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Performance and Key Parameter Analysis of Composite Underwater Compressed Air Energy Storage

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Abstract

Due to the operation mode of " Water-to-Power Determination " in hydropower plants, the phenomenon of solar and hydro power Curtailment requires a large-scale and high-efficiency energy storage system for auxiliary regulation. To solve this problem, this paper proposes a composite underwater compressed air energy storage system coupling concentrating solarthermal system (CCS) and electric heating (EH). Firstly, the electric-electric efficiency of constant volume air storage is reduced due to the operation of off-design conditions in the process of energy storage and release, the system uses the constant pressure environment of the hydropower plant reservoir and adopts the air storage scheme of underwater flexible airbag. Secondly, to improve the consumption capacity and thermodynamic performance of the system, based on the advanced adiabatic underwater flexible compressed air energy storage(AA-UFCAES), the CCS and EH are coupled to fully preheat the high-pressure air during the expansion power generation. Furthermore, The thermodynamic performance of the three heat supply schemes without external heat source, coupled CCS, coupled CCS and EH heat source is compared and analyzed. The results show that the electric-electric efficiency, the round-cycle efficiency and energy storage density of the system coupled with CCS and EH are significantly improved compared with the previous two systems. Finally, to determine the optimal working state of the system, the influence of key parameters of the system is studied, and the feasibility of the proposed scheme is verified by simulation.

Keywords

Compressed Air Energy Storage; Constant Pressure Air Storage; Concentrating Solar Thermal System and Electric Heating; Thermodynamic Analysis.

1. Introduction

With the acceleration of the pace of building a highland of clean energy industry, Qinghai has entered a fast lane in the development and utilization of clean energy by virtue of its abundant water, light and wind resources. The potential of photovoltaic power generation in Qinghai Province is abandant, with an annual power generation potential of about 9 trillion kilowatt hours [1]. At the same time, Qinghai has built a series of cascade hydropower plant in the upper reaches of the Yellow River [2], It provides a solid foundation for the development of hydro-solar complementary system(HSCS). Based on this advantage, Qinghai has built the world's largest Longyangxia HSCS plant, which can help the efficient consumption of photovoltaic power through the rapid adjustment characteristics of hydropower units [3][5]. However, in the HSCS, the operation mode of the hydropower plant is often

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affected by weather, season and flood control regulation [4], resulting in that the power generation capacity cannot be matched with the large power grid dispatching, and the HSCS still has the problem of Curtailment of solar and hydro power. Energy storage system is one of the key technical means to deal with the above problems [5]. Among them, compressed air energy storage (CAES) technology has a series of advantages such as large energy storage capacity, Low expenditure, high safety and reliability, and fast load response, and is widely favored by the energy storage industry[6][7].

Advanced adiabatic compressed air energy storage (AA-CAES) is currently the mainstream research direction, Literature [8] reviewed the research status of AA-CAES power plant modeling, energy efficiency improvement, operation planning and market operation, pointed out the current research bottlenecks, and clarified the focus of future application research. In order to improve the charging and discharging efficiency of AA-CAES system, a new variable pressure ratio AA-CAES system was proposed in literature [9]. The thermodynamic model was established, and the thermodynamic characteristics of different expansion unit stages and variable expansion ratio operation modes were analyzed. However, conventional AA-CAES systems store high-pressure air in rigid containers, mines or caves with fixed volumes. When the energy is released, the high-pressure air is throttled through the pressure reducing valve to reduce to the rated intake pressure, resulting in throttling loss[10][11]. If the constant pressure environment of the hydropower plant reservoir can be used to maintain the pressure stability in the air storage chamber according to local conditions, the efficiency reduction caused by the variable operating conditions of the compressor and the expander can be effectively avoided. Therefore, the Underwater Flexible Compressed Air Energy Storage (UWF-CAES) technology based on AA-CAES came into being.

With regard to underwater compressed air energy storage, Reference [12] highlights the advantages of underwater constant pressure air storage by comparing with traditional constant volume air storage. It is pointed out that underwater air storage can be divided into two forms : rigid container and flexible container, and it is concluded that flexible container has higher feasibility. At the same time, the key technology of using flexible container is analyzed. In order to improve the performance of underwater compressed air energy storage, Reference [13] studied and constructed the mathematical model of energy storage process, simulated and analyzed the key factors affecting performance, and discussed the technology of improving efficiency. Reference [14] established the physical model of offshore wind power generation and underwater compressed air energy storage system, determined the optimal rated working condition under the constraint of air storage pressure, and optimized the air flow and heat transfer oil flow under variable working conditions. In order to reduce the impact of offshore wind power on the power grid, a model of offshore wind power-underwater compressed air energy storage system was established in Reference [15], and the energy efficiency and economy of the system were analyzed. Based on the advanced adiabatic underwater compressed air energy storage system, a thermodynamic model of offshore wind power-energy storage considering the fluctuation of wind energy is established in the literature [16], which reveals the influence of the fluctuation of offshore wind energy on the performance of underwater compressed air energy storage.

This research have shown that the core of improving the electrical-electrical efficiency of compressed air energy storage is to increase the air temperature entering the turbine [17].In order to further solve this problem, researchers usually introduce external heat sources, such as Concentrating Solarthermal System (CSS), to combine with CAES. On the basis of AA-CAES system, the thermodynamic model of compressed air energy storage with composite CSS is established in the literature [18], and the operating characteristics of the system are analyzed in depth. Literature [19] proposed a solar thermal storage compressed air energy storage system, and analyzed the thermal performance of the trough solar thermal oil thermal storage CAES and the tower solar molten salt thermal storage CAES. Based on the operation model and economic analysis model of AA-CAES system, Reference [20] studied the AA-CAES system planning and power grid collaborative scheduling technology with CSS coupling heat source. However, the research on the thermodynamic performance and key parameters of the underwater constant pressure compressed air energy storage system coupled with external heat sources is still blank.

