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CONCEPTUAL DESIGN OF CLOSED-CYCLE HELIUM TURBINE SYSTEM FOR CENTRAL RECEIVER SOLAR ELECTRIC POWER GENERATION

Tingfeng Ke
Shanghai Advanced Research Institute
ketf@sari.ac.cn
Shanghai, China

Jingxuan Zhang
Shanghai Advanced Research Institute
zhangjx@sari.ac.cn
Shanghai, China

Yubo Zhu
Shanghai Advanced Research Institute
zhuyb@sari.ac.cn
Shanghai, China

Weiguang Huang
Shanghai Advanced Research Institute
huangwg@sari.ac.cn
Shanghai, China

ABSTRACT

A novel conceptual design of closed-cycle helium turbine system for tower solar to produce 50MWe power output is presented in this paper. A latent thermal energy storage utilizing high temperature phase change material as thermal storage medium is adopted to avoid the frequently shutdown of helium turbine due to the alternation of day and night. The conceptual design characteristics of the turbo-machine and heat exchangers show that the power system can achieve 45% thermal efficiency and has a more compact feature than closed-cycle air turbine system and in addition, the control method and part-load performance are presented. The helium turbine system would be promising choice of power conversion system for solar energy from the standpoints of efficiency, simplicity and reliability.

INTRODUCTION

Concentrated solar power (CSP) uses renewable solar resource to generate electricity while producing very low levels of greenhouse-gas emissions and has strong potential to be a key technology for mitigating climate change. When combined with thermal storage capacity, CSP plants can continue producing electricity, which makes it possible to provide reliable electricity [1]. There are four main CSP technology families categorised by the way they focus the sun's rays and the technology used to receive the sun's energy: parabolic trough, solar tower also known as central receiver systems (CRS), linear Fresnel and dish Stirling. Compared to the other technologies, the CRS can achieve very high temperature, thereby increasing the power conversion system (PCS) efficiency at which heat is converted into electricity and reducing the cost of thermal storage. CRS can also be easily integrated into fossil plants for hybrid operation with plenty options and potentially to generate more electricity annually via thermal storage [2] [3]. Currently, the typical design of solar PCS is the steam (Rankine) cycle system with a molten salt thermal storage system. However, the limitation of efficiency will increase the quantities of the heliostats, which comprise the most costly subsystem of the solar plant. In addition, good direct normal irradiance (DNI) is usually found in arid and semi-arid areas with reliably clear skies, where the water source is less too.

It is well known that the Brayton cycle is the best choice for high temperature heat source, since which has been so successful in aero engines and heavy gas turbine. The closed cycle gas turbine power conversion system is not a new concept and has been studied and operated since 1939 [4]. Kuo [5] evaluated the technological and economic feasibility of closed cycle air gas turbine with an advanced design central receiver for solar power generation in early 1979 and figured out that closed cycle with air or helium as the working fluid could increase the efficiency of solar electric power generation when the advanced central receivers of gas-cooled designs was integrated. The working fluid was air rather than helium in the conceptual design of the power system just because of the consideration about technology availability at that time [6]. The helium, being neurotically neutral and chemically inert, high specific heat, good thermal conductivity, regarded as the favoured working fluid for the close cycle with high temperature heat source, especially those plants with large power output. In the past decades, the helium turbine has attracted much more attentions and made great progress. These investigations have validated the design and operation of closed helium turbomachinery by means of test and well demonstrated the technical feasibility of the high efficiency closed-cycle helium turbine system. McDonald [7] reviewed the experience gained, and the lessons learned from the operation of helium gas turbine plants and related test facilities based on open literature sources. The Japan Atomic Energy Research Institute has carried out a series of research and design (R&D) on the closed Brayton cycle with helium as working fluid. Three scaled model test programs have been conducted, including helium compressor aerodynamic performance test, magnetic bearing development test and gas turbine system operation and control test [8]. In recent years, a closed Brayton cycle test loop was built in Chinese Academy of Sciences (CSA), which has contributed to verify the key technologies of closed-cycle helium turbine, including the design system of helium multi-stage compressor and turbine including aerodynamic and geometry, sealing method, power regulation and control strategies [9]. Furthermore, the developments of material and heat transfer technologies also reduce the difficulty to apply helium turbine system to high temperature heat source.

