate the incident sunlight on the thermal-electric conversion device, which is installed at the focus of the reflector. The thermal-electric conversion device, using common water as the working medium, would achieve the thermal-electric conversion process based on a closed Rankine cycle, and finally the electrical energy is output through a generator.

Next, based on the major components of the thermal-electric conversion device, the working principle is explained in detail.

2.2.1 Evaporation cavity

A flat plate receiver is designed for the evaporation cavity, which is composed of a glass cover plate, an absorber plate, heat tubes, thermal insulation layer, and so on, as illustrated in Figure 3.

The glass cover plate is designed to reduce heat losses and protect the absorber plate. The absorber plate, which is coated with a solar selective absorbing coating to increase the absorptivity of solar radiation and to decrease the emissivity on the infrared spectrum, is installed in the focal plane of the solar concentrator. Furthermore, its size is determined by the focusing performance of the solar concentrator. Because of the sun tracking in working process, the receiver absorbs heat at different angles along with the reflector. In order to ensure that the liquid water can absorb heat as evenly as possible, six coils with the same structure are uniformly distributed on the surface of the absorber plate, and the inlets are evenly distributed around the circumference.

Under the radiation of the incident sunlight reflected by reflective membrane, the temperature of absorber plate and heat tubes increase, and the solar energy is converted into heat energy. Condensed water flows into the coils from the surrounding inlets to absorb heat, which would lead to

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**FIGURE 2** Schematic diagram of the thermal-electric conversion device

**FIGURE 3** Schematic diagram of the flat plate receiver
the increase of water temperature, the occurrence of phase change, and the generation of high-temperature and high-pressure steam. After that, the steam will be divided into two parts at the outlets, one part enters the expansion cavity and the other part enters the booster cavity.

2.2.2 Expansion cavity

Most part of the high-temperature and high-pressure steam discharged from evaporation cavity would directly enter the expansion cavity to do work. In this paper, the scroll expander is selected as the expander, which is suitable for the occasion of small-scale applications, and has the advantages of high reliability, compact structure, relatively low revolution speed, and low noise level.\textsuperscript{45}

The scroll expander consists of a pair of scrolls called fixed and orbiting scrolls, respectively, as shown in Figure 4A, which are assembled at a relative angle of 180°. It can be observed that a number of closed crescent-shaped volumes or pockets are formed. The expansion process includes three stages: charging, expansion, and discharging. Firstly, the steam flows into the cavity from the central inlet of the scroll expander. During the expansion process, the steam pushes the center of orbiting scroll to go around the center of the fixed scroll. Thereafter, the two symmetric crescent-shaped pockets are moved toward the scroll periphery as the orbiting scroll rotates, as shown in Figure 4B. Finally, there is output work delivered through the eccentric shaft. And then, the worked steam will be discharged through the outlet of expansion cavity and enter the condensation cavity to change phase and to release heat.

2.2.3 Booster cavity

The booster cavity mainly consists of three parts: a moving cavity, an outer cavity, and a compression spring, as shown in Figure 5. The lower part of the side wall of the moving cavity is equipped with a compressible component, which can reciprocate under the actions of external force and compression spring, while the upper part of the side wall is machined with a curved steam inlet and a linear steam outlet, as shown in Figure 5A. For the outer cavity, there is a protrusion at the lower part of the side wall, and a trajectory groove with varying depth is processed on the inner side of the protrusion, as shown in Figure 5B. In addition, a steam inlet, a steam outlet, a water inlet, and a water outlet are machined on the side wall of the outer cavity.

At the initial state, the steam inlet of the moving cavity is coincident with the steam inlet of the outer cavity, while the steam outlets of these two cavities are not coincident, as shown in Figure 5C, 5D, and there is some low-pressure liquid water at the bottom of the outer cavity. At this time, the compressible component installed on the moving cavity is in
an uncompressed state, and its tip is located at the deepest point of the trajectory groove of outer cavity.

The movement of main components and the pressure change of liquid water in booster cavity are shown in Figure 5D-G. Small part of the high-temperature and high-pressure steam discharged from the evaporation cavity would enter the moving cavity through the coincident steam inlets, as shown in Figure 5D, causing the internal pressure of moving cavity to increase. Under the action of the differential pressure, the moving cavity would have a downward velocity. At this time, the trajectory groove on the side wall of the outer cavity limits the movement mode of the moving cavity, so that the moving cavity can only move downward along the curved trajectory groove. And because of the varying depth of the trajectory groove, the compression spring installed on the side wall of the moving cavity would be compressed continuously until the compressible component reaches to the end point of the curved trajectory groove. During the downward movement, the compression spring installed at the bottom of the outer cavity is also compressed continuously, and the pressure of liquid water in the outer cavity is increased, as shown in Figure 5E.

Until the pressure of water reaches the internal pressure of evaporation cavity, the high-pressure water would enter the evaporation cavity along pipelines under the action of gravity, absorb heat, and enter the next cycle.

When the compressible component reaches the end point of the curved trajectory groove, the compression spring installed on the moving cavity will recover half of the compression under the action of elastic force, as shown in Figure 5F. At this time, the steam inlet channel is closed, and the steam outlets of moving cavity and outer cavity would be coincident; that is, the steam outlet channel would be opened. And then, the worked steam in moving cavity would be discharged and enter the condensation cavity. Meanwhile, the internal pressure of moving cavity is decreased, and the moving cavity would obtain an upward velocity under the actions of differential pressure and the bottom compression spring. At this moment, the moving cavity can only move upward along the linear path under the restriction of the trajectory groove. During the upward movement, the compression of the compression spring installed on the moving cavity remains unchanged, as shown in Figure 5G. When the moving cavity moves to upper end, the compression will return to zero. At that point, the steam outlet channel would be closed and the steam inlet channel would be reopened. Moreover, during the upward movement, the low-pressure liquid water would re-enter the outer cavity along pipelines from the condensation cavity and wait for the pressure boost.