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In this regard, on the basis of coupling CSS, this paper further introduces Electrical Heating (EH) to make up for the problem of insufficient preheating of high-pressure air in expansion power generation due to insufficient light, and proposes UWFCAES + CSS + EH system. The energy of electric heating mainly comes from the Curtailment of solar and hydro power in the HSCS system.This can not only improve the absorption capacity of the water-light complementary system, indirectly increase the energy storage capacity of the CAES system, but also provide a more stable heat source for the air at the CAES expansion inlet to better adapt to the water-light complementary system. Then the energy flow characteristics of the designed UWFCAES + CSS + EH system are analyzed. Furthermore, the thermodynamic performance of energy storage and release of three UWFCAES systems without external heat source, coupled CSS heat source, coupled CSS and electric heating heat source is compared and analyzed. On this basis, the effects of main parameters such as the stages of compressor and expander, the convective heat transfer coefficient of air storage chamber on the performance of UWFCAES + CSS + EH system are further explored. The research results are of great significance for a comprehensive understanding of the thermal performance of the underwater composite compressed air energy storage system and the selection of parameters.

2. Construction of System Model

2.1 Overview of System Design and Operation Principle

Compared with the traditional AA-CAES system, the UWFCAES + CSS + EH system has two main differences : (1) the air storage is changed from the previous constant volume air storage to the underwater constant pressure flexible air storage ; (2) On the basis of AA-CAES, the CSS and EH subsystems are added. The operation process of the whole system is shown in Figure 1.

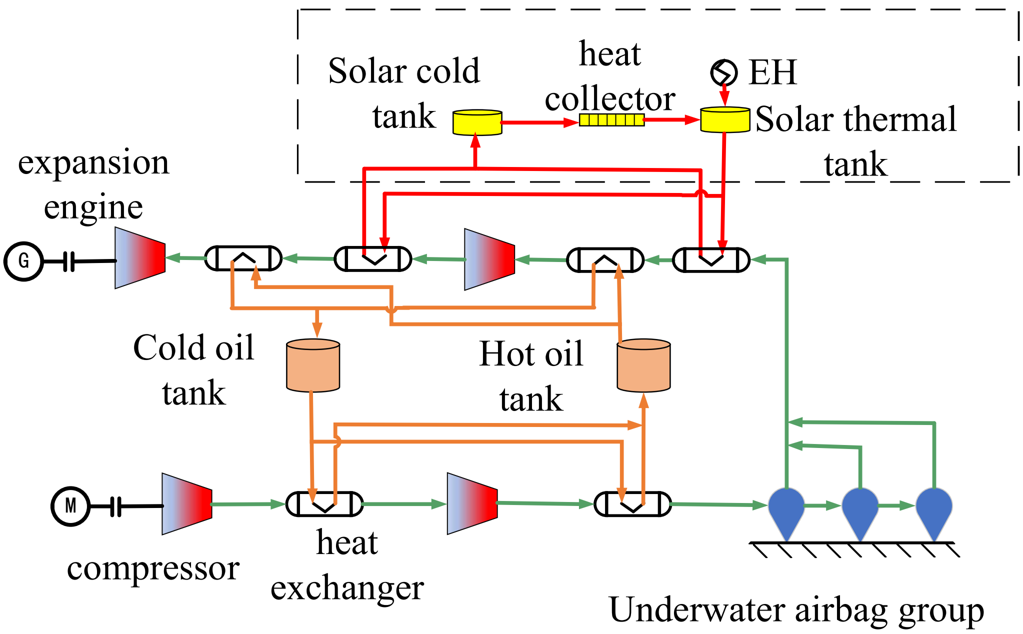

Figure 1. System operation flow chart

Operation principle : The compression energy storage stage can be divided into two parts : air storage and heat storage. Firstly, UWFCAES uses the surplus power of the water and light complementary system to drive the compressor to compress the air. Then, the air is compressed into a high temperature and high pressure air by a two-stage compressor, and the heat transfer is carried out in the heat exchanger with the heat transfer oil flowing out of the cold oil tank. The compressed heat is stored in the hot oil tank, and the cooled high-pressure air flows into the underwater constant pressure air storage for storage.

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At the same time, the low temperature heat storage medium from the solar cold tank is heated to the specified temperature through the solar collector, and then stored in the hot tank. In the case of insufficient light at night or cloudy days, the solar collector cannot heat the solar thermal storage medium to the specified temperature. At this time, the electric heater uses the additional power curtailment of the flood discharge of the hydropower plant to heat the solar thermal storage medium to the rated temperature.

In the static stage, the multi-stage compressed high-pressure low-temperature air is stored in the water constant pressure air storage, the high-temperature conductive medium heated by the solar collector is stored in the solar thermal tank, and the compressed heat collected by the inter-stage heat exchanger is stored in the hot oil tank.

In the energy release stage, when the power grid is in the peak period, the high-pressure air drives the generator to generate electricity through the expansion power generation subsystem. Firstly, the air is preheated to increase the air temperature by heat exchange with the high temperature medium flowing out of the solar thermal tank, then the high pressure air is further heated by heat exchange with the heat conduction oil of the regenerative subsystem, and then flows into the expander to do work to promote the power generation of the generator until the end of this cycle. After the expansion power generation stage, the system is in the static stage again. Different from the static stage, the temperature of various heat storage media stored in the low temperature tank is relatively low.

2.2 System Thermodynamic Model

In this study, the energy storage stage, the energy storage and release interval stage, and the energy release stage are divided, so as to establish a reasonable thermodynamic model. Different from the constant volume air storage mode, this study uses an underwater constant pressure air storage. The model maintains a constant pressure during air compression and expansion, and the volume and temperature change accordingly.

2.2.1 Energy Storage Phase

Compressor compression process : Air is compressed through the compressor, the compressor outlet temperature $T_{c,out}$ is calculated according to the formula of adiabatic variable process.

$$T_{c,out} = T_{c,in} \beta_c^{\frac{n_c - 1}{n_c}} \tag{1}$$

Among them, $T_{c,in}$ is the inlet temperature of the compressor, β_c is the compression ratio, and n_c is the adiabatic index. The power consumption per unit mass of air is :

$$W_c = c_p (T_{c,out} - T_{c,in}) \tag{2}$$

Here, c_p is the specific heat capacity of air at constant pressure.

