INTRODUCTION

Fossil fuels have played an important role in the world energy structure for a long time. But with the excessive consumption of fossil energy, the increasingly prominent environmental pollution problem has become worse and worse. The solar energy has the largest reserve among all kinds of renewable energy resources, and its large-scale and efficient utilization...
has become an inevitable requirement of adjusting the world's energy structure and realizing the sustainable energy development. The solar thermal power generation (STPG) is considered one of the most suitable emerging renewable energy technologies. Usually, the most common way is to use the large-scale solar heliostat field and produce the high-temperature steam to drive a steam turbine to produce the electricity. The International Energy Agency (IEA) estimates that by 2050, the solar energy is likely to become the world's largest energy source of electricity production.3

However, the independent STPG system has some inherent problems such as high cost, instability, and lower thermal efficiency. In order to solve these problems, it is proposed to integrate the solar energy into traditional fossil fuel-fired power generation systems, which not only solves the technical bottleneck of instability of solar energy utilization and high cost, but also improves the thermal efficiency of the traditional fossil fuel power generation system. At present, it has become the choice of the large-scale solar energy utilization.

The integrated solar combined cycle (ISCC) system was originally proposed by Johansson et al. At present, the ISCC system has successfully been demonstrated and operated around the world. The Iranian established the world's first ISCC system. Other ISCC systems are either under construction like Agua Prieta II power station or in the planning phase including Palmdale, Victorville2, Abdaliya power station.7

Now, the parabolic trough solar energy collector technology is the most mature and has been applied in several operated ISCC power stations. Bag hernejad and Yaghoubi8 analyzed the exergoeconomic performance of the ISCC system by using the genetic algorithm and studied the impact of power unit cost on important system parameters by using the sensitivity analysis method. Bag hernejad et al.9,10 studied the performance of ISCC plant by using both the energy and exergy analysis methods, and explained the process of exergy loss from the perspective of thermodynamics and the relationship between the leveled cost of electricity (LCOE) and the solar energy integration mode. Liu et al11 used the least-squares method to model a trough solar collector and analyzed the relationships among the trough solar collector efficiency, the heat transfer fluid (HTF) inlet temperature, the HTF mass flow rate, and the direct normal irradiance (DNI). The research results of Giovanni and Sergio12 show that different solar integration modes have different effects on the ISCC system performance. Popov13 made the performance analysis of the ISCC system and the research results showed that under sufficient DNI conditions, the system could obtain the best generation efficiency when the high-pressure feedwater heater was replaced. The research results from Zhong and Chen14 showed that under a certain DNI, the collecting heat efficiency was inversely proportional to the HTF working temperature.

Though there are many researches on ISCC system, the volatility of solar energy usually results in low system thermal efficiency when no or low DNI is available. In order to maximize the utilization of solar energy, this paper proposes a new optimization operation strategy of the ISCC system with changeable integration mode under different DNI conditions. The model of ISCC system is built by using the EBSILON software. The new operation optimization strategy is applied to the traditional ISCC system. This paper deeply analyzes both the thermodynamic and economic performance of ISCC system using the new operation optimization strategy under variable DNI conditions.

2 | SYSTEM DESCRIPTION

2.1 | The traditional ISCC system

The flowchart of the traditional ISCC system is shown in Figure 1. The traditional ISCC system uses the HTF technology to integrate the solar energy with the high-pressure boiler (HPB) section of HRSG. The oil-water heat exchanger uses a shell and tube heat exchanger. The high-temperature oil flows inside the tube, and water/steam flows outside the tube. The oil-water heat exchanger is divided into two parts, which includes the preheating process and the boiling process in this paper. The feeding water out from the high-pressure economizer 2 (HPE2) is divided into two parts, one part flows into the HPB, another part flows into the oil-water heat exchanger to absorb the heat energy of the HTF to be the saturated steam; finally these two parts saturated steam mix and enter into the high-pressure superheater (HPS).

Figure 2 shows the seasonal DNI distributions of Dunhuang. As shown in Figure 2, the DNI data with lower values occupy a large proportion. The DNI with lower values usually does not meet the parameter requirements such as the HTF temperature and mass flow, if the solar thermal energy is integrated into the HP section, or even if the DNI with lower values can meet the parameter requirements, due to the limitation of the collecting heat characteristics of solar collectors, the collecting heat efficiency will be very low. So, for the traditional ISCC system, the annual utilization rate of the solar energy is very low. Hence, an optimization operation strategy of the ISCC system with changeable integration mode under different DNI conditions is proposed.

2.2 | The ISCC system with new optimization strategy

In order to overcome the disadvantage of the traditional ISCC system with lower solar energy utilization rate, this paper proposes an optimization operation strategy of the ISCC system with changeable integration mode under
different DNI conditions on the basis of the simultaneous considerations of both the solar collector field collecting efficiency and ISCC system thermal efficiency.

Figure 3 shows the ISCC system with the new optimization strategy of changeable integration mode. The change of the solar energy integration mode can be realized by the control and switch of valves. For example, the ISCC system will be integrated into the HPB section by opening valves b1, b2 and closing valves a1, a2. The high-pressure feedwater is preheated and boiled in the oil-water heat exchanger. The oil-water heat exchanger is divided into two parts, which includes both the preheating process and the boiling process. The heat exchanger area of the boiling process is 7000 m². The heat exchanger area of the preheating process is 1450 m². Similarly, the ISCC system will be integrated into the HPE2 section by opening valves a1, a2 and closing valves b1, b2. As shown in Figure 3, in this case, the heat exchanger for the boiling is in fact also a heat exchanger for preheating due to the lower DNI. The high-pressure feedwater is only preheated in oil-water heat exchanger without the boiling
process. Therefore, through adding the valve switching system, the solar energy integration mode of ISCC system can be changed under different DNI values, which can achieve the optimal operation of ISCC system.

