Investigation of off-design characteristics of an improved recompression supercritical carbon dioxide cycle for concentrated solar power application

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Summary
The off-design characteristics of an improved recompression supercritical carbon dioxide cycle integrated with a two-stage intercooled main compressor are investigated with a focus on the concentrated solar power application. An off-design model is established for each crucial component of the cycle system of 100-megawatt scale. Four cycle control schemes with different main compressor configurations or/cycle maximum pressure modes are evaluated and compared. A sensitivity analysis is performed on the parameters related to the cycle thermal input and ambient condition to predict the off-design characteristics due to the plant dispatch and ambient condition change in a solar power plant. The off-design results regarding the cycle thermodynamic performance and operational issue prevention are presented. The effect of the design-point value of the main compressor inlet temperature on the off-design characteristics is evaluated with the comparison among the results at three design points. The results reveal that the compressor surge may occur to the main compressor with basic configuration as the main compressor inlet temperature decreases to a certain value beneath the corresponding design point. By contrast, the surge risk can be prevented with the modified main compressor configuration by activating the recirculation system and the cycle can thus operate normally in the entire off-design range of main compressor inlet temperature. The off-design change in thermal input has overall limited effects on the cycle system control. No operational compressor issues occur for the main compressor with either basic or modified configuration as the thermal input deviates from the design points and varies in the studied ranges. The cycle maximum pressure mode has slight effects on the cycle thermodynamic performance as the thermal input deviates from the design point. The flexible cycle maximum pressure mode has slightly lower sensitivity to the thermal input variation in net output power due

Abbreviations: B-FP, basic configuration with fixed pressure; B-VP, basic configuration with variable pressure; CSF, concentrated solar power; DNI, direct normal irradiance, W/m²; HTF, heat transfer media; HTR, high-temperature recuperator; ICMC, intercooled main compressor; LTR, low-temperature recuperator; MC, main compressor; M-FP, modified configuration with fixed pressure; M-VP, modified configuration with variable pressure; PHX, primary heat exchanger; TES, thermal energy storage.
to the counteraction of mass flowrate variation. The selection of the design-point value of the main compressor inlet temperature has significant effects on the off-design characteristics of the cycle. A low design-point value of the main compressor inlet temperature leads to less demanding control action for operational issue prevention whereas a high design-point main compressor inlet temperature results in overall more stable thermodynamic performance under off-design conditions. Among other schemes, the fixed maximum pressure mode with the modified main compressor configuration is found to be the most satisfactory one due to the consequent superior efficiency, steady net output power, and free of hazardous operating issues despite the relatively demanding task of compressor surge prevention. The developed control scheme can be further improved by implementing parametric optimization during the off-design operation.

**KEYWORDS**
compressor surge prevention, concentrated solar power plant, control scheme, off-design modeling, supercritical carbon dioxide cycle

## 1 | INTRODUCTION

As a potential solution to the climate change and energy shortage, concentrated solar power (CSP) is regarded as an important renewable energy technologies in the future energy market due to its capability of utility-scale electricity production and dispatchable power supply with the integration of thermal energy storage.\(^1\)\(^2\) Heat transfer fluids, such as molten salts, synthetic oil or steam are adopted as the medium to transfer the solar energy into thermal energy to activate the power cycle system for power generation or store the thermal energy to facilitate the continuous operation when the desired solar power is unavailable. Limited by the degradation temperatures of different heat transfer media, the power block can operate at a maximum temperature level of around 350°C-700°C.\(^3\) The commercialized CSP plants employ a traditional subcritical steam cycle system for the power block, which requires a complex system configuration and large footprint and does not exhibit desirable limited thermal efficiency. The upgrade of the power cycle system is an important approach to maximizing the system performance of the CSP power and achieving a competitive cost of the CSP power generation. Among many other candidates, supercritical carbon dioxide (S-CO\(_2\)) cycle systems have been seen as a competitive option for the CSP plant due to the superior efficiency and compact configuration.\(^4\)\(^5\)