The current paper mainly focuses on the design of helium turbine system coupled with high temperature thermal energy storage (TES) for CSP. It should be stated that the heliostat system design is out of the scope of this paper and all the requirement of the power block and TES could be satisfied well. The paper is organized as follows. The description of the central receiver system with helium turbine is given first and next is the preliminary designs of turbomachinery, heat exchanger, TES, and control strategies. A comparison between helium turbine system and other PCS is presented and some conclusions are given at the end.

DESCRIPTION OF HELIUM CENTRAL RECEIVER SYSTEM

A typical central receiver system comprises solar collector field, solar receiver and power block. TES is also a necessary component for CSP, which could store heat energy for short periods of time and thus have a "buffering" capacity that allows them to smooth electricity production considerably and eliminate the short-term variations other solar technologies exhibit during cloudy days and nights.

The simple schematic diagram of central receiver PCS with energy storage capability is shown in Figure 1. There are three main working cycles in the operation of PCS. Cycle1 and cycle2 operates separately with the receiver as heat source when solar radiation is sufficient. The helium of cycle1 and cycle2 both flow through the receiver and transfers the solar thermal to power block and TES respectively. In the cloudy day and night, the power block will be supplied by the heat storage in TES, shown as cycle3. The major characteristics of the system are that the helium serves as the HTF and at the same time as the working fluid of power block and the high temperature latent TES was integrated with power block.

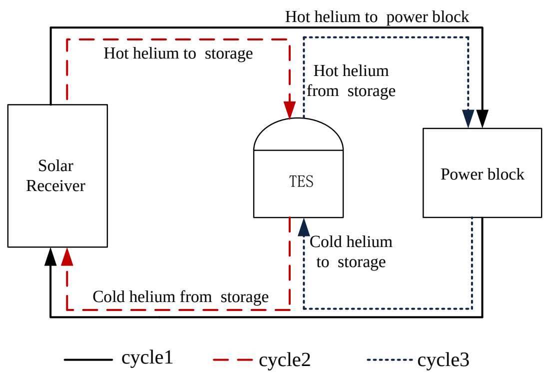

Figure 1 Simple schematic diagram of operating cycles

Figure 2 shows the helium central receiver system flow schematic. The whole work process of the system can be divided into two parts. At sun-time, the cycle1 and cycle2 are separated by closing valve e&f. For cycle1, the helium is compressed in low-pressure compressor first, recooled in the intermediate cooler, and then brought to the maximum process pressure in the high-pressure compressor. Subsequently, the helium is preheated in the recuperator and heated up to the maximum process temperature in the receiver. After expansion in the turbine, the helium releases the heat still contained in it to the helium from the high-pressure compressor part. Finally, in the precooler the helium is recooled to the lowest process temperature and ready for a new cycle. For cycle2, the helium from the receiver flows through TES directly, where it transfer the heat to the high temperature molten salt. An axial fan is used to compensate the pressure loss in the flow process. Another work process of the helium central receiver system is at cloudy days and nights. Under such condition, the cycle3 will start up by closing the valve a/b/c/d and opening the valve e&f. The TES will serve as the receiver to provide the heat to power block to continue to produce the electricity, which is beneficial to the grid and turbomachinery service life.

The selection of the proper thermal capacity of TES is a key factor for the central receiver system design. The larger helium velocity in the pipes causes the greater pressure drop. In addition, the helium velocity is a function of the helium mass flow rate and system pressure level for a certain pipe diameter, which means the non-optimized thermal distribution and pressure level will increase the pressure loss, i.e. increase the power consumed by axial fan and then reduce the net power output of the system.