3 | MATHEMATICAL MODEL AND SYSTEM PERFORMANCE EVALUATION

3.1 | Solar collector subsystem

The LS-2 type parabolic trough collector is applied in this paper. The trough solar collector field model refers to the 30 MW SEGS V1 solar power plant where Therminol VP-1 is selected as the thermal conduction oil. Table 1 shows the trough solar collector field parameters.

The solar thermal energy inputted to the ISCC system ($Q_{sol}$) is calculated as follows:

$$Q_{sol} = DNI \times A_{surface} \times \eta = m_{oil} \cdot c \cdot (T_{col,o} - T_{col,i})$$

(1)

where, $m$ is the mass flow rate; $A_{surface}$ is the trough solar collector reflection area; $\eta$ is the collecting heat efficiency; $c$ is the specific heat capacity; $T_{col}$ is the collector temperature. Subscripts: o stands for the outlet; i stands for the inlet; oil stands for the thermal conduction oil.

The collecting heat efficiency ($\eta$) is calculated as follows:

$$\eta = k_1 (73.3 - 0007276 \Delta T) - 0.496 (\Delta T/I) - 0.0691 (\Delta T^2/I)$$

(2)

where, $k_1$ is the incident angle correction; $\Delta T$ is the temperature difference between the collector working temperature and the ambient temperature, $\Delta T = (T_{col,o}\cdot T_{col,i})/2 - T_a$.

When $k$ is equal to 1, the variation curves of the $\eta$ with $\Delta T$ under different DNI values are as shown in Figure 4. When the $\Delta T$ is constant, $\eta$ decreases as DNI decreases. When the DNI is constant, $\eta$ decreases as $\Delta T$ increases. However, the integration mode of ISCC system has a great influence on $\Delta T$, an operation optimization strategy is proposed in this paper.

The heat collected by the solar field is introduced to the GTCC system through the oil-water heat exchanger. The basic heat transfer equation can be expressed as follows:

FIGURE 3 The traditional integrated solar combined cycle system with changeable integration mode
3.2 | GTCC system model

3.2.1 | Gas turbine modeling

The following expansion processes and blade cooling model are assumed to simplify the thermodynamic calculation. Main equations for the required cooling air flow fraction are as follows:

\[
\begin{align*}
\epsilon_c &= \frac{(T_m - T_{\text{met,ext}})}{(T_m - T_c)} \\
B_1 &= B_{\text{tbc}} \cdot (1 - \epsilon_c) / (1 - \epsilon_c) \cdot B_{\text{met}} \\
\phi &= \frac{m_a}{m_g} = \frac{k_{\text{cool}}}{(1 + B_1)} \left( \epsilon_c - \epsilon_f \right) / \left( 1 - \eta_{\text{inc}} \left( 1 - \epsilon_c \right) \right) / \left( 1 - \eta_{\text{inc}} \left( 1 - \epsilon_f \right) \right)
\end{align*}
\]

where, both \( T_c \) and \( T_m \) are the temperatures of the inlet cooling air and the mainstream gas, respectively; \( T_{\text{met,ext}} \) stands for the allowable temperature of the metal surface; \( B_{\text{tbc}} \) stands for the metal Biot number; \( B_{\text{met}} \) stands for the thermal barrier coating Biot number; \( m_a \) is the mass flow rate of the cooling air; \( m_g \) is the mass flow rate of the mainstream gas; \( k_{\text{cool}} \) stands for the cooling flow factor; \( \eta_{\text{inc}} \) is the efficiency of the internal cooling; \( \epsilon_f \) is the film cooling coefficient; \( \epsilon_c \) is the blade cooling coefficient; \( \phi \) is the cooling air flow fraction.

The flow equation proposed by Flieger is usually used to calculate the variation of turbine parameters under variable working conditions. In this paper, the improved Flieger formula is used as follows:

\[
\sigma \frac{m_{\text{in}}}{Ap_{\text{in}}} = \gamma \left( 2 + \frac{\gamma}{\gamma + 1} \right) \frac{1}{\sqrt{\gamma}}
\]

where, \( A \) is the turbine inlet area; \( \sigma \) is a constant; \( p \) is the pressure; \( m \) is the mass flow. Subscripts: \( t \) stands for the gas turbine; \( i \) stands for the inlet.

The each stage turbine efficiency is corrected with the following Equation under off-design conditions:

\[
\eta_i = \frac{\eta_{\text{opt}}}{\eta_{\text{a}}} = \eta_{\text{a}} \sqrt{\frac{T_{30}}{T_3}} \left( 1 - \frac{\delta_{\text{opt}}}{\delta} \right) \left( Y - (Y - 1) \frac{T_{30}}{T_3} \left( 1 - \frac{\delta_{\text{opt}}}{\delta} \right) \right)
\]

where, \( \delta \) is the expansion ratio; \( \eta_{\text{a}} \) is the each stage turbine efficiency; \( n \) is the rotational speed, r/min; \( \gamma \) is the specific heat ratio; \( Y \) is the constant (\( Y = 2.083 \) in this paper). Subscripts: \( \text{opt} \) stands for the maximum efficiency points; \( a \) stands for the inlet.