In order to realize the reciprocating motion of moving cavity successfully, it is necessary to ensure that the steam inlet channel is always open and the steam outlet channel is always closed during the downward movement, while the opposite is true during the upward movement. If both the downward movement and the upward movement are simple linear motions, it will inevitably occur that the steam inlet channel and the steam outlet channel are simultaneously opened for a period of time, so the design of the combination of curved trajectory and linear trajectory is adopted to solve this problem. In addition, the design of the trajectory groove with varying depth can ensure that the moving cavity can only move along the curved path or along the linear path at the starting points of the two motions, rather than choose freely between the two paths. Furthermore, the stepped cylindrical cavity is selected for both the moving cavity and the outer cavity, which ensures that high-pressure liquid water can be obtained even considering the steam pressure drop.

In the traditional Rankine cycle, the condensed working fluid is delivered to the evaporator through a mechanical booster pump, which inevitably consumes part of electric energy. However, the booster cavity proposed in this paper uses high-temperature and high-pressure steam to pressurize the low-pressure working fluid without external force driving, with no electricity consumed. In addition, the steam and liquid water is working in different cavities, avoiding the complex two-phase mixing flow.

### 2.2.4 Condensation cavity

The structure of the condensation cavity is a water tank with immersive spiral coil, as shown in Figure 6. The spiral coil is placed in a water tank filled with cold water to dissipate heat through heat conduction, heat convection and heat radiation. In order to enhance the heat dissipation performance, a radiator is installed outside the water tank. The circumferentially uniformly distributed fins can greatly increase the heat dissipation area, so that the water tank will transfer heat faster to the environment.

The steam discharged from expansion cavity and booster cavity all enters the condensation cavity along the spiral coil to change phase and to release heat. After that, the low-temperature liquid water will reenter the outer cavity of booster cavity.

### 2.2.5 Generator

Once the scroll expander rotates, it will drive the connected generator to output electricity.
Overall, the Rankine cycle has been achieved through a thermodynamic process where the water is evaporated, expanded, condensed and pressurized successively, as shown in Figure 7.

3 | PERFORMANCE ESTIMATION

Although the thermal-electric conversion device has been analyzed from the points of view of structure and working principle, it is also essential to estimate its working performance.

3.1 | Focusing performance analysis

The ultimate goal of the thermal-electric conversion device is to convert solar energy into electrical energy. For the successful power generation of thermal-electric conversion device, it is the primary condition that the device effectively absorbs the sunlight reflected by solar concentrator. In addition, the focusing performance of the solar concentrator determines the heating area of the heat-absorbing plate of thermal-electric conversion device. Therefore, it is necessary to analyze the focusing performance of solar concentrator firstly.

In the present study, the TracePro software, which employed Monte Carlo Ray Tracing method to trace the rays, is used to analyze the heat flux distribution on the absorber. This simulation tool is widely used in the field of solar energy utilization to evaluate the optical performance of reflector, its validity has been verified in many studies. The Monte Carlo Ray Tracing method replicates the real photon interactions, in which stochastic paths of a large number of rays are followed as they interact with surfaces. The rays can be emitted, transmitted, reflected, absorbed or scattered. The behavior of each ray is automatically tracked and recorded until the ray is lost from the system or completely absorbed, and then the next bunch of the ray will be followed. Finally, the optical performance can be predicted by aggregating the recorded results.

Before the analysis, we need to identify some parameters. 

**Solar concentrator:** The solar concentrator is a paraboloid cable mesh reflector with 5 m diameter and 2 m focal length, and its outer circumference is divided into 48 sections. The reflectivity of the solar concentrator is set to 96%.
Absorber: The absorber is positioned at the focal region of the reflector, and the absorbing surface is defined as perfect absorber, while the relevant heat losses are considered in the thermodynamic analysis of Section 4.

Light source: The light source is circular with the same diameter as the diameter of reflector, 5 m, and it is placed 2.5 m from vertex of parabolic reflector. The intensity of the solar radiation is set to 500 W/m², and all incoming sunrays are perpendicular to the aperture area.

Before analyzing the simulation results, the verification of the independence of the number of rays is conducted. Usually, for most ray tracing simulations, as the number of rays increases, so does the simulation accuracy. Of course, if the number of rays increases to a certain extent, it will play a limited role in improving the simulation accuracy, but the demand for computation time and memory is greatly increased. Figure 8 shows that how different number of rays affect the heat flux distribution on the absorbing surface. As can be seen, results stabilize beyond one million rays, including number of absorbed rays, absorbed heat and average heat flux of absorber surface, and the focusing efficiency of solar concentrator. So, one million rays are used hereinafter, and the simulation results are shown in Figure 9.

In the process of designing absorber, one always seeks for the right minimum size. With small absorber size one can reduce heat losses as well as weight and cost of whole system. As can be observed from Figure 9B, the simulation is carried out under the condition that the radius of the absorber is 300 mm. At this time, the total amount of solar radiation absorbed by this area is about 7491.5 W, with a 76.3% focusing efficiency of the solar concentrator.

But in fact, most of the reflected rays are focused on a circle with the radius of 200 mm, as the area defined by the circle drawn in Figure 9B. At this time, it just happens that all the areas delineated are covered by rays, almost no areas that are not covered. After further analysis, the total amount of solar radiation at this time is about 6878.4 W, with a 70% focusing efficiency, as shown in Figure 9C. Compared with the case of 300 mm radius, although the solar radiation has a decrease, it can ensure that the surface of the heat-absorbing plate is fully heated, and reduce the area of heat-absorbing plate, thereby reducing its mass and cost, as well as reducing the deformation of the reflector caused by the thermal-electric conversion device.