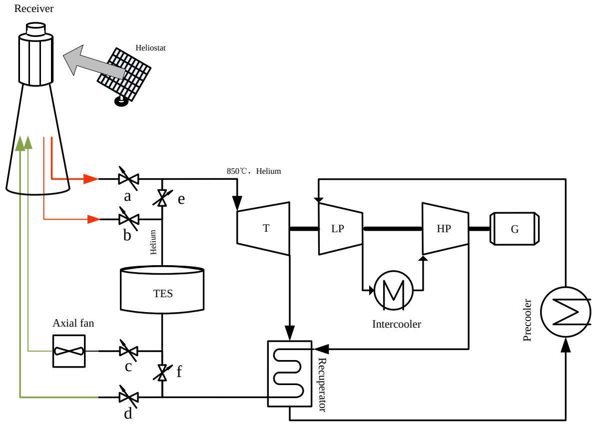

Figure 2 Helium central receiver system flow schematic

COMPONENT DESGIN AND PERFORMANCE OF HELIUM CENTRAL RECEIVER SYSTEM

Design point of the PCS

The power conversion cycle is theoretically based on the Brayton cycle with recuperated, intercooled and precooled sub-processes. The net specific power output of the cycle can be expressed as following:

$$w_{net}=C_pT_1\left[\tau\eta_T(1-\frac{1}{\sigma\sigma_R\gamma^{\varphi}})-\frac{\gamma_L^{\varphi}-1}{\eta_L\eta_M}-\frac{\xi}{\eta_H\eta_M}(\frac{\gamma^{\varphi}}{\sigma_{\Pi}\gamma_L^{\varphi}}-1)\right]$$

$$\sigma=(\sigma_A\sigma_{\mathrm{Pre}})^{\varphi},\quad \sigma_{\Pi}=(\sigma_I)^{\varphi/2},\quad \sigma_R=(\sigma_{R-H}\sigma_{R-C})^{\varphi}$$

The specific heat supplied and the cycle efficiency can be defined as:

$$q_{in}=C_pT_1\left\{(\tau-\xi)-\frac{\xi}{\eta_H\eta_M}(\frac{\gamma^{\varphi}}{\sigma_{\Pi}\gamma_L^{\varphi}}-1)-\alpha\left[\tau-\tau\eta_T(1-\frac{1}{\sigma\sigma_R\gamma^{\varphi}})-\xi-\frac{\xi}{\eta_H\eta_M}(\frac{\gamma^{\varphi}}{\sigma_{\Pi}\gamma_L^{\varphi}}-1)\right]\right.$$

$$\eta_e=\frac{w_{net}}{q_{in}}=\frac{\tau\eta_T(1-\frac{1}{\sigma\sigma_R\gamma^{\varphi}})-\frac{\gamma_L^{\varphi}-1}{\eta_L\eta_M}-\frac{\xi}{\eta_H\eta_M}(\frac{\gamma^{\varphi}}{\sigma_{\Pi}\gamma_L^{\varphi}}-1)}{(\tau-\xi)-\frac{\xi}{\eta_H\eta_M}(\frac{\gamma^{\varphi}}{\sigma_{\Pi}\gamma_L^{\varphi}}-1)-\alpha\left[\tau-\tau\eta_T(1-\frac{1}{\sigma\sigma_R\gamma^{\varphi}})-\xi-\frac{\xi}{\eta_H\eta_M}(\frac{\gamma^{\varphi}}{\sigma_{\Pi}\gamma_L^{\varphi}}-1)\right]}$$

For any given value of the optimum pressure ratio for maximum cycle efficiency can be achieved by differentiating the function with respect to $x=\gamma^{(\kappa-1)/\kappa}$ and equating to zero, the result is as following.

$$x_{\eta opt}=\frac{\tau\eta_T(2\alpha-1)+\sqrt{\tau^2\eta_T^2(2\alpha-1)^2+\frac{1}{\xi}\tau\eta_T\eta_{2C}\sigma\sigma_R\sigma_{\Pi}\gamma_L^{\varphi}(B-\alpha A)[B-A(1-\alpha)]}}{\sigma\sigma_R[B-A(1-\alpha)]}$$