3.2.2 | HRSG modeling

The off-design models are built based on the heat transfer and energy balance equations. The detailed equations for different heating surfaces are described as follows:

Energy balance equation is as follows:

\[
Q = m_g c_p \left( t_{g,\text{in}} - t_{g,\text{out}} \right) = m_s \left( h_{s,\text{out}} - t_{s,\text{in}} \right)
\]

Heat transfer equation is as follows:
$Q = (UA)_p \Delta t \quad (11)$

$$\Delta t = \frac{(t_{g,\text{out}} - t_{s,\text{in}}) - (t_{g,\text{in}} - t_{s,\text{out}})}{\ln \left( \frac{(t_{g,\text{out}} - t_{s,\text{in}})}{(t_{g,\text{in}} - t_{s,\text{out}})} \right)} \quad (12)$$

The equation for superheater and re heater is as follows:

$$(UA)_p = m_g^{0.65} F_g K \left( \frac{m_s}{m_{s,d}} \right)^{0.15} \quad (13)$$

The equations for economizer and evaporator are as follows:

$$(UA)_p = m_g^{0.65} F_g K_1 \quad (14)$$

$$K_1 = Q_d \left( \frac{\Delta t_d m_g^{0.65} F_{g,d}}{m_{s,d}} \right) \quad (15)$$

$$F_g = c_p^{0.33} k^{0.67} / \mu^{0.32} \quad (16)$$

where, $m_g$ is gas mass flow; $c_p$ is the specific heat capacity of gas at constant pressure; $t_g$ is gas temperature; $m_s$ is water/steam mass flow; $h_s$ is water/steam enthalpy; $\Delta t$ is the log-mean temperature difference; $(UA)_p$ is the product of the heat transferring area and overall heat transfer coefficient; $F_g$ is the gas physical property coefficient; $K_1$ is the design factor; $k$ is the gas thermal conductivity; $\mu$ is the gas dynamic viscosity. Subscript: d stands for the design condition, s stands for the water/steam.

### 3.2.3 Steam turbine modeling

Main equations of the steam turbine modeling are described as follows:\(^{25}\):

The equation for the low-pressure cylinder (critical condition) is as follows:

$$\frac{m_s}{m_{s,d}} = \frac{p_m}{p_{m,d}} \sqrt{\frac{T_{in,d}}{T_{in}}} \quad (17)$$

The equation for the high-/intermediate-pressure cylinder (subcritical condition) is as follows:

$$\frac{m_s}{m_{s,d}} = \sqrt{\frac{p_{in}^2 - p_{\text{out}}^2}{p_{in,d}^2 - p_{\text{out,d}}^2}} \sqrt{\frac{T_{in,d}}{T_{in}}} \quad (18)$$

The turbine efficiency is expressed by the following Equation\(^{25}\):

$$\eta_{\text{reduction}} = 0.191 - 0.409 \left( \frac{m}{m_d} \right) + 0.218 \left( \frac{m}{m_d} \right)^2 \quad (19)$$

$$\eta_{\text{turbine}} = (1 - \eta_{\text{reduction}}) \eta_{\text{turbine,d}} \quad (20)$$

where, $\eta_{\text{reduction}}$ is the percent reduction in efficiency; both $\eta_{\text{turbine}}$ and $\eta_{\text{turbine,d}}$ are isentropic expansion efficiencies at the actual operating state and reference state; both $m$ and $m_d$ are the mass flows at operating state and reference state.

The design data of the PG9351FA gas turbine are shown in Table 2. The simulation results of the GTCC system are shown in Table 3. As shown in Table 4, the simulation results of system are compared with the data in the reference.\(^{26}\) And the relative errors compared to the data in the reference\(^{26}\) are $<2\%$ and in the allowable ranges. So, this simulation model of the system is feasible.

### 3.3 Performance evaluation criteria

Solar power generation ($W_{\text{sol,net}}$) is defined as follows:\(^{27}\):

$$W_{\text{sol,net}} = W_{\text{ISCC}} - \frac{W_{\text{GTCC}} \cdot F_{\text{fuel,ISCC}}}{F_{\text{fuel,GTCC}}} \quad (21)$$

where, $W_{\text{ISCC}}$ is the ISCC system power output; $W_{\text{GTCC}}$ is the power output of GTCC system without integration with solar energy; $F_{\text{fuel,ISCC}}$ is the fuel consumption amount by the ISCC system; $F_{\text{fuel,GTCC}}$ is the fuel consumption amount by the GTCC system.

The system thermal efficiency ($\eta_{\text{net}}$) is calculated as follows:

$$\eta_{\text{net}} = \frac{W_{\text{net}}}{Q_i + Q_{\text{sol}}} = \frac{W_{\text{net}}}{F_{\text{fuel}} \times \text{LHV} + Q_{\text{sol}}} \times 100\% \quad (22)$$

where, $W_{\text{net}}$ is the system power output; $Q_i$ is the fossil fuel energy input.

The energy grade of solar energy ($\psi$) is calculated as follows:\(^{28}\):

$$\psi = 1 - \frac{4}{3} \times \frac{T_a}{T_s} + \frac{1}{3} \times \left( \frac{T_a}{T_s} \right)^4 \quad (23)$$

where, $T_s$ is the solar surface temperature; $T_a$ is the ambient temperature.

The system exergy efficiency ($\eta_{\text{ex,net}}$) is defined as follows:\(^{13}\):

$$\eta_{\text{ex,net}} = \frac{W_{\text{net}}}{E_{\text{in}} + Q_{\text{sol}} \times \psi} = \frac{W_{\text{net}}}{1.04 \times F_{\text{fuel}} \times \text{LHV} + Q_{\text{sol}} \times \psi} \times 100\% \quad (24)$$
where, \( E_{\text{CH}_4} \) is the chemical exergy of the fuel CH\(_4\); it is assumed that the chemical exergy of CH\(_4\) is approximately 1.04 times its LHV\(_f\); LHV\(_f\) is the fuel lower heating value.