Heat exchanger cooling : when passing through the heat exchanger, ignoring the heat loss of the heat exchanger, the outlet air temperature is the temperature $T_{c,cool}$ through the heat exchanger efficiency ε adjustment, cooling formula is :

$$T_{c,cool} = \varepsilon (T_{\mathrm{s,cold}} - T_{c,out}) + T_{c,out} \tag{3}$$

Among them, $T_{s,cold}$ is the cold tank temperature.

The pressure loss coefficient of the heat exchanger is described by the following formula :

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$$L_p = 0.0083\varepsilon(1-\varepsilon) \tag{4}$$

The heat absorbed by the unit mass heat storage medium in the heat exchanger is :

$$q_c = c(T_{c,\mathrm{out}} - T_{c,\mathrm{cool}}) \tag{5}$$

Here c is the specific heat capacity of air.

Air volume change : Under the condition of constant underwater pressure, the pressure of air in the air storage is always constant. According to the ideal air state equation, the volume and temperature of the air change with time, and the relationship between volume V and temperature T is :

$$V = \frac{nRT}{P_0} \tag{6}$$

Among them, P_0 is the constant pressure of the air storage under water.

Pressure ratio and temperature change : The relationship between pressure ratio β and temperature T with time can be described by the following equation :

$$\frac{d\beta}{dt} = \frac{c_p T_{\mathrm{in}} \dot{m}_c + h_c A(T_{\mathrm{rw}} - T)}{c_v p_0 V} R_g \tag{7}$$

$$\frac{dT}{dt} = \frac{c_p T_{\mathrm{in}} \dot{m}_c + h_c A(T_{\mathrm{rw}} - T) - c_v \dot{m}_c T}{c_v p_0 V \beta} R_g T \tag{8}$$

Among them, T_{in} is the inlet temperature of the air storage chamber, h_c is the convective heat transfer coefficient, A is the surface area of the air storage chamber, T_{rw} is the wall temperature of the air storage chamber, c_p is the specific constant volume and specific constant pressure heat capacity of the air, R_g is the air constant, m_c is the air mass flow rate.

2.2.2 Energy Storage and Release Interval Stage

Due to the outward heat dissipation of the air during the energy storage and release interval, the relationship between the pressure ratio, temperature and time in the air storage chamber during the energy storage and release interval can be obtained.

$$\frac{\mathrm{d}\beta}{\mathrm{d}t} = \frac{R_g h_c A(T_{rw} - T)}{c_v p_0 V} \tag{9}$$

$$\frac{\mathrm{d}T}{\mathrm{d}t} = \frac{h_c A(T_{rw} - T)}{c_v p_0 V \beta} R_g T \tag{10}$$

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2.2.3 Energy Release Stage

In the energy release stage, according to the presence or absence of external heat source, the establishment of thermodynamic model in the energy release stage can be divided into two cases.

There is an external heat source, that is, when there is light heat or light heat and electric heating, the air released by the air storage chamber first flows through the heat exchanger of the external heat source system, and then flows through the heat exchanger of the UWFCAES system. The air temperature at the outlet of the heat exchanger of the external heat source system is :

$$T_{\mathrm{t,f}} = \varepsilon(T_{\mathrm{h}} - T_{\mathrm{t,out}}) + T_{\mathrm{t,out}} \tag{11}$$

In the formula, T_h is the temperature of high temperature CSS medium ; $T_{t,\,out}$ is the outlet temperature of the expander.

The inlet temperature of expander is :

$$T_{\mathrm{t,in}} = \varepsilon(T_{\mathrm{s,hot}} - T_{\mathrm{t,f}}) + T_{\mathrm{t,f}} \tag{12}$$

Here $T_{\mathrm{s,\,hot}}$ is the temperature of hot oil tank.

The heat absorbed by unit mass air in the auxiliary heat exchanger is :

$$q_s = c(T_{\mathrm{t,f}} - T_{\mathrm{t,out}}) \tag{13}$$

The heat absorbed by unit mass air in the heat exchanger of UWFCAES system is :

$$q_{\mathrm{t}} = c(T_{\mathrm{t,in}} - T_{\mathrm{t,f}}) \tag{14}$$

When there is no external heat source, the air released from the air storage chamber only flows into the heat exchanger of the UWFCAES system to absorb heat.

The relationship between the pressure ratio and temperature of the air storage room and the time is :

$$\frac{\mathrm{d}\beta}{\mathrm{d}t} = \frac{h_{\mathrm{c}}A(T_{\mathrm{rw}} - T) - c_p T_{\mathrm{in}} \dot{m}_t}{c_v p_0 V} R_{\mathrm{g}} \tag{15}$$

$$\frac{\mathrm{d}T}{\mathrm{d}t} = \frac{h_{\mathrm{c}}A(T_{\mathrm{rw}} - T) + (c_v - c_p)\dot{m}_t}{c_v p_0 V \beta} R_{\mathrm{g}} T \tag{16}$$

Expander work : air flows into the expander to do work, expansion work W_t is :

$$W_t = c_p(T_{t,in} - T_{t,\mathrm{out}}) \tag{17}$$

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2.3 Performance Index

2.3.1 Round-Cycle Efficiency

The round-cycle efficiency reflects the ratio of the expansion work W_t output by the system to the input energy of the system during the whole cycle. The system energy input consists of three parts :

The electric energy consumed by the compression process W_c ; the heat energy provided by the solar collector part is equivalent to electric energy after conversion, which can be expressed by the effective input of solar energy $\eta_s Q_s / \eta_j$, where η_s is the power generation efficiency of the solar collector system and η_j is the collector efficiency. The electric energy directly consumed by the electric heating unit is W_e, so the round-cycle efficiency is :

$$\eta_{cyc} = \frac{W_t}{W_c + \frac{\eta_s Q_s}{\eta_j} + W_e} \times 100\% \tag{18}$$

2.3.2 Electric-Electric efficiency

The electric-electric efficiency focuses on the evaluation of the conversion of the pure electric energy part of the system, that is, the compressor and the electric auxiliary heat unit are both electric energy input. The efficiency of converting this part of the electric energy into the output electric power of the expander is as follows:

$$\eta_{ee} = \frac{W_t}{W_c + W_e} \times 100\% \tag{19}$$

2.3.3 Energy Storage Density

The energy storage density represents the effective expansion work stored in the unit air storage volume. For the constant pressure air storage process, since the volume V of the air storage chamber is variable, when the whole cycle ends, the energy storage density is defined as the ratio of the output power of the expander to the volume of the air storage chamber :

$$ESD = \frac{W_t}{V_m} \tag{20}$$

Here, Vm is the volume of the air storage chamber at the end of charging.