In the equation:

$$A=\tau\eta_T-\frac{1}{\eta_L\eta_M}(\gamma_L^{\varphi}-1)+\frac{\xi}{\eta_H\eta_M}$$

$$\mathrm{B}=(1\text{-}\alpha)(\tau-\xi+\xi/\eta_H\eta_M)+\alpha\tau\eta_T$$

The optimized pressure ratio of HP and LP could be calculated respectively as following.

$$\gamma_L=(\frac{\eta_L\eta_M}{\eta_H\eta_M})^{\frac{k}{2(k-1)}}(\frac{1}{\sigma_I})^{1/2}\gamma_c^{\ 1/2}$$

$$\gamma_H=(\frac{\eta_H\eta_M}{\eta_L\eta_M})^{\frac{k}{2(k-1)}}(\frac{1}{\sigma_I})^{1/2}\gamma_c^{\ 1/2}$$

In generally, the same pressure ratio is assigned on the HP and LP compressors in order to simplify the thermal calculation and analysis. Therefore, the pressure ratio is defined as:

$$\gamma_L=\gamma_{\mathrm{H}}=(\frac{\gamma_c}{\sigma_I})^{1/2}$$

It can be seen that the cycle efficiency and work output are a function of a series of cycle parameters, which mainly depend on the attainable performances of its components and the cycle pressure losses. In the calculation of optimum efficiency, the following parameters are used in qualitative analysis, see table 1. The cooling tower has the ability to provide water with a minimum temperature 20℃. Thus, the helium would be cooled to 21.47℃ in the intercooler before entering the compressors. The system pressure drop depends on all of the pipeline loss, pressure drop of heat exchangers including intercooler, precooler, recuperator and receiver/TES. The pressure loss is assumed 1.5% of inlet pressure in the receiver and TES with accessorial pipes, 1% in high pressure side and 3% in low pressure side of recuperator, 1.5% in the intercooler and 2% in the precooler. The low-pressure drop of the system, high inlet turbine temperature and the high efficiency of each component based on technology availability could be beneficial to achieve high cycle thermal efficiency.

Table 1 Preferred Power conversion system design point requirement

Turbine inlet temperature (TIT)		850℃
Compressor inlet temperature		20/21.47℃
Basis power output		50MW
Design maximum pressure		3.0MPa
Component pressure losses (helium), $\Delta P/P_{in}$		
Pre-cooler		2%
Inter-cooler		1.5%
Rec.	high pressure side	1%
	low pressure side	3%
Receiver + ducting		1.5%
TES+ ducting		1.5%
Component efficiency		
compressor		88%
turbine		90%
Recuperator effectiveness		90%

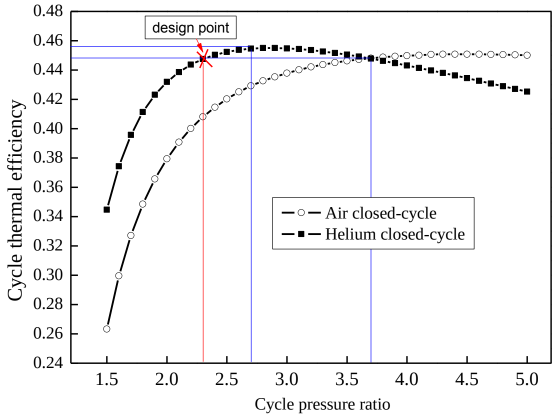

Figure 3 The relationship between cycle pressure ratio and thermal efficiency (TIT=850℃)

Conceptual design of turbomachinery and heat exchangers

Based on the design condition presented Table 1, the helium closed-cycle is optimized by in-house code and the relationship between cycle pressure ratio and thermal efficiency is shown in Figure 3. The air closed-cycle has been calculated as well. Compared to the air closed-cycle, the helium has a little higher maximum efficiency with smaller cycle pressure ratio. The calculation is based on same turbine inlet temperature 850 ℃ and same performance of components. The maximum pressure 3MPa of the system is both selected based upon a trade-off between component size and cost of present power scale 50MWe. The helium closed-cycle turbine system has all the potential advantages of air closed-cycle gas turbine system mentioned by Horton [5].