The solar photoelectric efficiency (\( \eta_{\text{sol,net}} \)) is defined as follows\(^{13} \):

\[
\eta_{\text{sol,net}} = \frac{W_{\text{sol,net}}}{DNI \times A_{\text{surface}} \times N} \times 100\% \quad (25)
\]

The solar photoelectric exergy efficiency (\( \eta_{\text{ex,sol}} \)) is defined as follows\(^{13} \):

\[
\eta_{\text{ex,sol}} = \frac{W_{\text{sol,net}}}{DNI \times A_{\text{surface}} \times N \times \eta_s} \times 100\% \quad (26)
\]

The solar energy contribution rate (\( X_{\text{sol}} \)) is defined as follows\(^{13} \):

\[
X_{\text{sol}} = \frac{Q_{\text{sol}}}{Q_{\text{fuel,ISCC}} \times \text{LHV}_f + Q_{\text{sol}}} \times 100\% \quad (27)
\]

The levelized cost of electricity (LCOE) of solar energy is calculated as follows\(^{29} \):

\[
LCOE = \frac{LC_{\text{INV}} + LC_{\text{O&M}}}{E_{\text{sol,net}}} \quad (28)
\]

\( LC_{\text{INV}} \) is defined as follows\(^{29} \):

\[
LC_{\text{INV}} = CRF \cdot \text{INV} = \left( \frac{i_{\text{eff}} (1 + i_{\text{eff}})^n}{(1 + i_{\text{eff}})^n - 1} \right) \cdot \text{INV} \quad (29)
\]

where, \( CRF \) is the capital recovery factor; \( \text{INV} \) is the total investment; \( i_{\text{eff}} \) is the effective discount rate; \( n \) is the economic life of the power plant, \( n \) is equal to 15 in this paper.

### TABLE 2

| Parameters                  | Values |
|-----------------------------|--------|
| **Compressor**              |        |
| Inlet temperature (°C)      | 15     |
| Pressure ratio              | 15.4   |
| Inlet flow (kg/s)           | 621.58 |
| **Turbine**                 |        |
| Inlet temperature (°C)      | 1327   |
| Outlet temperature (°C)     | 610    |
| **Gas turbine**             |        |
| Output power (MW)           | 255.6  |
| System thermal efficiency (%)| 36.9   |

### TABLE 3

Design and simulation data of GTCC system

| Parameters                  | Values       |
|-----------------------------|--------------|
| **Compressor**              |              |
| Pressure loss (%)           | 1            |
| Pressure ratio              | 15.4         |
| Isentropic efficiency (%)   | 87           |
| Mechanical efficiency (%)   | 98           |
| **Combustor**               |              |
| Pressure loss (%)           | 3            |
| **Gas turbine**             |              |
| Pressure loss (%)           | 1            |
| Isentropic efficiency (%)   | 90           |
| Mechanical efficiency (%)   | 98           |
| Inlet air temperature (°C)  | 1327         |
| Exhaust gas temperature (°C)| 610          |
| **Steam turbine**           |              |
| Isentropic efficiency (HP/IP/LP) (%) | 87.5/89.5/89 |
| **HRSG**                    |              |
| Pinch temperature difference (°C) | 12       |
| Approach temperature difference (°C) | 8          |
| Hot end temperature difference (°C) | 45       |
| Exhaust flue gas temperature (°C) | 86.08     |
| Steam pressure (HP/IP/LP)(MPa) | 16.5/3.5/0.4 |
| Superheated steam temperature (HP/IP/LP)(°C) | 565/350/265 |
| **Brayton cycle**           |              |
| Output power (MW)           | 255.6        |
| Thermal efficiency (%)      | 36.9         |
| **Rankine cycle**           |              |
| Output power (MW)           | 132.52       |
| **Combined cycle**          |              |
| Output power (MW)           | 388.12       |
| Thermal efficiency (%)      | 55.27        |

### TABLE 4

Comparison of thermodynamic calculation results of the GTCC system

| Parameter                                      | Value (design) | Value (simulation) |
|------------------------------------------------|----------------|--------------------|
| Compressor inlet flow (kg/s)                   | 621.58         | 621.58             |
| Gas Turbine exhaust gas temperature (°C)       | 610            | 609.5              |
| Gas Turbine exhaust flow (kg/s)                | 635.66         | 635.88             |
| Superheated steam temperature (HP/IP/LP)(°C)   | 565/350/265    | 565/350/265        |
| Superheated steam flow (HP/IP/LP) (kg/s)       | 73.02/10.97/17.01 | 72.89/11.08/16.94 |
| HRSG exhaust flue gas temperature (°C)         | 86.08          | 85.84              |
| Gas turbine output power (MW)                  | 255.6          | 254.14             |
| Steam turbine output power (MW)                | 132.52         | 133.11             |
| Gas turbine cycle efficiency (%)               | 36.90          | 36.69              |
| Combined cycle efficiency (%)                  | 55.27          | 55.67              |

### TABLE 2

The design data of PG9351FA gas turbine

| Parameters                  | Values |
|-----------------------------|--------|
| **Compressor**              |        |
| Inlet temperature (°C)      | 15     |
| Pressure ratio              | 15.4   |
| Inlet flow (kg/s)           | 621.58 |
| **Turbine**                 |        |
| Inlet temperature (°C)      | 1327   |
| Outlet temperature (°C)     | 610    |
| **Gas turbine**             |        |
| Output power (MW)           | 255.6  |
| System thermal efficiency (%)| 36.9   |