3. Comparative Analysis of Thermodynamic Performance of Three Systems

Based on previous studies, the addition method of auxiliary heat system is that air flows through the auxiliary heat storage subsystem first and then flows through the energy storage and heat storage subsystem. Therefore, the coupling system with external heat source involved in this paper is based on this method. This paper uses Matlab software to compile relevant programs, and then compares and analyzes the three systems of UWFCAES, UWFCAES + CSS and UWFCAES + CSS + EH. The basic parameters of the system operation are shown in Table 1.

The initial temperature of the cold storage tank is set as the ambient temperature, and the maximum volume of all flexible airbags is regarded as the end of charging state. When the volume of all flexible airbags is 0, the energy release is regarded as the end state. Ignoring the interval between the two cycles, the temperature of the cold tank at the end of the last cycle is used as the starting temperature of the next cycle. With the increase of the number of cycles and the adjustment of the operation mode, the parameters of the system will change accordingly.

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Table 1. The basic parameters of system operation

parameter	numerical value
Rated compression power P_{CAESC} / MW	80
Rated expansion power P_{CAESG} / MW	100
Compressor rated efficiency η_c	0.92
Compressor rated efficiency η_g	0.87
Environmental pressure P_0 / Pa	10^5
The wall temperature of air storage chamber T_{rw} / K	293
Convective heat transfer coefficient hc / (W· m^2)·K^{-1}	25
The total maximum air storage volume of the flexible airbag group V_m / m3.	1.5×10^5
Heat exchanger efficiency ε	0.75
The initial temperature of the cold tank $T_{s\text{-}cold}$ / K	293
Energy storage interval Δt / h	4
Solar collector efficiency η_j	0.65
Insufficient illumination CSS medium temperature $T_{h\ /K}$	300
The temperature of electric heating CSS medium $T_{h\ /K}$	400

As shown in Figure 2, as the number of cycles increases, the temperature of the cold tank and the hot tank gradually increases and tends to be stable after a certain number of cycles. These phenomena are mainly caused by the absorption or release of heat by the heat storage medium of each heat storage subsystem during the energy storage or release stage.

Specifically, in the UWFCAES system, the temperature of the cold and hot tanks is stabilized after about 10 cycles ; in the UWFCAES + CSS system, the temperature is stable after about 9 cycles ; in the UWFCAES + CSS + EH system, the temperature is stable after about 8 cycles, and the overall temperature level of the cold and hot tanks of the UWFCAES + CSS + EH system is higher than that of the other two systems except that the temperature of the hot tanks of the three systems is equal after the first cycle.

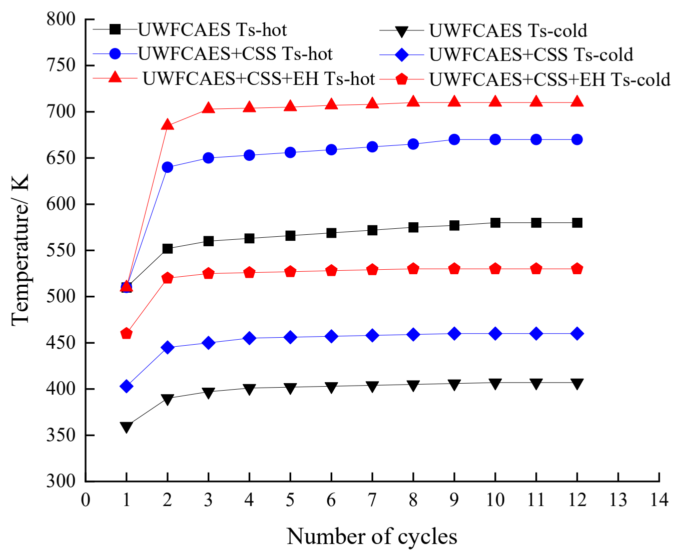

Figure 2. The temperature of cold / hot tank changes with the number of cycles

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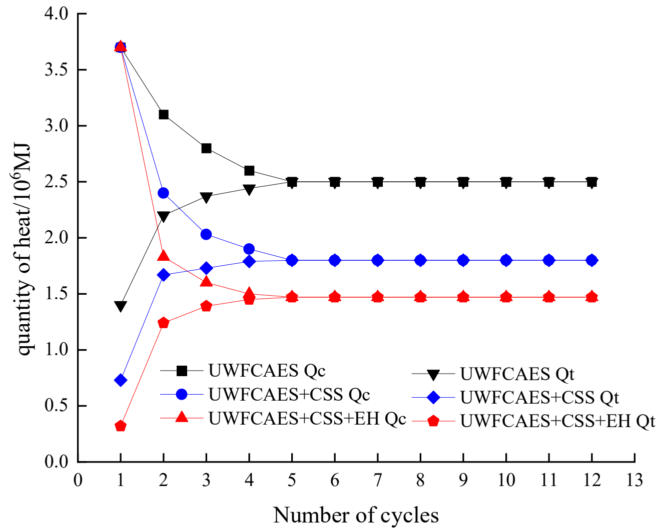
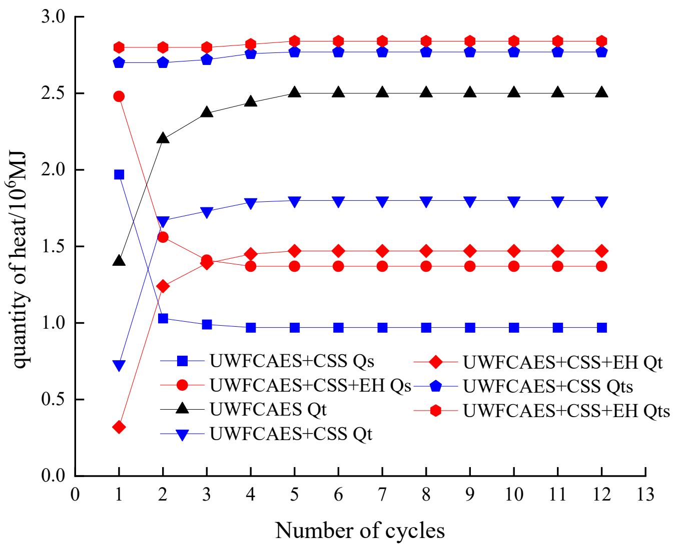
As shown in Figure 3 (a), with the increase of the number of cycles, the heat absorbed by the energy storage and heat storage subsystem (i.e., the heat released by the air Qc) gradually decreases, while the heat released by the heat storage subsystem in the energy release stage (i.e., the heat absorbed by the air Qt) gradually increases, and finally the two reach equilibrium and no longer change. There are two main reasons :