Table 2 Conceptual design of helium turbomachinery

Working fluid	air	helium
Low pressure compressor		
Inlet total pre., kPa	671.1	1276.6
Inlet total temp., ℃	20	20
Pressure ratio, -	2.13	1.545
Number of stages, -	6	10
Rational speed, rpm	5465	6137
Tip diameter, mm-through flow	940	1047
Axial length, mm-through flow	578	840
High pressure compressor		
Inlet total pre., kPa	1406.1	1942.3
Inlet total temp., ℃	21.5	21.5
Pressure ratio, -	2.13	1.545
Number of stages, -	6	12
Rational speed, rpm	5465	6137
Tip diameter(max.), mm-through flow	880	955
Axial length, mm-through flow	536	950
Turbine		
Inlet total pre., kPa	2925.45	2925.45
Inlet total temp., ℃	850	850
Expansion ratio, -	4.038	2.178
Number of stages, -	4	6
Rational speed, rpm	5465	6137
Tip diameter(max.), mm-through flow	1190	1221
Axial length, mm-through flow	750	1143
Total axial length, mm-through flow	1864	2933
Tip diameter(max.), mm-through flow	1190	1221

The conceptual design for both the turbomachinery and heat exchangers are presented with air and helium as working fluid respectively. The design point of the helium closed-cycle is fixed with cycle pressure ratio 2.38, which is smaller than that of the optimum cycle. The design pressure ratio of helium cycle could achieve the same high thermal efficiency as air closed cycle, shown as Figure 3. Smaller pressure ratio of helium cycle could reduce the problem caused by aerodynamic design and rotor dynamic stability of helium turbomachinery.

The properties of helium have pros and cons of effects on the design of components in the system. Substitution of helium for air modifies the turbomachine aerodynamic requirments because of the high sonic velocity of helium removes Mach number effects. However, there are some disadvantageous effects on structure, including long slender rotor and high hub-to-tip ratio blades. The conceptual design results of helium turbomachinery show these characteristic obviously, as shown in Table 2.

However, the high specific heat and excellent heat transfer capacity compared to air are beneficial to acquire more compact and higher efficiency heat exchangers. As shown in Table 3, the size of all the heat exchangers with air is 2.08 times larger than the one with helium. The large size of heat exchangers with air negate much of the advantage in the size of turbomachinery. As a result, the prominent advantage of helium closed-cycle system is still its compactness compared with air closed-cycle system based on same thermal efficiency.

Table 3 Comparison between air and helium closed cycle

	Air cycle : helium cycle
Thermal eff.	1:1
Size ratio of turbo-set	1:1.61
Size ratio of heat exchangers	2.08:1

Size ratio of turbo-set = Total axial length of through flow (air)/ Total axial length of through flow (helium)
Size ratio of heat exchangers = Heat transfer areas (air)/ Heat transfer areas (helium)

Design of the TES

The effective TES can realize uninterruptable electric supply from solar tower, which is also in favour of turbomachine preventing from fatigue damage because of repeatedly shutdown.

Nowadays, two-tank molten salt sensible storage is the most commonly used storage technique in power tower concept. In addition, research has been conducted to use low-cost filler materials to reduce the overall required amount of the relatively higher cost molten salt storage medium. In sensible thermal storage systems, the energy is charged/discharged by raising/lowing the temperature of the storage medium. Latent thermal storage system using phase changing materials (PCM), which take advantage of the large latent heat charged/discharged by changing a material from one phase to another. The amount of the energy stored is dependent on the specific heat and the phase change enthalpy. Due to the significant quantity of energy converted during the phase change, latent storage system offers an isothermal heat storage/releasing process and a higher storage density compared to the sensible storage system with the same temperature change. It potentially enables a smaller and lower cost storage system compared to the sensible storage system. Currently, no commercial PCM storage system has been used in utility-scale CSP applications [10] [11]. For maximizing utilization of the solar thermal energy and acquiring the constant turbine inlet temperature, the storage medium prefer to high temperature PCM.