### TABLE 3

Design and simulation data of GTCC system

| Parameters                  | Values       |
|-----------------------------|--------------|
| **Compressor**              |              |
| Pressure loss (%)           | 1            |
| Pressure ratio              | 15.4         |
| Isentropic efficiency (%)   | 87           |
| Mechanical efficiency (%)   | 98           |
| **Combustor**               |              |
| Pressure loss (%)           | 3            |
| **Gas turbine**             |              |
| Pressure loss (%)           | 1            |
| Isentropic efficiency (%)   | 90           |
| Mechanical efficiency (%)   | 98           |
| Inlet air temperature (°C)  | 1327         |
| Exhaust gas temperature (°C)| 610          |
| **Steam turbine**           |              |
| Isentropic efficiency (HP/IP/LP) (%) | 87.5/89.5/89 |
| **HRSG**                    |              |
| Pinch temperature difference (°C) | 12       |
| Approach temperature difference (°C) | 8          |
| Hot end temperature difference (°C) | 45       |
| Exhaust flue gas temperature (°C) | 86.08     |
| Steam pressure (HP/IP/LP)(MPa) | 16.5/3.5/0.4 |
| Superheated steam temperature (HP/IP/LP)(°C) | 565/350/265 |
| **Brayton cycle**           |              |
| Output power (MW)           | 255.6        |
| Thermal efficiency (%)      | 36.9         |
| **Rankine cycle**           |              |
| Output power (MW)           | 132.52       |
| **Combined cycle**          |              |
| Output power (MW)           | 388.12       |
| Thermal efficiency (%)      | 55.27        |
\[ LC_{O&M} \] is defined as the following equation:

\[ LC_{O&M} = C_{0\, O&M} \cdot CELF_{O&M} = C_{0\, O&M} \cdot \frac{k_{O&M} \left( 1 - k_{O&M}^2 \right)}{1 - k_{O&M}} \cdot CRF \]  

(30)

where, \( C_0 \) is the cost; \( CELF \) is the constant-escalation levelization factor; \( r_n \) is the nominal escalation rate, \( r_i \) is the real escalation rate; \( r_{eff} \) is the average annual inflation rate. Subscripts: INV, O and M stand for the investment, the operation and maintenance, respectively.

Steam turbine output power \( (W_{ST}) \) of ISCC system is calculated as follows:

\[ W_{ST} = \Delta F_{HP} \times \Delta h_1 + (\Delta F_{IP} + \Delta F_{IP}) \times \Delta h_2 + (\Delta F_{IP} + \Delta F_{IP} + \Delta F_{LP}) \times \Delta h_3 \]  

(33)

where, \( \Delta F \) is the steam mass flow rate; \( \Delta h \) is the steam enthalpy drop. Subscripts: HP, IP, and LP stand for the high, intermediate, and low pressure, respectively. 1, 2, and 3 stand for the high-, intermediate-, and low-pressure steam cylinder, respectively.

4 | RESULT ANALYSIS AND DISCUSSION

4.1 | System performance analysis in a typical autumn day

In this paper, Dunhuang meteorological data are obtained from the TRNSYS. The meteorological data of a typical autumn day in Dunhuang are shown in Figure 5.

In the process of simulation, the time interval between every two measuring points is set as 20 minutes. Then the performances of both ISCC system with the new optimization strategy and the traditional ISCC system without the optimization strategy are simulated. The simulation adequately considers the changes of the ambient temperature, DNI, and solar azimuth and other weather conditions.

Figures 6 and 7 show the solar power generation and the total power generation of both the ISCC system with the new optimization strategy and the traditional ISCC system without the optimization strategy, respectively. Combined with Figure 5, it can be seen that when the DNI value is greater than 400 W/m², the solar collector field will be integrated into the HPB. When the DNI value is between 100 and 400 W/m², and the corresponding times are between 10:20-13:20 and 16:20-16:40, the solar collector field will be integrated into the HPE2. And both the solar power generation and total power generation of ISCC system with the new optimization strategy are greater than those of the traditional ISCC system without the optimization strategy.

Figures 8-10 show that all performance parameters of the ISCC system with the new optimization strategy are greater than those of the system without the optimization strategy. Their maximum differences are 0.29%, 6.08%, and 6.1%, respectively.

The thermodynamic performance parameters of the ISCC system with the new optimization strategy and the traditional ISCC system without the optimization strategy during the autumn typical day are shown in Table 5. It is found that all performance parameters of the ISCC system with the new optimization strategy are greater than those of the traditional ISCC system without the optimization strategy.

4.2 | Steam turbine power generation

For the traditional ISCC system without the optimization strategy, when the DNI is <400 W/m², due to the decrease of the solar collector field collecting heat efficiency, the low-level DNI cannot meet the parameter requirement if the solar energy is integrated into HPB, which results in that the solar energy with low DNI cannot be utilized adequately.