So the heat-absorbing plate is preliminarily determined to be a circular plate with a 200 mm radius in this paper. This is only the result for some specific parameters. If we want to get better focusing performance, the precision of the solar concentrator need to be improved, which is not the focus of this paper.

3.2 | Thermodynamic analysis

3.2.1 | Thermodynamic process

Conversion efficiency is one of the main parameters for evaluating the performance of thermal-electric conversion device. Based on the first law of thermodynamics, the thermodynamic process of the proposed device is analyzed, and its output power and conversion efficiency are obtained.

The following assumptions are made before the analysis:

- The system is assumed to perform under stable operation status.
- Pressure drops and heat losses in the pipelines are neglected.
- The heat losses of the heating absorbing plate due to conduction, radiation and convection are taken into account by the heat-absorption efficiency, which is set to a constant 85.38%, and the detailed calculations are given in Section 4.

![Figure 8](image-url) Verification results of independence of the number of rays
The thermodynamic process is mainly composed of four different processes:

1. **Isobaric endothermic process**

Low temperature water absorbs heat from the heat-absorbing plate in evaporation cavity, which is considered to be an isobaric endothermic process. The heat flux $q_1$ can be expressed as:

$$ q_1 = m \left( h_1 - h_4 \right), $$

where $m$ is the mass flow rate at the inlet of evaporation cavity; and $h_1$ and $h_4$ are the enthalpy of working medium at the outlet and inlet of evaporation cavity, respectively.

2. **Isentropic expansion process**

Most part of the high-temperature and high-pressure steam discharged from evaporation cavity enters the expansion cavity to generate power, which can be simplified to an isentropic expansion process because there is no heat exchange with the environment. The output power of turbo expander $w_t$ is defined by:

$$ w_t = m_1 (h_1 - h_2), $$

where $m_1$ is the mass flow rate at the outlet of expansion cavity and $h_2$ is the outlet enthalpy of the expansion cavity.

Considering an electric generator efficiency, $\eta_g$, the electric output power $w_p$ can be expressed as:

$$ w_p = w_t \eta_g = m_1 (h_1 - h_2) \eta_g, $$

3. **Isobaric condensation process**

The steam discharged from expansion cavity and booster cavity all enter the condensation cavity to release heat, which is simplified to an isobaric condensation process. The heat release $q_2$ can be calculated by:

$$ q_2 = m_1 (h_2 - h_3) + m_2 (h_2 - h_3), $$

where $m_2$ is the mass flow rate at the outlet of booster cavity, with $m = m_1 + m_2, h_2$ is the outlet enthalpy of the booster cavity; and $h_3$ is the outlet enthalpy of the condensation cavity.

4. **Isentropic compression process**

The liquid water from condensation cavity is pressed by the booster cavity to reenter the evaporation cavity, which is simplified to an isentropic compression process, and the work consumed by compression $w_b$ is given by:

$$ w_b = m_2 (h_4 - h_3), $$

Finally, the conversion efficiency $\eta_c$ is expressed as:

$$ \eta_c = w_p / q_1. $$

### 3.2.2 Parameter calculation

In Rankine cycle, the parameters of condensing pressure, condensing temperature, evaporating pressure, and evaporating temperature all affect the system performance.

**Condensing pressure**

With other parameters unchanged, the lower the condensing pressure, the higher the efficiency. Considering the operability and portability of the power generation system, the condensing pressure is set to be the atmospheric pressure, which eliminates the relevant operation of maintaining the pressure in condensation cavity.

**Condensing temperature**

The coil cooling method in water tank is adopted in this scheme. For the Ranking cycle, the condensing temperature needs to be lower than the saturated water temperature under the condensation pressure and higher than the current ambient temperature. In fact, the lower the condensing temperature, the more heat released into the environment, resulting in more heat required for the endothermic process and lower cycle efficiency. Meanwhile, lower condensing temperature would lead to higher requirement for the condensing capacity, more need for coolant, and increased mass of the condensation cavity. Considering the above factors, the condensing temperature is tentatively set at 353.15 K, when the state of working fluid is unsaturated water.

**Evaporating pressure**

When other parameters remain unchanged, the higher the evaporating pressure, the higher the efficiency, as shown in the Figure 10. However, the increase in evaporating pressure also places higher demands on the supercharging capacity of the booster cavity and the pressure-bearing capacity of the device. Meanwhile, the total heat that the system can provide is certain when the size of the reflector is fixed. According to the equation of endothermic process, the higher the evaporating pressure is, the lower the maximum mass flow rate of working fluid is, as shown in Figure 10, and too low mass flow rate is not conducive to the size design of each component. Therefore, the evaporating pressure is determined to be 1 MPa.

**Evaporating temperature**

With other parameters unchanged, increasing evaporating temperature can improve the cycle efficiency, but also reduce
FIGURE 9 Simulation analysis of solar concentrator: A, schematic diagram of analysis model, B, absorber with a 300 mm radius, and C, absorber with a 200 mm radius.
the maximum mass flow rate. In order to make the steam quality of working fluid after expansion process closer to 1, to avoid damage to the expander, the superheat degree is tentatively designed to be about 20.

The mass flow ratio of steam entering the expansion cavity and the booster cavity is tentatively assumed to be 9:1, and the electric generator efficiency is tentatively assumed to be 95%, which can be optimized in subsequent studies.

The relevant parameters are listed in Table 1. Combined with the analysis results of Section 3.1, the related calculation results of thermodynamic properties are listed in Table 2. As can be seen, the input heat of Rankine cycle, that is, the absorbed solar heat, is 5872.78 W. After the thermal-electric conversion, the output power is 807 W, and the conversion efficiency is 13.74%. This is just the calculation results under the current parameters. In further study, parameter optimization can be performed to obtain higher efficiency and output power.