In order to meet the high temperature requirements of PCS, PCM of inorganic salts eutectics and metals alloys are used as thermal storage medium. Salts have been the most studied storage medium to reduce the cost of thermal storage. For the current design, sodium chloride was chosen as PCM, which has highest melting temperature among candidate high temperature chloride salts, as shown in Table 4 and low price. Sodium chloride has freeze point of 800℃ and large latent heat, both of which can reduce the mass flow rate requirement at certain heat storage capacity.

The tough problem is the high corrosive of the high temperature chloride salts to common containment materials. The criteria for containment material selection have been discussed in [12] [13].

Table 4 Physical properties of chloride salts

	NaCl	KCl	$MgCl_2$	$CaCl_2$
Freeze temp. (℃)	800	771	714	775
Latent heat (kJ/kg)	481	356	452	255
Heat capacity 25℃(kJ/kg · K)	1.164	1.442	1.3339	1.523
Heat capacity 847℃(kJ/kg · K)	1.198	0.987	1.0034	0.95
Density(kg/m3) (liquid, 847℃)	1515	1465	1630	2040
Thermal conductivity (W/m · K)(solid)	6.615 (27℃)	6.57 (27℃)	1.82 (100℃)	-
Thermal conductivity (W/m · K) (liquid, 847℃)	1.004	0.943	1.678	1.583

The TES is designed to supply the heat continuously for 14 hours at least. As shown in Figure 4, with the increase of the total amount of heat to be stored in the TES, the volume of the storage tank required would increase and so do the mass flow rate of HTF in cycle2 and power consumption of axial fan. Therefore, large capacity of TES is not appropriate by considering the cost of TES, and auxiliary electricity-supply. The final TES is designed to have the capacity of 14h 10% load (~5MW) operation. The minimum thermal heat stored by TES is 17MW.

As mentioned before, the charging process of TES is separated from the power system. It means the TES system could avoid the high-pressure environment for the power system required in the design. The design pressure of the TES is linked with the thermal capacity, for 17MW, the pressure required is only about 0.35MPa. Low-pressure requirement of TES is beneficial to reduce the material cost.