After using the changeable integration mode, the high- and intermediate-pressure feedwater mass flow rates of ISCC system are different from those of the traditional ISCC system. As shown in Figure 11, when the solar
energy integrates into HPB, as the feedwater mass flow rate of the solar collector field side increases, the total mass flow rate of the high-pressure (HP) feedwater will increase, the mass flow rate of the intermediate-pressure (IP) feedwater will decrease, and the mass flow rate of the low-pressure (LP) feedwater will decrease. However, the LP feedwater mass flow rate only has a small change in general. As can be seen from Figure 12 when the solar energy integrates into HPE2, as the feedwater mass flow rate of the solar collector field side increases, the mass flow rate of the HP feedwater will decrease, the mass flow rate of the IP feedwater will increase, and the mass flow rate of the LP feedwater will decrease slightly. Figure 13 shows the mass flow rate absolute differences of both the HP feedwater and the IP feedwater between these two integration modes. It is found that the absolute difference of the IP feedwater mass flow rate is greater than that of the
HP feedwater flow rate. When the integration mode of the ISCC system changes from the HPB to the HPE2, the increase of the HP feedwater mass flow rate is less than that of the IP feedwater mass flow rate.

According to Equation (33), the steam turbine output power of ISCC system mainly depends on the variations of the HP, IP, and LP steam mass flow rates. Although the capacity to do work of the HP steam is greater than that of the IP steam, the increase of the IP feedwater mass flow rate is greater than that of the HP feedwater mass flow rate. In the autumn typical day, when the DNI value is between 100 and 400 W/m², and the corresponding times are from 10:20 to 13:20 and from 16:20 to 16:40, the steam turbine output power of ISCC system integrated with HPE2 is greater than that of ISCC system integrated with HPB. So, using the changeable solar energy integration mode is helpful to increase the output power of the overall ISCC system and make the best of the solar energy.

|                      | The ISCC system with the new optimization strategy | The traditional ISCC system |
|----------------------|--------------------------------------------------|----------------------------|
| System electricity production (MW·h) | 9.39 × 10³ | 9.38 × 10³ |
| Solar electricity production (MW·h) | 68.22 | 63.96 |
| Solar contribution rate (%) | 1.55 | 1.44 |
| Solar photoelectric efficiency (%) | 17.18 | 15.97 |
| Solar photoelectric exergy efficiency (%) | 18.41 | 17.11 |
| System thermal efficiency (%) | 56.43 | 55.63 |
| System exergy efficiency (%) | 60.46 | 59.61 |

**FIGURE 11** Feedwater mass flow rate change of the integrated solar combined cycle system integrated with HPB

**FIGURE 12** Feedwater mass flow rate change of the integrated solar combined cycle system integrated with high pressure economizer 2

**FIGURE 13** The mass flow rate absolute differences of both high- and intermediate-pressure feedwater between two integration modes
4.3 Annual system thermal performance analysis

In the process of the annual system performance simulation, the time interval between each two measurement points is set as one hour. As can be seen from Figure 2, compared with the DNI distributions in autumn and winter, the DNI data in spring and summer are higher, so the solar energy resources in spring and summer are more adequate.

Due to the ambient temperature, the DNI, the solar field settings, and other factors, the solar collector field collecting heat efficiency varies greatly in the morning and afternoon time, especially in the spring and summer. While in autumn and winter, the solar collector field collecting heat efficiency varies little in the morning and afternoon time. So, the annual operation strategy of ISCC system is as follows:

The solar energy resources are usually sufficient in both spring and summer. At the morning time, when DNI is >250 W/m², open valves b1, b2 and close valves a1, a2, the solar energy will be integrated with HPB. When DNI is >100 W/m² and <250 W/m², open valves a1, a2 and close valves b1, b2, the solar energy will be integrated with HPE2. When DNI is lower than 100 W/m², the ISCC system operates as the traditional GTCC system. At the afternoon time,

**TABLE 6** The typical annual system thermodynamic performance parameters

|                      | The ISCC system with the new optimization strategy | The traditional ISCC system |
|----------------------|--------------------------------------------------|-----------------------------|
| System electricity production (MW-h) | 3.445 × 10⁶ | 3.444 × 10⁶ |
| Solar electricity production (MW-h) | 2.37 × 10⁴ | 2.27 × 10⁴ |
| Solar contribution rate (%) | 1.95 | 1.9 |
| Solar photoelectric efficiency (%) | 19.08 | 17.99 |
| Solar photoelectric exergy efficiency (%) | 19.11 | 18.01 |
| System thermal efficiency (%) | 55.42 | 55.40 |
| System exergy efficiency (%) | 53.74 | 53.73 |

**TABLE 7** System thermal performance parameters in spring and summer of Dunhuang

|                      | The ISCC system with the new optimization strategy | The traditional ISCC system |
|----------------------|--------------------------------------------------|-----------------------------|
| Solar electricity production (MW-h) | 1.868 × 10⁴ | 1.804 × 10⁴ |
| Solar photoelectric efficiency (%) | 22.15 | 21.09 |
| Solar photoelectric exergy efficiency (%) | 22.15 | 21.10 |

**TABLE 8** System thermal performance parameters in autumn and winter of Dunhuang

|                      | The ISCC system with the new optimization strategy | The traditional ISCC system |
|----------------------|--------------------------------------------------|-----------------------------|
| Solar electricity production (MW-h) | 5.20 × 10³ | 4.65 × 10³ |
| Solar photoelectric efficiency (%) | 13.91 | 12.10 |
| Solar photoelectric exergy efficiency (%) | 13.92 | 12.11 |

**TABLE 9** System thermal performance parameters in winter of Dunhuang

|                      | The ISCC system with the new optimization strategy | The traditional ISCC system |
|----------------------|--------------------------------------------------|-----------------------------|
| Solar electricity production (MW-h) | 0.92 × 10³ | 0.73 × 10³ |
| Solar photoelectric efficiency (%) | 11.02 | 8.85 |
| Solar photoelectric exergy efficiency (%) | 11.03 | 8.86 |
when DNI is >250 W/m² and <430 W/m², open valves a1, a2 and close valves b1, b2, the solar energy will be integrated with HPE2. When DNI is <250 W/m², the ISCC system operates as the traditional GTCC system.