4 CYCLING FEASIBILITY ANALYSIS

The cycle process of the thermal-electric conversion device is a complex multicomponent and multiphysics coupling problem. It involves various processes, such as the thermodynamic state change and flow of working medium, phase change, heat transfer, energy conversion, reciprocating motion and rotational movement, which correspond to different mathematical equations, respectively. At present, there is no mature computing software that can simultaneously simulate the multicomponent and multiphysics coupling problem involved in the whole system. The usual method for this situation is that the mathematical models are established firstly on the basis of reasonable simplification for each module of the device, and then they will be transformed into a simulation model to be analyzed by employing a suitable simulation platform. The MATLAB/Simulink software is just suitable for this method, so it is adopted to carry out the cycling feasibility analysis in this paper.

4.1 Mathematical model

4.1.1 Evaporation cavity

Heat transfer always occurs from a higher temperature object to lower temperature object, and there are three heat transfer modes in evaporation cavity: conduction, convection, and radiation. To simplify the complexity of the heat transfer model, several assumptions are made:

- The solar heat flux is uniform.
- The heat transfer is considered as one dimensional.
- The physical properties of component materials are assumed to be constant.

The model is based on the heat transfer analysis of glass cover plate and hot plate, which includes the absorber plate and the heat tubes.

**Energy balance equation**

The energy balance equations for glass cover plate are given by:

- inner wall:
  \[ Q_{\text{gcond}} = Q_{\text{pgc}} + Q_{\text{pgr}} \]  
  \[ Q_{\text{fp}} = Q_{\text{acond}} \]  

- outer wall:
  \[ Q_{a} + Q_{\text{gcond}} = Q_{\text{gar}} + Q_{\text{gar}} \]

The energy balance equations for hot plate are given by:

- inner wall:
  \[ Q_{\text{fp}} = Q_{\text{acond}} \]  

- outer wall:
  \[ Q_{a} + Q_{\text{gcond}} = Q_{\text{gar}} + Q_{\text{gar}} \]
outer wall:
\[ Q_p + Q_{fp} = Q_{pgc} + Q_{pga} + Q_c + Q_t, \]  
(10)

The total heat loss:
\[ Q_{loss} = Q_{pgc} + Q_{pga} + Q_c + Q_t. \]  
(11)

The heat-absorption efficiency can be expressed as:
\[ \eta = 1 - Q_{loss}/Q_{total}. \]  
(12)

where the subscript \( g \) is the glass cover plate, \( p \) is the hot plate, \( a \) is the atmosphere, \( f \) is the working fluid, \( e \) is the side insulation, \( r \) is the radiation, \( c \) is the convection, \( cond \) is the conduction, \( total \) is the total absorbed heat, and \( \alpha \) is the absorption of glass cover plate.

Heat transfer coefficients
Conductive heat transfer coefficient. The conductive heat transfer coefficients of glass cover plate, absorber plate and thermal insulation layer all can be given by:
\[ h_c = \lambda_c/\delta_c, \]  
(13)

where \( \lambda_c \) and \( \delta_c \) are the thermal conductivity and the thickness of study object, respectively.

The conductive heat transfer coefficient of heat tubes can be given by:
\[ h_t = 1/(D_o \ln(D_o/D_i)/(2\lambda_t)), \]  
(14)

where \( D_i \) and \( D_o \) are the internal and external diameters of tubes, respectively.

Radiative heat transfer coefficient. The radiative heat transfer coefficient between glass cover plate and atmosphere is given by:
\[ h_{gar} = \epsilon_g \sigma (T_{g1}^2 + T_{a1}^2) (T_{g1} + T_{a1}), \]  
(15)

where \( \sigma \) is the Stefan-Boltzmann constant and \( \epsilon_g \) is the emittance of glass cover plate.

The radiation heat transfer coefficient between absorber and glass cover plate can be calculated from the equation:
\[ h_{pgc} = \sigma(T_{pl}^2 + T_{g2}^2)(T_{p1} + T_{g2})/(\epsilon_p^{-1} + \epsilon_g^{-1} - 1), \]  
(16)

where \( \epsilon_p \) is the emittance of absorber plate.

Convective heat transfer coefficient. The convective heat transfer coefficient between glass cover plate and atmosphere is given by:
\[ h_g = 5.7 + 3.8v_{wind}^2, \]  
(17)

where \( v_{wind} \) represents the wind speed.

The convection heat transfer coefficient between absorber plate and glass cover plate calculated by the Nusselt number can be given by:
\[ h_{pgc} = Nu_{pgc} \lambda/\delta_{pg}, \]  
(18)

where \( \lambda \) is the air thermal conductivity and \( \delta_{pg} \) is the air gap distance between the glass cover plate and absorber plate, and the Nusselt number is given by:
\[ Nu_{pgc} = 1 + 1.44[1 - 1780/Racos\alpha]^+ \]  
\[ [1 - 1780(sin 1.8\alpha)^{1.6}/Ra \cos \alpha] + [(Ra/5830)^{1/3} - 1]^+, \]  
(19)

where the sign + is used to account only for positive values, \( Ra \) is the Rayleigh value, and \( \alpha \) is the angle at which the receiver is tilted.

The flow state of working fluid in receiver tubes can be single-phase or two-phase.