1) Although the outlet temperature of the compressor remains basically constant, the temperature of the cold tank gradually increases with the increase of the number of cycles. When the efficiency of the heat exchanger is constant, the increase of the temperature of the cold tank will reduce the temperature difference in the heat exchanger, resulting in the gradual decrease of the heat absorption Qc in the energy storage stage.

2) Due to the small change of air temperature in the underwater flexible airbag, whether the external heat source is introduced or not, the increase of the temperature of the hot tank will lead to the gradual increase of the heat Qt in the energy release stage. When the heat absorption Qc in the energy storage stage is equal to the heat released Qt in the energy release stage, the system enters a stable cycle. After that, the performance parameters will no longer change with the number of cycles.



(a) Heat absorption / release of compression regenerative system



(b) The expansion end of the air heat absorption

Figure 3. Heat changes with the number of cycles

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By comparing the three systems, it can be found that the Qc and Qt of UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH systems decrease in turn except that the Qc of the first cycle end system is equal. The reasons are as follows :

Taking UWFCAES + CSS + EH as an example, since the CSS and electric heating of the first cycle are not involved in the energy storage stage, the Qc at the end of the cycle is equal to the Qc of UWFCAES, and the UWFCAES + CSS system is the same. In the rest of the cycle, because the temperature of the cold tank of the three systems : UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH increases in turn, the effect of cooling compressed air is weakened in turn, so Qc decreases in turn. Although the hot tank temperature of UWFCAES + CSS + EH and UWFCAES + CSS is higher than that of UWFCAES, the air of the first two expansion power generation is preheated by the auxiliary heat system, and the temperature is much higher than the outlet temperature of the flexible air reservoir, so their Qt is smaller than that of UWFCAES system, and because the air temperature of the UWFCAES + CSS + EH system after auxiliary heat is higher, its Qt is further reduced.

As shown in Figure 3 (b), for the heat absorption of the air at the expansion end, although the Qt of the three systems of UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH decreases in turn, due to the external heat source of the UWFCAES + CSS and UWFCAES + CSS + EH systems, the air during the expansion power generation needs to be heat-exchanged twice before entering the expander, so the expansion power generation of the three systems is the total heat absorption Qts of the air (When there is no external heat source, it is Qt.) instead increases in turn.

As shown in Figure 4, in the first few cycles, the energy storage time t_1 of the three systems gradually decreases with the increase of the number of cycles. After the cycle reaches a stable state, its value tends to be stable and no longer changes. However, the energy release time t_2 of the three systems is almost unchanged. The reason is shown in Figure 3. The heat Qc absorbed by the compression regenerative system gradually decreases, which increases the temperature of the air entering the underwater flexible airbag. Due to the fixed underwater water depth, the air storage is always in a constant pressure state, so that the pressure ratio β of the air storage chamber remains constant at 1, and the maximum volume of the underwater flexible airbag is constant. Therefore, the higher the air temperature entering the airbag, the less the total mass of the stored air, the shorter the energy storage time, that is, the smaller the t_1 ; On the other hand, due to the decrease of the total mass of the compressed air, the expansion power generation time t_2 will decrease, but due to the increase of the temperature of the hot tank, the density of the air at the expansion end will decrease, resulting in the increase of t_2.

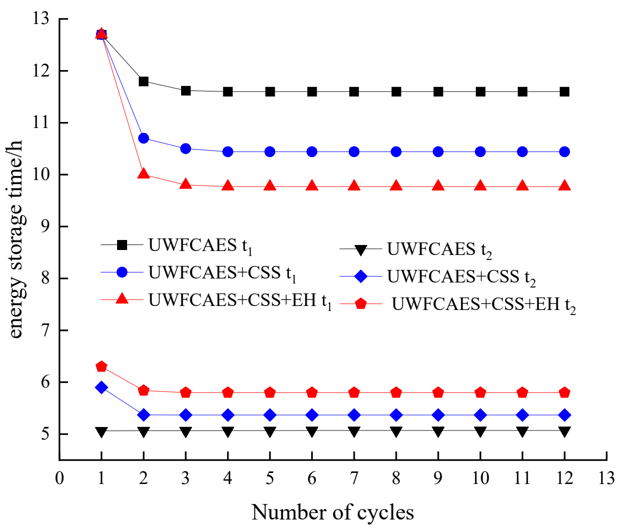

Figure 4. The energy storage / release time changes with the number of cycles

Comparing the three systems, it is found that the external heat source of the first cycle does not participate in the energy storage stage so that the energy storage time of the first cycle is equal to t_1. In other cases, the energy storage time t_1 of UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH decreases in turn. This is because the Qc of the above three systems decreases in turn except for the first cycle, so that the temperature of the compressed air entering the airbag increases in turn, so that the density of the air decreases in turn. When the maximum volume of the airbag is constant, the total mass of the air decreases in turn, so the energy storage time t_1 decreases in turn. The decrease of the total mass of the air will also reduce the energy release time t_2 of the above three systems. However, as shown in figure 3 (b), the total heat absorption Qts of the three systems at the expansion end increase in turn, so that the density of the air before entering the expander decreases in turn, which will make the energy release time t_2 increase in turn. Under the combined effect, the t_2 of the above three systems increases in turn.