The solar energy resources are often insufficient in both autumn and winter. When DNI is >100 and <400 W/m², open valves a1, a2 and close valve b1, b2, the solar energy will be integrated into the HPE2. When DNI is >400 W/m², open valves b1, b2 and close valves a1, a2, the solar energy will be integrated into the HPB. When DNI is <100 W/m², the ISCC system operates as a traditional GTCC system.

The safe operation of the HRSG system can be maintained by adjusting the mass flow rate of feedwater in each pressure section to reduce the heat load fluctuations in the HRSG. In addition, to ensure the security of the ISCC system, the valve switching system only switches between the HPB integration mode and the HPE2 integration mode.

The comparison of thermodynamic performances of these two ISCC systems with or without the optimization strategy in a typical year is shown in Table 6. The result shows that using the operation optimization strategy contributes to increase the ISCC system’s thermodynamic performance. The comparison results of thermodynamic performance parameters of these two ISCC systems with or without the optimization strategy in spring and summer, in autumn and winter, in winter are as shown in Tables 7-9, respectively. The results of Tables 7-9 show that due to the different DNI distributions in a whole year, the larger the distribution proportion of DNI with the lower value is, the greater the thermodynamic performance advantage of ISCC system with the new optimization strategy is.

### 4.4 Economic performance analysis

An economic performance analysis of the studied ISCC system is also made and the LCOE for the ISCC system with the new optimization strategy is calculated and compared with the LCOE of the traditional ISCC system. The main parameters of the economic performance analysis are shown in Table 10.

Figure 14 shows the annual net solar power generation of different ISCC systems. It is found that the annual net solar power generation increases with the increase of the number of parabolic trough collector.

Figure 15 shows the levelized cost of electricity of different ISCC systems. It is found that the levelized cost of electricity decreases with the increase of the number of parabolic trough collector.

| Investment | Value | Financial parameters | Value |
|-------------|-------|----------------------|-------|
| Specific investment cost for solar field ($/m²) | 206.8 | $i_{eff}$ (%) | 10 |
| Specific land cost ($/m²) | 2.03 | n (y) | 15 |
| Specific HTF cost ($/kg) | 2.03 | $r_i$ (%) | 3 |
| Surcharge for construction, engineering and contingencies (%) | 10 | $r_i$ (%) | 1 |
| Solar field specific O and M costs ($/m²·y) | 9.26 | O and M equipment cost percentage of investment per year (%) | 1 |

**TABLE 10** Cost assumption data used for the economic analysis
of parabolic trough collector in different ISCC systems. In addition, the net solar power generation in the ISCC system with the new optimization strategy is obviously higher than that in the traditional ISCC system. Figure 15 shows the LCOE of different ISCC systems. It is found that the LCOE first decreases with the increase of the number of parabolic trough collector under different ISCC system, then increases after it reaches the lowest value. Because of the annual net solar power generation increases with the number of the parabolic trough collector, the parabolic trough collector investment also increases with the increase of the number of parabolic trough collector. So it has an optimal number of the parabolic trough collector for different ISCC systems. And the optimal number of the parabolic trough collector corresponding to the ISCC system with the new optimization strategy is about 500. The lowest cost analysis results of different ISCC systems are shown in Table 11. As presented in Table 11, the LCOE of the ISCC system without the optimization strategy is 0.233 $/kW·h, while the LCOE of the system with the new optimization strategy is 0.223 $/kW·h, 0.01 $/kW·h lower than that of the ISCC system without the optimization strategy. So, the ISCC system with the new operation optimization strategy has better economic performance advantage.

5 | CONCLUSIONS

In this paper, an optimization operation strategy of the ISCC system with changeable integration mode under different DNI conditions is proposed. Both the thermodynamic and economic performances of ISCC system with/without the optimal operation strategy are analyzed and compared. Some conclusions are obtained as follows:

1. The ISCC system with the new optimization strategy can change the integration mode through the valve switching according to the DNI change in different seasons. The comparison results show that in a typical autumn day, the solar photoelectric efficiency of ISCC system with the new optimal operation strategy is increased by 1.17%, the solar energy contribution rate is increased by 0.05%, and the solar photoelectric exergy efficiency is increased by 1.17%. The annual thermodynamic performance analysis results show that all performance parameters of ISCC system with the new optimization strategy are improved. The solar photoelectric efficiency is increased by 1.1%, the solar energy contribution rate is increased by 0.05%, and the solar photoelectric exergy efficiency is increased by 1.1%;

2. The total power generation of the traditional ISCC system is influenced by the solar collecting efficiency, the solar to electricity efficiency, DNI distribution time comprehensively. So the optimal number of the parabolic trough collector and the integration mode will be different under different DNI distributions. Through the economic performance analysis of ISCC system with the new optimal operation strategy and the traditional ISCC system without the optimization strategy, it is found that the LCOE first decreases with the increase of the number of the parabolic trough collector under different ISCC systems, then increases after it reaches the lowest value. There is an optimal number of the parabolic trough collector corresponding to the minimum LCOE, and the optimal number of the parabolic trough collector corresponding to the new optimization strategy is 500. And the LCOE of the ISCC system with the new optimization strategy is 0.01 $/kW·h lower than that of the traditional ISCC system without the optimization strategy. So the ISCC system with the new optimization strategy also has a better economic performance.