Single-phase:
Laminar flow:
\[ Nu_l = 4.36, \]  
(20)

Turbulent flow:
The Nusselt number is calculated by the Gnielinski correlation:
\[ Nu_l = (f/8)(Re - 1000)Pr_l/(Pr_g)0.11/(1 + 12.7\sqrt{f}/8(Pr_g^{2/3} - 1)). \]  
(21)

\[ f = (1.82lgRe - 1.64)^{-2}, \]  
(22)

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

Two-phase:
When there is two-phase convection inside the receiver tube, it becomes flow boiling problem.
\[ Fr = G^2/(\rho_l^2 gD_i). \]  
(23)

When \( Fr < 0.04 \), it is stratified flow, heat transfer coefficient is calculated by the Shah correlation:
\[ h_2/h_i = 3.9Fr^{0.24}(x/(1-x))^{0.64}(\rho_l/\rho_g)^{0.4}, \]  
(24)
\[ h_i = 0.023(\lambda_l/D)(G(1-x)D/\mu_l)^{0.8}(Pr_l)^{0.4}, \]  
(25)
TABLE 3  Main parameters of evaporation cavity

| Preset parameters | Glass cover plate | Absorber plate | Heat tubes | Insulation | Air space | Atmosphere | Calculated parameters |
|-------------------|-------------------|----------------|------------|------------|-----------|-------------|-----------------------|
| Density: 2220 kg/m³ | Material: aluminum | Material: copper | Material: mineral wool | Thickness: 0.02 m | Xi’an | Wind speed: 2.1 m/s | Length of each heat tube |
| Thermal conductivity: 0.76 W/mK | Density: 2790 kg/m³ | Density: 8978 kg/m³ | Density: 30 kg/m³ | Specific heat capacity: 1380 J/kgK | | Temperature: 288.75 K | 1.435 m |
| Specific heat capacity: 840 J/kgK | Thermal conductivity: 210 W/mK | Thermal conductivity: 387 W/mK | Thickness: 0.025 m | | | | Heat-absorption efficiency 85.38% |
| Dimensions: D 0.4*0.032 m | Dimensions: D 0.4*0.032 m | Dimensions: D 0.004*0.0005 m | Dimensions: D 0.004*0.0005 m | | | | |
| Emissance: 0.88 | Emissance: 0.95 | Emissance: 0.05 | Emissance: 0.05 | | | | |
| Transmissivity: 0.92 | Absorptivity: 0.95 | Absorptivity: 0.95 | Absorptivity: 0.95 | | | | |

where Fr is the Froude number; G is the mass flux; h₁ is the heat transfer coefficient assuming the tube is fully occupied by liquid; x is the steam quality; and ρₙ and ρₙ are the densities of liquid and vapor phases, respectively.

When Fr > 0.04, it is annular flow, and the heat transfer coefficient is calculated by Chen correlation:

\[ h_2 = h_{NB} S + h_1 F \]

\[ h_{NB} = 9690 \left( 9 + \frac{1}{(1 - (P/P_{cr})^2)} \right) \]

\[ (q/20000)^{0.9-0.3(P/P_{cr})^{0.15}} (P/P_{cr})^{2.27} \]

\[ S = (1 + 1.15 \times 10^{-6} P^2 (Re)^{1.17})^{-1} \]

\[ F = 1 + (2.4 \times 10^4) (Bo)^{1.16} + 1.37 (Xn)^{-0.86} \]

\[ Bo = q/G\Delta h, \quad (30) \]

\[ X_n = (\rho_e/\rho_l)^{0.5} (\mu_e/\mu_l)^{0.1}(1-x)^{0.9}, \quad (31) \]

where \( P_{cr} \) is the critical pressure of water, \( P_{cr} = 22.1 \text{MPa} \); \( q \) is the heat flux; \( Bo \) is the boiling number; \( \Delta h \) is the vaporization heat; \( X_n \) is the Martinelli number; and \( \mu_e \) and \( \mu_l \) are the dynamic viscosities of liquid and steam phases, respectively.

Relevant parameters of evaporation cavity are shown in Table 3.

4.1.2  Expansion cavity

The fundamental geometric form which describes a scroll machine is an involute of a circle, created by unwrapping a circle. The equations for the involutes of scroll wrap can be given by:

inner wall:

\[ x_i = a[ \cos (\varphi + \alpha) + \varphi \sin (\varphi + \alpha)] \]

\[ y_i = a[ \sin (\varphi + \alpha) - \varphi \cos (\varphi + \alpha)] \]

outer wall:

\[ x_o = a[ \cos (\varphi - \alpha) + \varphi \sin (\varphi - \alpha)] \]

\[ y_o = a[ \sin (\varphi - \alpha) - \varphi \cos (\varphi - \alpha)] \]

where \( a \) is the radius of basic circle, \( \alpha \) is the initial angle of involute, \( \varphi \) is the involute angle, and the subscript is \( i \) for the inner wrap or \( o \) for the outer wrap.

Some basic parameters of the scroll expander are given by:

scroll pitch: \( P = 2\pi a \).

thickness of the scroll wrap: \( t = 2a \alpha \).

orbiting radius: \( R = P/2 - t \).

ending angle of the expansion process: \( \theta_e = 2\pi (N-1) \).

Expansion pockets: \( N \).

The working chamber volume is determined by the profiles of the fixed and orbiting scrolls. The built-in volume ratio of an expander is defined as the volume of the expansion chamber at the end of the expansion process over the one at the beginning of the expansion. It can be obtained from:

\[ \varepsilon = V_d/V_s, \quad (34) \]

where

\[ V_d = \pi P(P-2\pi)[2N-1]H, \quad (35) \]

\[ V_s = \pi P(P-2\pi)[1+\theta_e/\pi]H, \quad (36) \]
where $\theta_0$ is the initial expansion angle and $H$ is the height of the scroll wrap.

Based on the above equations and the set expansion ratio, the specific structure of the scroll expander can be designed.