As shown in Figure 5, the change curve of the work of the compressor and the expander of the system is shown. Under the condition that the compression / expansion power is constant, the work of the compressor and the expander of the system is linearly related to their respective energy storage time, so the trend change is the same as that in Figure 4.

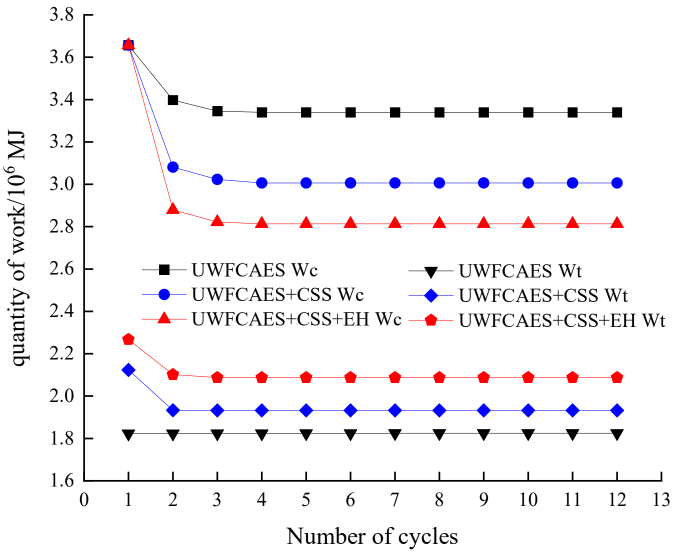

Figure 5. The work of compressor / expander varies with the number of cycles

As shown in Figure 6 (a), the variation curves of the round-cycle efficiency and the electro-electric conversion efficiency of the three systems show that the trend changes of the round-cycle efficiency and the electro-electric conversion efficiency are basically the same. The reason is as shown in Figure 5, before the system is stable, the work done by the compressor is decreasing, while the work done by the expander remains basically unchanged. At the same time, whether the effective input quantity Qs in the round-cycle efficiency is converted into electric energy, or the input electric energy We in the two efficiencies is one order of magnitude smaller than Wc and Wt, which has little effect on the trend change, so the round-cycle efficiency and the electric-electric conversion efficiency increase with the increase of the number of cycles.

For the round-cycle efficiency, compared with the three systems, it is found that the round-cycle efficiency of the three systems in the first cycle : UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH is reduced in turn, and the gap is not large. The round-cycle efficiency of the three systems under the other cycles is increased in turn. This is mainly because the compression work of the three systems is equal in the first cycle, as shown in Figure 5.Although the expansion work of the three systems increases in turn, the difference is not large, while UWFCAES + CSS and UWFCAES + CSS

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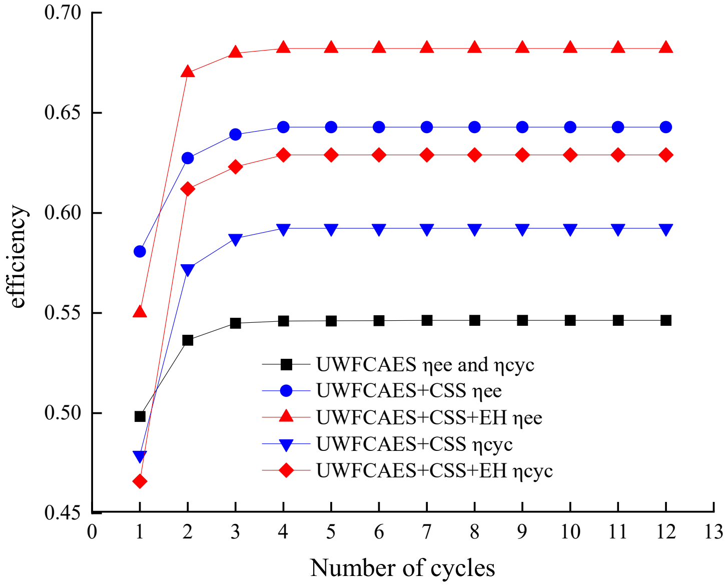
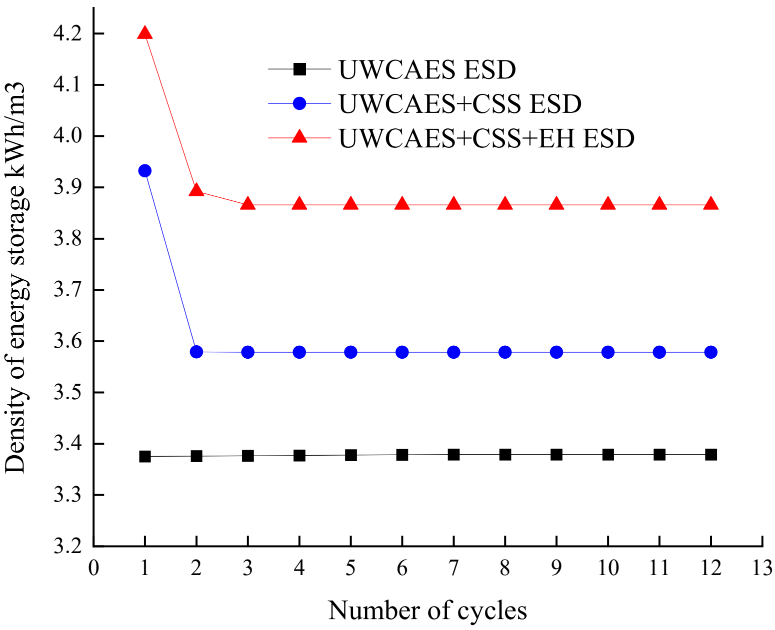
+ EH are introduced due to external heat sources, resulting in a decrease in the round-cycle efficiency, and the combined effect leads to a decrease in the round-cycle efficiency in turn. In the other cases, the compression work becomes the main factor determining the round-cycle efficiency, which makes the round-cycle efficiency of the three systems increase in turn.