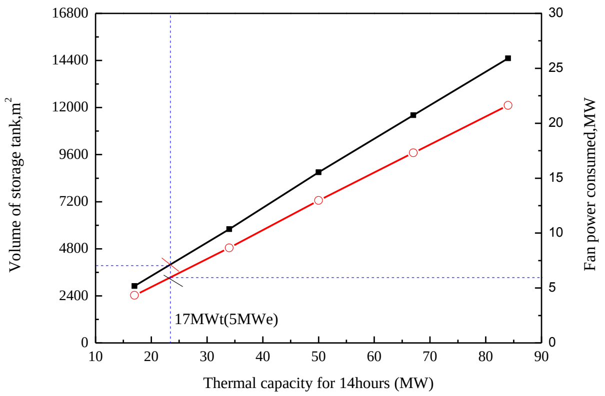

Figure 4 Thermal capacity of TES versus its size and power consumed

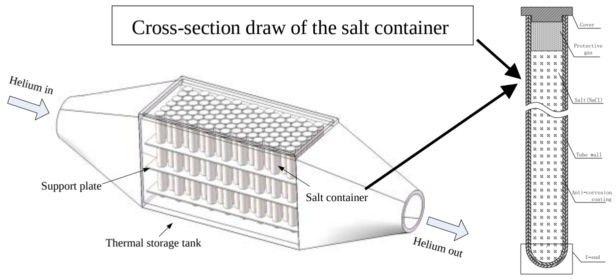

Figure 5 Schematic of TES

The structure type of TES is shown in Figure 5. The storage tank to be packed into the salt container could be cuboid or cylinder capsule. The solid salt container could be cylinder capsule same as test tube with anti-corrosion coating, like SiC ceramics, on the inner wall, as shown in Figure 5. The U-end type of the salt container benefits the heat expansion when the salt is heated by hot helium from receiver. The protective gas, such as nitrogen, fills the space between the solid salt and the cover of the container for safety. The arrangement of the salt containers are fixed after heat transfer simulation, and the number is determined by the storage thermal requirement. High temperature molten salt with high corrosion makes the research and design of TES difficult. Otherwise, TES needs high quality insulation measure to keep the salt with high temperature, and then reduce the melting thermal for TES at charging process.

Control method and Start-up procedure

The closed Brayton cycle offers many potentially attractive characteristics at part-load condition. The most important control method is named “inventory control”, which has also been verified in present closed-cycle helium power system. As shown in Figure 6, with the constant turbine inlet temperature, the system could provide reduced power output at almost constant high thermal efficiency. Even the thermal source has been changed from solar recevier to TES, the reduced turbine inlet temperature and reduced power output determined by thermal capacity could also acquire 80% of thermal efficiency at design point. Otherwise, the other control method would result in different system characteristic. It can been seen in the Figure 6 that the efficiency decreases well-proportioned with the decrease of power output when keeping the compressor inlet pressure constant.

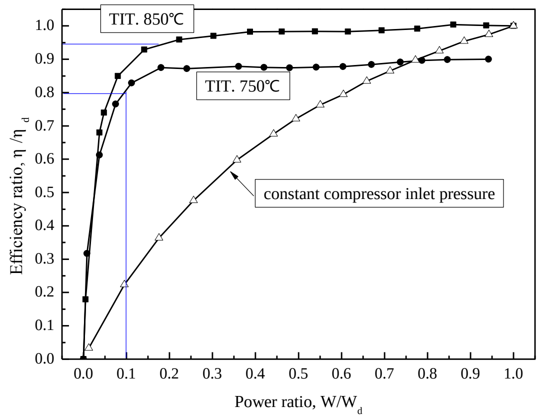

Figure 6 Part load performance of the closed-cycle helium turbine system

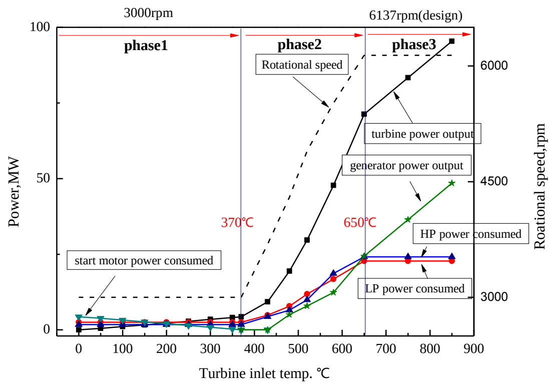

Figure 7 Start-up of the helium power system

There are some different start-up strategies of the helium closed-cycle system. One possible method is presented in Figure 7. The whole start-up process can be divided into three phases. Phase 1, the system was charged to the design working pressure and then the compressors and turbine are driven by an electric motor up to 3000rpm. With the increase of TIT, the motor power decreases eventually and when TIT approaches 370℃, the power output of turbine equals to the power consummation of compressors. At this point, the generator begins to produce electricity and the motor stops working. Phase 2, the rotational speed of the turbomachinery will increase with the TIT. When the TIT increases to 650℃, the design rotational speed (6137rpm) and the design mass flow achieved. Phase three, with the TIT increases further, the output of generator increases too to keep the rotational speed constant. When the TIT increases to 850℃ the system will work at the design condition, namely, the output power is 50MWe. When the power system operates at design condition, the collector field should be carefully adjusted to make sure the constant TIT solar energy input.

Comparison with other HTFs

Conventional HTFs used in experiment and commercial solar tower are listed in Table 5. All of these materials have advantages and disadvantages as HTFs.