Take two symmetric crescent-shaped pockets as the control volume, the parameters of the steam in the control volume are assumed to be uniform, and then, the energy balance equation can be expressed as

$$dU = dQ + h_i dM_i - h_o dM_o - dW,$$  \hspace{1cm} (37)

where $dQ$ is the heat exchanged between the expansion cavity and the external environment, and $h_i$ and $h_o$ are the enthalpy of the incoming and outgoing steam, respectively;

$dU$ is the variation in thermodynamic energy of steam in the expansion cavity, and it can be given as:

$$dU = d(Mu) = u dM + M du.$$  \hspace{1cm} (38)

The van der Waals equation is selected as the state equation of steam, which can be expressed as:

$$p = RT/(v - b) - a/v^2,$$  \hspace{1cm} (39)

where $a$ and $b$ are the van der Waals constants; $v$ and $T$ are the specific volume and the temperature of steam, respectively; and $R$ is the gas constant.

The specific thermodynamic energy $u$ of steam can be can be derived as:

$$u = C_v T - a/v + u_0,$$  \hspace{1cm} (40)

where $C_v$ is specific heat at constant volume and $u_0$ is a constant.

The specific enthalpy $h$ can be can be derived as:

$$h = C_v T - a/v + pv + u_0.$$  \hspace{1cm} (41)

$dW$ is the expansion work done by steam in the expansion cavity and it can be given as:

$$dW = pdV.$$  \hspace{1cm} (42)

The volume of the control volume can be expressed as:

$$dV/d\theta = P(P - 2t)H,$$  \hspace{1cm} (43)

where $\theta$ is the shaft angle.

Here, the heat exchange between the steam and the walls is ignored, and the expander process is regarded as an isentropic one. Combing $\theta = \omega t$ and $\omega = 2 \pi n/60$, the temperature of the control volume can be expressed as:

$$dT/d\theta = v[m/\omega(h_i - h_o) - (RT/(v - b))dV/d\theta]/VC_v.$$  \hspace{1cm} (44)

and the pressure of the control volume can be expressed as:

$$dp/d\theta = R v m / [(v - b)VC_v \omega] (h_i - h_o)$$

$$+ [2a/v^2 - v(p + a/v^2)(C_v + R)/(V(v - b)^2 C_v)] dV/d\theta.$$  \hspace{1cm} (45)

Then, the variations in steam parameters such as pressure and temperature during expansion process can be obtained.

Basic parameters of the scroll expander are shown in Table 4.

### 4.1.3 Booster cavity

Taking the moving cavity as the object, the motion equation is given by:

$$m_a c_a = m_1 g + p_1 A_1 - p_2 A_2 - p_0 (A_1 - A_2) - k x,$$  \hspace{1cm} (46)

where $m_a$, $a$, $x$ are the mass, acceleration, and displacement of the moving cavity, respectively; $A_1$ and $A_2$ are the action areas of steam and liquid water, respectively; $g$ is the gravitational acceleration; $k$ is the elastic coefficient of compression spring installed on the bottom of the outer cavity; and $p_0$ is the pressure of external environment; $p_1$ is the pressure of steam filled in the moving cavity and its variation law accords with the variation law of pressure in the process of charging and releasing gas in variable volume; the equation of $p_1$ can be derived based on the van der Waals equation given in Section 4.1.2 and the first law of thermodynamics:

$$dp_1/dt = [\nu^2 RT_1 (R + C_v)/(v_1 - b)^2 V_1 C_v) - 2a/(v_1 V_1)] dM/dt$$

$$+ [2a/v^2 V_1 - v_1 RT_1 (R + C_v)/(v_1 - b)^2 V_1 C_v)] dV_1/dt,$$  \hspace{1cm} (47)

where $V_1$ and $dM/dt$ are the volume of steam in the moving cavity and the mass flow rate of steam entering the moving cavity, respectively;

$p_2$ is the pressure of liquid water in the outer cavity, and it is given as:

$$dp_2/dt = -(dV_2/dt)/V_2 \beta,$$  \hspace{1cm} (48)

where $\beta$ and $V_2$ are the volume compression coefficient and the volume of water, respectively.

Main design parameters of the booster cavity are shown in Table 5.

### 4.1.4 Condensation cavity

The model is based on the heat transfer analysis of spiral coil and body of water tank, which include water tank and radiator.
Energy balance equation:
The energy balance equations for spiral coil are given by:
inner wall:
\[ Q_{f_{cc}} = Q_{c_{cond}} \quad (49) \]
outer wall:
\[ Q_{c_{cond}} = Q_{w_{cc}} \quad (50) \]
The energy balance equations for the body of water tank are given by:
inner wall:
\[ Q_{w_{bc}} = Q_{b_{cond}} \quad (51) \]
outer wall:
\[ Q_{b_{cond}} = Q_{abc} + Q_{abr} \quad (52) \]

### Table 4  Basic parameters of scroll expander

| Preset parameters         |               |               |
|---------------------------|---------------|---------------|
| Scroll pitch              | 0.012 m       |               |
| Scroll thickness          | 0.0035 m      |               |
| Pressure ratio            | 10            |               |
| Initial angle of involute | 0.9163 rad    |               |
| Rotation speed            | 850 rpm       |               |
| Material                  | Alloy steel   |               |
| Calculated parameters     |               |               |
| Built-in volume ratio     | 7.28          |               |
| Initial expansion angle   | 1.601 rad     |               |
| Scroll height             | 0.014 m       |               |

### Table 5  Basic parameters of booster cavity

| Outer cavity               | Upper part:                     | Lower part:                     |
|----------------------------|---------------------------------|---------------------------------|
| Inner diameter             | 0.104 m                        | 0.084 m                        |
| Height                     | 0.057 m                        | 0.05 m                         |
| Moving cavity              | Upper part:                     | Lower part:                     |
| Inner diameter             | 0.1 m                          | Inner diameter: 0.08 m          |
| Height                     | 0.04 m                         | Height: 0.05 m                  |
| Cavity thickness           | 0.002 m                        |                                |
| Maximum displacement of    | 0.015 m                        |                                |
| moving cavity              |                                |                                |
| Spring stiffness           | 10 N/mm                        |                                |
| Main material              | Aluminum                       |                                |

where the subscript \( c \) is the coil, \( w \) is the cool water, \( f \) is the working fluid, \( b \) is the body of water tank, including the radiator, \( a \) is the atmosphere, \( r \) is the radiation, \( c \) is the convection, and \( cond \) is the conduction.