For the electro-electric conversion efficiency, comparing the three systems, it is found that the first cycle is due to the equal work of the compressor, but the UWFCAES + CSS + EH introduces an external power supply, resulting in a slightly lower electro-electric conversion efficiency in the first cycle. In addition to the first cycle, the compression work has become the main factor determining the electric-electric conversion efficiency, so the electric-electric conversion efficiency of UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH is improved in turn.



(a) System efficiency



(b) Density of energy storage

Figure 6. The change of system evaluation index with the number of cycles

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As shown in Figure 6 (b) and Figure 4, the change trend of energy storage density of the three systems is exactly the same as that of expansion work. This is because the maximum air storage volume V_m of the flexible airbag is constant, so the energy storage density and the work of the expander are linear.

From the analysis of the above stable operating conditions, it can be seen that compared with UWFCAES, UWFCAES + CSS improves the round-cycle efficiency of the system by 8.41 %, the electric-electric conversion efficiency by 17.66 %, and the energy storage density by 5.9 %. Compared with UWFCAES + CSS, the round-cycle efficiency of the system is further improved by 6.2 %, the electric-electric conversion efficiency is increased by 6.1 %, and the energy storage density is increased by 8.03 %.

4. Key Factors on Systemic

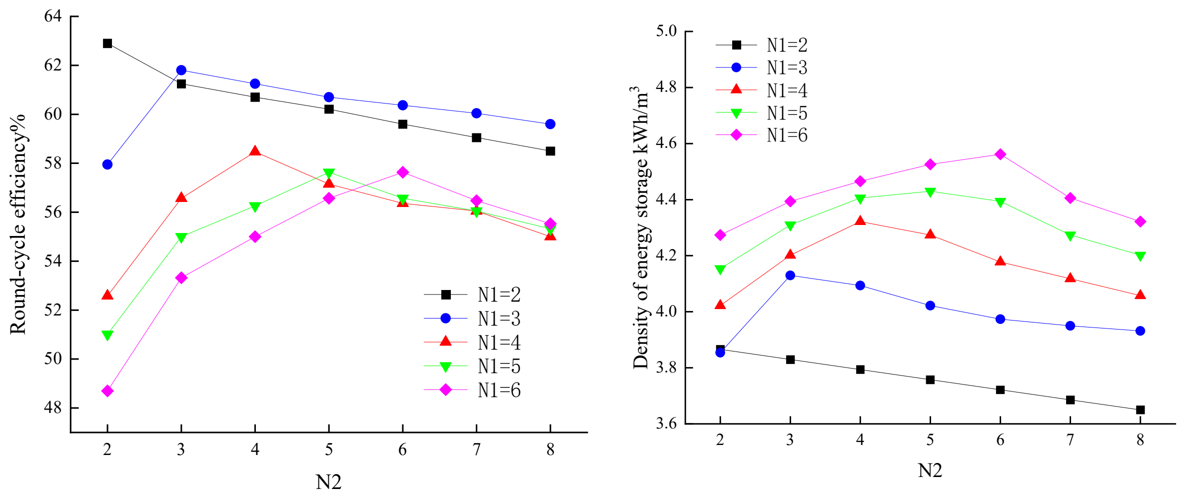
In order to further study the performance of UWFCAES + CSS + EH system, the compressor stage N1, the expander stage N2 and the convective heat transfer coefficient hc will be further studied, and the research is based on the process after the system cycle is stable. Because the change trend of the round-cycle efficiency and electric-electric conversion efficiency is basically the same, this paper only analyzes the influence of the above key parameters on the round-cycle efficiency and energy storage density of UWFCAES + CSS + EH system.

4.1 Compressor and Expander Stages

The series N here refers to the number of times air flows through a single compressor or expander. If the UWFCAES + CSS + EH system adopts a single-stage compressor or a single-stage expander, the pressure ratio during the cycle will be too high, resulting in the working temperature of the compressor and expander may exceed the tolerance limit of the material. Therefore, the compressor and expander stages studied in this paper are greater than 1, and different N1 and N2 combinations are set up to explore their effects on system performance.

As shown in Figure 7 (a) and 7 (b), for the hybrid underwater compressed air energy storage system, no matter how N1 and N2 are combined, when N1 is equal to N2, the round-cycle efficiency and energy storage density are the highest, and the smaller the difference, the better the system performance. The reason is that when N1 is greater than N2, the compression ratio of the single-stage compressor is less than the expansion ratio of the single-stage expander, resulting in the ' excess ' energy release stage of the heat storage medium in the energy storage stage, and some of the heat storage medium is not utilized in the energy release stage, so that the system performance is greatly reduced. When N1 is less than N2, taking N1 as an example, although the larger the N2 is, the smaller the range of single-stage expansion ratio is, and the closer the expander is to the rated operating condition, the larger the N2 is, the more the ' amount ' of air in the single-stage heat exchanger in the energy release stage is more than the ' amount ' of the heat storage medium, that is, the larger the heat transfer temperature difference in the heat exchanger, the greater the irreversible loss, and the larger the expansion series, the greater the pressure loss of air in the heat exchanger, and the lower the round-cycle efficiency due to the combined effect. Although the larger the N2 is, the closer the expander is to the rated operating condition, the ' quantity ' of the air in the energy release stage is more than the ' quantity ' of the heat storage medium, so that the air parameters before entering the expander are smaller. The combined effect reduces the expansion work and the energy storage density.