Table 5 Conventional HTFs of solar tower

	Temperature	pressure
Saturated steam	260℃	4MPa
Superheated steam	400-500℃	5-12MPa
Molten salt	500-600℃	0.1MPa
Air	700-1000℃	0.1MPa
	800-900℃	1.5MPa

Water/steam has initially been adopted in solar towers such as Solar One, PS10, PS20, and Beijing Badaling [11] [14]. Water/steam has the potential advantages for HTF, such as high thermal conductivity, low pump power consumed. In addition, the steam can drive current mature steam turbine directly without inter-medium heat exchanger. However, producing the saturated steam especially the superheated steam in the receiver means that it will operate at high pressure with thick-walled tubes. The heavy-walled tubes limited the solar heat transfer to the steam, limited the solar flux that could be applied, and increased tube stresses [14].

Molten salts are the most commonly used HTF in current central receiver technology because of their high volume heat capacity, low vapour pressure, good heat transfer and low cost, which makes them economical enough to be used as a large bulk storage medium while their thermodynamic properties permit compact and efficient receivers. However, all the equipment in contact with the salt should be equipped electric tracing, and the thermal insulation throughout the system must be of the highest quality to make sure the nitrate salt temperature higher than freezing point, which will increase the system cost and reduce the system safety obviously.

The ECOSTAR' study [15] pointed out that one of the lowest "levelized electricity costs" for large scale CSP-plants would be for solar tower concept with pressurized air. There are a certain number of researches and experiments using volumetric air receiver integration of Brayton cycle or Rankine cycle or combined cycle as power block. JÜLISH solar power is a typical experiment used atmospheric air as the HTF to generate steam to drive a Rankine cycle. Buck have tested another advanced concept using air as HTF to feed the combustion chamber of the gas turbine, which can get a greater efficiency due to the high operating temperature and an efficient hybridization [16]. These concepts at several configurations and the potential of such technology would become competitive with conventional plants.

Helium, as mentioned in this paper, may be a more favourite HTF comparing to air for higher heat transfer coefficient. The results comparison between helium turbine system and air turbine system show that the former is more economic because the compactness of the heat exchange devices.

DISCUSSIONS

The conceptual design of helium central receiver system shows that the helium turbine system is a promising choice of power conversion system for solar energy. First, being gas, helium can utilize the maximum cycle temperature than Rankine cycle dose and then achieve higher efficiency, especially in part-load condition Second, helium has high specific heat and good thermal conductivity, which can reduce the geometry scale the heat exchange device largely and the save the cost. Third, the working pressure of helium turbine system is much lower than that of Rankine cycle (or s-CO2 cycle), which can decrease the thick-walled tube consumption and then save the save the cost. Finally, helium turbine system has accumulated so many experiences in operation and manufacture that it is possible to apply it in to CSP.

NOMENCLATURE

CSP	Concentrating solar power
CRS	Central receiver systems
PCS	Power conversion system
DNI	Direct normal irradiance
CAS	Chinese Academy of Sciences
TES	Thermal energy storage
HTF	Heat transfer fluid
PCM	Phase change material
w	Specific power
W	Power
C_p	Specific heat at constant pressure
q	Specific heat
Greek letters	
η	Efficiency
α	Recuperator effectiveness
ξ	Coolant coefficient $\xi = T_{2b} / T_1$
γ	Compression ratio
π	Turbine expansion ratio
κ	Specific heat ratio
τ	Temperature ratio $\tau = T_4 / T_1$
σ	Pressure recover coefficient
φ	$\varphi = (\kappa - 1) / \kappa$
Subscripts	
1	LP compressor inlet
2b	LP compressor outlet
4	Turbine inlet
C	Compressor
L	Low pressure compressor
H	High pressure compressor
T	Turbine
M	Mechanical
A	Receiver
Pre	Precooler
I	Intercooler
R	Recuperator
R-H	High pressure side of the recuperator
R-L	Low pressure side of the recuperator
in	Required
e	Economic
net	Network output

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