For the condensation cavity, the corresponding calculation of the convective and radiative heat transfer coefficients between the external surface of the radiator and the atmosphere, as well as the thermal conductivity coefficients of the spiral coil, water tank, and radiator, is similar to the relevant analysis of the evaporation cavity, which will not be repeated here.

For the heat transfer between the cool water and the water tank, the Nusselt number is given by the Churchill-Chu correlation:
\[ Nu = \left( 0.825 + 0.387(Gr \cdot Pr)^{1/6}/[1 + (0.492/Pr)^{9/16}]^{27/27} \right)^{2} \quad (53) \]

For the heat transfer between the cool water and the spiral coil, the Nusselt number is given by the Churchill-Chu correlation:
\[ Nu = \left( 0.60 + 0.387(Gr \cdot Pr)^{1/6}/[1 + (0.559/Pr)^{9/16}]^{23/23} \right)^{2} \quad (54) \]

The heat transfer between the working fluid and the spiral coil can be single-phase or two-phase.

**Single-phase:**

Laminar flow:
\[ Nu_f = 4.36 \quad (55) \]

Turbulent flow:

The Nusselt number correlation is given by the Gnielinski correlation:
\[ Nu_f = (f/8)(Re - 1000) Pr_f (Pr_f/Pr_w)^{0.11} c_R \sqrt{1 + 12.7 \sqrt{f/8} (Pr_f^{2/3} - 1)} \quad (56) \]
\[ c_R = 1 + 10.3 (d_i/R)^3 \quad (57) \]

where \( c_R \) is the correction coefficient for spiral coil and \( d_i \) and \( R \) are the internal diameter of spiral coil, respectively.

Two-phase:

The Nusselt number correlation is given by the Shah correlation:
\[ Nu = 0.023(G(1-x)D/\mu)^{0.8} Pr^{0.4} (1-x)^{0.8} + 3.8x^{0.76} (1-x)^{0.04} / Pr_f^{0.38} \quad (58) \]

Relevant parameters of condensation cavity are shown in Table 6.

### 4.2  MATLAB/Simulink implementation

In the thermodynamic analysis, the thermodynamic parameters of water and steam, such as pressure, temperature,
enthalpy, entropy, are required at any time. So it is necessary to establish the calculation model for water and steam firstly. The IAPWS-97 formulation,\textsuperscript{52} which is widely regarded as the standard equation of state for water and steam, is adopted to calculate the physical parameters of water and steam at any state in this paper.

According to the preset parameters and the established mathematical models in the previous sections, the subsimulation modules of each module, including evaporation cavity, expansion cavity, condensation cavity, and booster cavity, are established respectively to obtain the changes of relevant parameters during working process. Then these modules are combined and the thermodynamic model for calculating cycle performance is also added to get the simulation model of the whole system. The simulation model established in MATLAB/Simulink for proposed device is shown in Figure 11. The input and output parameters of each module are as follows:

### 4.2.1 Evaporation cavity

- **Input parameters:** pressure and temperature of high-pressure water discharged from booster cavity, superheat degree of working fluid after evaporation, and total heat supplied by reflector;
- **Output parameters:** pressure, temperature, mass flow rate, and enthalpy of working fluid.

### 4.2.2 Expansion cavity

- **Input parameters:** pressure, temperature, and mass flow rate of steam discharged from evaporation cavity, condensing pressure, and rotating speed of the expander.
- **Output parameters:** pressure, temperature, mass flow rate, and enthalpy of worked working fluid.

![FIGURE 11 MATLAB/Simulink implementation of proposed device](image-url)
4.2.3 Condensation cavity

Input parameters: pressure, temperature, and mass flow rate of worked steam discharged from expansion cavity and booster cavity; and condensing temperature.

FIGURE 12 Simulation results: A, evaporation cavity, B, scroll expander, C, P-V diagram for scroll expander; D, condensation cavity, E, downward movement of moving cavity, F, upward movement of moving cavity, and G, temperature-entropy diagram of the cycle process.
Output parameters: pressure, temperature, mass flow rate, and enthalpy of condensed working fluid.

4.2.4 | Booster cavity

Input parameters: pressure, temperature, and enthalpy of condensed water discharged from condensation cavity; pressure, temperature, and mass flow rate of steam discharged from evaporation cavity; and evaporating pressure.

Output parameters: pressure, temperature, and enthalpy of pressurized water; and pressure, temperature, and mass flow rate of worked steam.

4.2.5 | Thermodynamics

Input parameters: output enthalpy of each component.

Output parameters: temperature-entropy diagram and cycle efficiency.

5 | RESULTS AND DISCUSSION

The variations in steam quality, temperature of the working fluid along the heat-absorbing tube in evaporation cavity are shown in Figure 12A. It can be observed that the temperature of the working fluid gradually increases in the supercooling stage and the superheating stage, and stays constant in the evaporation stage, while the steam quality increases in the evaporation stage, and stays constant in the supercooling stage and the superheating stage. In addition, the heat transfer characteristics between working fluid and tubes are also analyzed, as shown in Figure 12A. In the supercooling stage, although the heat transfer coefficient increases gradually, the range is very small. After entering the two-phase stage, with the change of steam quality, the heat transfer coefficient of steam increases rapidly until it becomes saturated steam; and then in the superheating stage, the heat transfer coefficient decreases sharply, and is lower than the heat transfer coefficient in the supercooling stage. Eventually, the steam produced is divided into two parts, one would be expelled to the expansion cavity, and the other part would be expelled to the booster cavity.