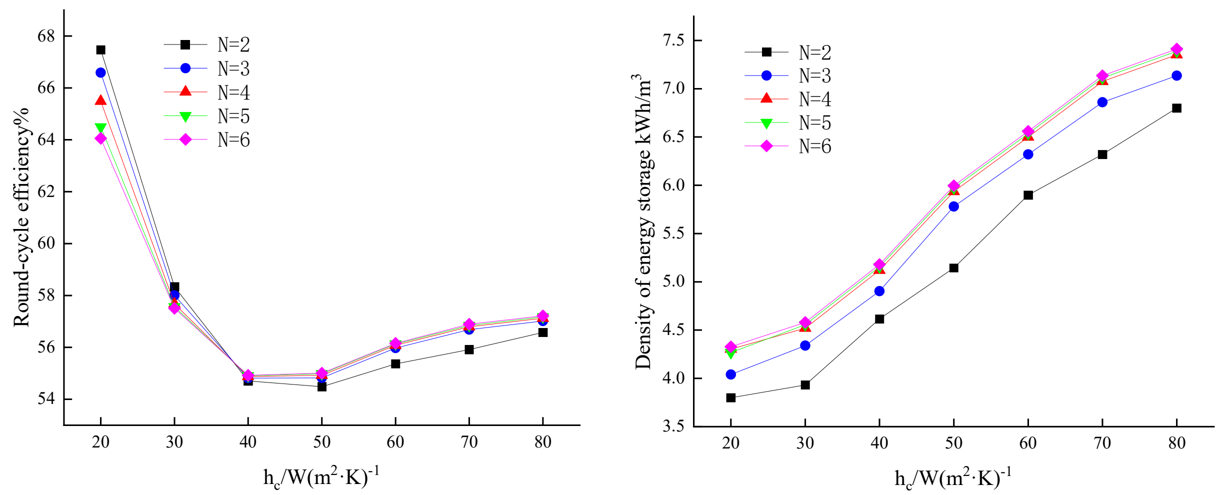
From Figure 7 (a) and Figure 7 (b), it can also be seen intuitively that within the parameter range studied, when N1 = N2 = N, the larger the N, the smaller the round-cycle efficiency and the larger the energy storage density. The reason is that under the condition of other parameters unchanged, the influence of the increase of the system stage on the system performance is mainly determined by the following three aspects : 1) The smaller the range of single-stage pressure ratio change, the closer the system is to the rated operating condition ; 2) With the increase of heat transfer times, the pressure loss of air in each heat exchanger increases. 3) When the single-stage pressure ratio decreases, the outlet air temperature of the compressor decreases.



(a) Round-cycle efficiency **(b)** Density of energy storage

Figure 7. The influence of compressor / expander stages on the system

4.2 Coefficient of Convective Heat Transfer



(a) Round-cycle efficiency **(b)** Density of energy storage

Figure 8. The influence of convective heat transfer coefficient on the system

For the constant wall flexible air storage chamber model, during the cycle process, the heat exchange between the air storage room air and the wall surface will occur, and the convective heat transfer coefficient hc will directly affect the heat transfer, thus affecting the system performance. The research shows that when the compression series and the expansion series are equal, the system round-cycle efficiency and energy storage density are the highest, so the following research is based on the two series equivalence. As shown in Figure 8 (a), as hc increases, the round-cycle efficiency first decreases significantly and then increases slowly. The calculation shows that when N = 6, hc = 40, the round-cycle efficiency is 54.93 %, which is only 0.22 % higher than the lowest round-cycle efficiency of 54.71 %. It can be seen from Fig.8 (b) that the energy storage density continues to increase. The reason is that the larger the hc is, the stronger the heat transfer between the indoor air and the wall is, the more the heat loss is, and the lower the round-cycle efficiency is. However, when hc is too large, the air expands rapidly in the energy release stage. In the later stage of expansion, the air temperature is lower than the wall temperature, and it no longer dissipates heat to the wall. On the contrary, a part of heat is absorbed from the wall, which increases the round-cycle efficiency. The larger the hc, the lower the indoor air temperature at the end of the energy storage, the more the air quality stored, the greater the expansion work, and the greater the energy storage density. It can also be seen from Figure 8 (a) and (b) that when hc is small, the larger the N is, the lower the round-cycle efficiency is. When hc is large, the larger the N is, the higher the round-cycle efficiency is. The energy storage density increases with the increase of N, but the larger the N is, the smaller the effect of the same hc on the system performance is. For example, the round-cycle efficiency and energy storage density curves of N = 4, N = 5 and N = 6 are basically coincident. The reason is determined by the three factors that affect the performance of the system.

5. Conclusion

In this paper, a constant pressure air storage model of composite underwater compressed air energy storage coupled with CSSand EH is established. MATLAB is used to simulate and analyze the thermodynamic performance of UWFCAES, UWFCAES + CSS, UWFCAES + CSS + EH systems. At the same time, the series of compressors , expanders and the convective heat transfer coefficient are analyzed. The influence of the performance of the UWFCAES + CSS + EH system can be concluded by analysis :

(1) Compared with the basic UWFCAES system, the round-cycle efficiency and energy storage density of the UWFCAES + CSS system are significantly improved after the introduction of CSS. The round-cycle efficiency of the system is increased by 8.41 %, and the electric-electric conversion efficiency is increased by 17.66 %. In addition, after the introduction of EH, compared with UWFCAES + CSS, the round-cycle efficiency of UWFCAES + CSS + EH is further improved by 6.2 %, the electric-electric conversion efficiency is increased by 6.1 %, and the energy storage density is increased by 8.03 %.

(2) For the UWFCAES + CSS + EH, regardless of how the compression series N1 and the expansion series N2 are set, when the two are equal, the round-cycle efficiency and energy storage density perform best, and the smaller the gap between the two, the better the overall performance of the system.

(3) With the increase of the convective heat transfer coefficient, the round-cycle efficiency first decreases significantly, and then gradually rises, while the energy storage density continues to increase. When hc is small, increasing N will lead to a decrease in the round-cycle efficiency ; when hc is large, the situation is opposite ; the energy storage density increases with the increase of N, but with the further increase of N, the influence of hc on the system performance is gradually weakened.

It should be noted that this paper only analyzes the key thermodynamic parameters of the composite underwater flexible compressed air energy storage system coupled with CSS and EH. How to reasonably output UWFCAES + CSS + EH in the optimal operation and scheduling of the hydro-solar complementary system, and the variable working condition output characteristics of UWFCAES + CSS + EH when considering the change of water level are the problems that need to be further studied in the next step.

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