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NOMENCLATURE

Abbreviation

AC, Air compressor; CC, Combustion chamber; DNI, Direct normal irradiance; G, Generator; GT, Gas turbine; GTCC, Gas turbine combined cycle; HFP, High-pressure feedwater pump; HP, High pressure; HPB, High-pressure boiler; HPE1, High-pressure economizer 1; HPE2, High-pressure economizer 2; HPS, High-pressure super heater; HRSG, Heat recovery steam generator; HT, High-pressure turbine; HTF, Heat transfer fluid; IFP, Intermediate-pressure feedwater pump; IP, Intermediate pressure; IPB, Intermediate-pressure boiler; IPE, Intermediate-pressure economizer; IPS, Intermediate-pressure super heater; IT, Intermediate-pressure turbine; ISCC, Integrated solar combined cycle; LFP, Low-pressure feedwater pump; LP, Low pressure; LPB, Low-pressure boiler; LPE, Low-pressure economizer; LPS, Low-pressure super heater; LT, Low-pressure turbine; RH Reheater

| TABLE 11 The levelized cost of electricity of different integration cases |
|-----------------|------------------|
| Parameters      | Value            |
| LCOE ($/kW·h) of the traditional ISCC system | 0.233 |
| LCOE ($/kW·h) of the ISCC system with the new optimization strategy | 0.223 |
Symbols

- \( A \)  
  - turbine inlet area, \( m^2 \)

- \( A_{owhe} \)  
  - heat exchanger area, \( m^2 \)

- \( A_{surface} \)  
  - trough solar collector single reflection area, \( m^2 \)

- \( B_{i} \)  
  - metal Biot number

- \( B_{ith} \)  
  - thermal barrier coating Biot number

- \( c \)  
  - the specific heat capacity, \( J/(kg\cdot{}C) \)

- \( c_p \)  
  - specific heat capacity of gas at constant pressure, \( kJ/(kg\cdot{}K) \)

- \( C_0 \)  
  - cost of the operation and maintenance

- \( CELF \)  
  - constant-escalation levelization factor

- \( CFR \)  
  - capital recovery factor

- \( E_{sol,net} \)  
  - solar electricity production, MW-h

- \( E_{Xf} \)  
  - the chemical exergy of the fuel \( CH_4 \), MW

- \( F_{fuel,ISCC} \)  
  - fuel mass flow of ISCC, kg/s

- \( F_{fuel,GTCC} \)  
  - fuel mass flow of GTCC, kg/s

- \( F_g \)  
  - gas physical property coefficient

- \( i_{eff} \)  
  - effective discount rate

- \( INV \)  
  - total investment

- \( k \)  
  - gas thermal conductivity, \( N\cdot{}s/m^2\)

- \( k_i \)  
  - incident angle correction

- \( k_{cool} \)  
  - cooling flow factor

- \( K \)  
  - incident angle modifier

- \( K_1 \)  
  - design factor

- \( K_{owhe} \)  
  - heat transfer coefficient, \( W/(m^2\cdot{}C) \)

- \( LCOE \)  
  - levelized cost of electricity

- \( LC_{INV} \)  
  - levelized costs of the investment

- \( LC_{O&M} \)  
  - levelized costs of the operation and maintenance

- \( LHV_f \)  
  - lower heating value of fuel, \( kJ/kg \)

- \( m \)  
  - mass flow, kg/s

- \( n \)  
  - economic life, year

- \( p \)  
  - pressure, MPa

- \( Q_f \)  
  - fuel energy input, MW

- \( Q_{sol} \)  
  - solar thermal energy added to the system, MW

- \( W_{ISCC} \)  
  - power output of ISCC, MW

- \( W_{GTCC} \)  
  - power output of GTCC, MW

- \( W_{ST} \)  
  - power output of steam turbine, MW

- \( W_{sol,net} \)  
  - solar power generation, MW

- \( r_i \)  
  - average annual inflation rate

- \( r_n \)  
  - nominal escalation rate

- \( r_e \)  
  - real escalation rate

- \( R \)  
  - gas constant

- \( t \)  
  - celsius temperature, \( {}C \)

- \( T \)  
  - kelvin temperature, K

- \( T_0 \)  
  - ambient temperature, K

- \( T_a \)  
  - solar collector temperature, K

- \( T_{col,i} \)  
  - inlet temperature of solar collector, K

- \( T_{col,o} \)  
  - outlet temperature of solar collector, K

- \( T_{met,ext} \)  
  - allowable temperature of the metal surface, K

- \( \Delta t \)  
  - log-mean temperature difference, K

- \( \Delta T \)  
  - temperature difference between the collector working temperature and the ambient temperature, K

\[ \Delta F \]  
steam mass flow difference, kg/s

\[ \mu \]  
- gas dynamic viscosity, \( W/(m\cdot{}K) \)

Greek Symbols

- \( \eta \)  
  - collecting efficiency

- \( \eta_i \)  
  - each stage turbine efficiency

- \( \eta_{net} \)  
  - system thermal efficiency

- \( \eta_{ex,net} \)  
  - system exergy efficiency

- \( \eta_{ex,sol} \)  
  - solar photoelectric exergy efficiency

- \( \eta_{sol,net} \)  
  - solar photoelectric efficiency

- \( X_{sol} \)  
  - solar energy contribution

- \( \delta \)  
  - expansion ratio

- \( \psi \)  
  - solar energy grade

- \( \varphi \)  
  - cooling air flow fraction

- \( \gamma \)  
  - specific heat ratio

- \( \epsilon \)  
  - cooling effectiveness

- \( c_f \)  
  - film cooling coefficient

- \( c_c \)  
  - blade cooling coefficient

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