Under the action of superheated steam, the scroll expander is driven to rotate. Figure 12B shows how the relevant parameters of steam in control volume vary with the rotation angle during expansion process. Figure 12C shows the calculated P-V diagram. With the rotation of the orbiting scroll, the volume of steam in the control volume increases gradually, while the pressure, temperature and enthalpy decrease accordingly. This also means that the internal energy of steam is converted to the mechanical energy output. The specifications of the scroll expander are as follows: The chamber volume when the expansion process starts is about 3.98 cm³ and that at the end of the expansion process is 29 cm³, which results in a built-in volume ratio of 7.28.

The variations in steam quality, temperature of the working fluid along the spiral coil in condensation cavity are shown in Figure 12D. The steam discharged from the expansion cavity is wet steam with a quality of about 0.9. It can be observed that the temperature of the working fluid stays constant in the two-phase stage and then gradually decreases in the supercooling stage, while the steam quality decreases in the two-phase stage, and stays constant in the supercooling stage. In addition, the heat transfer characteristics are also analyzed, as shown in Figure 12D. As can be observed, the heat transfer coefficient decreases both in the two-phase stage and in the supercooling stage. The difference is that the value of heat transfer coefficient and the decreasing speed of heat transfer coefficient in the two-phase stage are much higher than those in the supercooling stage.

After the steam is discharged from the evaporation cavity and enter the booster cavity, there would be a reciprocating motion of moving cavity under the actions of differential pressure and compression spring, also a pressure rise of liquid water. Figure 12E, 12 shows the pressure variations of the liquid water in outer cavity and the displacement variations of moving cavity during the downward and the upward movement. The maximum stroke of the reciprocating motion of moving cavity is preset to 15 mm. It is evident that the liquid water is compressed in the booster cavity when the moving cavity moves downward. During this time, the pressure of liquid water increases from 0.1 to 1 MPa in a short time, as shown in Figure 12E, and then the high-pressure liquid water would enter the evaporation cavity to be reheated, until the moving cavity reaches its maximum displacement. When the moving cavity moves upward, the pressure of liquid water would decrease rapidly form 1 to 0.1 MPa, as shown in Figure 12F, and the low-pressure liquid water from condensation cavity would be replenished into outer cavity, until the moving cavity returns to its initial state. The pressure losses of steam and liquid water in pipelines and elsewhere are not considered in the above analysis, but even if the pressure drop is taken into account, the stepped cavity design

| TABLE 7 | Preliminary estimated weight of the proposed device |
| Component | Weight (kg) |
| --- | --- |
| Evaporation cavity | 2.5 |
| Expansion cavity | 2.4 |
| Booster cavity | 0.8 |
| Condensation cavity | 3.5 |
| Generator | 2.0 |
| Others (pipelines, connection components, and so on) | 1.8 |
| Total | ~13.0 |
of the moving cavity and the outer cavity can still ensure the desired supercharging effect.

The temperature-entropy diagram of the cycle process of the proposed device is shown in Figure 12G. It consists of four processes: isobaric endothermic process (4-1), isentropic expansion process (1-2s), isobaric condensation process (2s-3), and isentropic compression process (3-4), which are conformed to the characteristics of the Rankine cycle. It is obvious that the working process of the Rankine cycle can be successfully implemented, and the cycling feasibility of the proposed device is verified; that is, the thermal-electric conversion can be realized.

6 | CONCLUSIONS

In this paper, a new conceptual design of thermal-electric conversion device in lightweight CSTP system is proposed and studied at a conceptual study level. From the above analyses, we can get the following conclusions:

1. Through the analyses of working principle, performance estimation and cycling feasibility, it is verified that the Rankine cycle and the thermal-electric conversion process can be successfully achieved.

2. The preliminary estimated weight of the proposed device is listed in Table 7. Compared with existing thermal-electric conversion devices, especially the Stirling engines which have the small order of magnitude of output power, the proposed device in this paper has higher power-to-weight ratio, as shown in Table 8. Even considering sealing, friction and other factors, the power-to-weight ratio of the proposed device is still higher than that of other existing devices. In addition, with little commercial experience to draw on, realistic cost estimate for the proposed device are extremely difficult to make. However, compared with Stirling engines, the proposed scheme does not need expensive products such as special materials used for regenerator, so its cost will not be higher.

3. Due to the compact and lightweight structure design, the proposed device can be used for the lightweight cable mesh reflector structure, and form a lightweight CSTP system with the solar tracker. The forming power generation system has the advantages of small land area, low construction cost, and portability, which can be used individually as a stand-alone system and for single-family houses, especially for the areas not covered by power network, such as remote mountainous areas.

4. The mechanical booster pump in conventional Rankine cycle is replaced by the proposed booster cavity structure. It means that no electricity or external power is needed during the thermal-electric cycle, and the device would be more compact and portable.

5. Although the analyses of the thermal-electric conversion device in this paper are based on the solar concentrator with 5 m diameter, it can still be used in other applications with different diameters, as long as the relevant parameters are changed.

In summary, the implementation of the proposed device is very conductive to the popularization and application of distributed layout CSTP system. Though there are still many technical problems on the materialization of thermal-electric conversion device, such as performance optimization, experimental verification, which are being solved and will be introduced in other articles, they would not affect the significance of this research topic.

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