Numerous cycle designs of S-CO\(_2\) cycle were proposed and evaluated for applications in different power generation scenarios, such as nuclear power plant,\(^6\) coal-fired plant,\(^7\) waste heat recovery\(^8\) and CSP plant.\(^9\) Many previous works were carried out on the parametric studies of the S-CO\(_2\) cycle with an emphasis on the CSP application. Among different cycle designs, the recompression with an intercooled main compressor (ICMC) was considered as a competitive option due to the outstanding cycle performance and large temperature differential in the cycle hot end for the integration of the thermal storage system.\(^10\)\(^11\) Padilla et al\(^12\) carried out an exergetic comparison between several S-CO\(_2\) cycle configurations with the integrated exergetic model of the solar receiver and S-CO\(_2\) cycle system. The authors indicated that the recompression cycle with an ICMC was the cycle configuration with the highest overall exergetic efficiency. Ma et al\(^13\) conducted a superstructure-based optimization on the S-CO\(_2\) cycle system for CSP application with the integrated thermo-economic model of the entire CSP plant employing an S-CO\(_2\) cycle-based power block. This work also pointed out that the adoption of the recompression cycle with an ICMC can minimize the levelized cost of electricity for the CSP plant.

An S-CO\(_2\) cycle system is expected to frequently operate under off-design conditions in a CSP plant due to the variations in ambient conditions and cycle thermal input. The studies regarding the cycle off-design characteristics and operational strategy development are therefore essential for the further deployment of S-CO\(_2\) cycle systems in CSP plants. Dyreby\(^14\) developed an off-design model for a 10 MW recompression cycle which allows for the design analysis of the cycle system. The cycle performance is predicted with the model under various design and boundary conditions. Celle\(^15\) improved Dyreby’s heat exchanger model by considering the effects of variations in CO\(_2\) properties and investigate the impacts of yearly variation of the ambient condition on the thermodynamic performance of
the S-CO$_2$ cycle. Wang et al$^{16}$ performed a performance evaluation of a solar thermal power plant employing a direct air-cooled supercritical carbon dioxide Brayton cycle. The focus has been put on the design of the air-cooled heat sink and the effects of it under off-design conditions. The results revealed that the directed air-cooled system can accommodate the lower solar intensities without deterioration in electricity. Son et al$^{17}$ developed an one-dimension mean-line off-design model for S-CO$_2$ turbomachinery with the assistance of Deep Neural Network. With the developed models, it was claimed that more accurate off-design performance prediction could be achieved in even shorter time than that with the traditional correction model, which would greatly facilitate the off-design performance evaluation of the entire cycle system. Liu$^{18}$ investigated the approach of flexible operation of S-CO$_2$ cycle in a CSP plant with the assistance of a fossil-fuel heat source. Two operational modes regarding the turbine inlet temperature control were proposed and compared. According to the comparison, flexible temperature mode was reported to outperform constant temperature mode in terms of fossil fuel saving. Wright et al$^{19}$ conducted an experimental study on a small-scale Brayton cycle loop and compared the results with their developed models. The experimental results exhibited controllable and stable operation in the critical region, and the measured results were claimed to show good coherence with their modeling data. Carsten$^{20}$ developed control strategies for the dynamic operation of a recompression S-CO$_2$ cycle. Several control strategies, namely, high and low-temperature control, turbine bypass control, and inventory control, were compared under different operating modes including part-load operation, loss-of-load, loss of heat sink, over-power, and start-up/shut-down. Yang et al$^{21}$ carried out part-load performance analysis and comparisons among various S-CO$_2$ cycle configurations, and concluded that the modifications on the cycle configuration can lead to different effects in different scenarios.

The cycle configurations involved in the previous off-design studies were mostly recompression S-CO$_2$ cycle and simple-recuperated cycle. The results regarding the off-design characteristics of more promising S-CO$_2$ cycle designs for the CSP application, for example, the recompression cycle with an ICMC, were rarely reported. Besides, very few studies focused on the effects of the potential hazardous operational issues and the associated control schemes on off-design operation. Operating in the critical region, the variation in inlet temperature of the main compressor can lead to drastic changes in CO$_2$ density, which may cause surge or choke to the compressor.$^{14,22,23}$

This article develops an off-design model for the recompression cycle with an ICMC. Sensitivity analyses are conducted on the parameters related to different boundary conditions, namely, the thermal input and the ambient temperature, to simulate the cycle performance under various off-design conditions encountered in a real CSP plant. The cycle performance and the prevention of potential hazardous issues of the main compressor (MC) under off-design conditions are highlighted in the off-design study. Four operation modes featuring different MC configuration and/or cycle maximum pressure control are proposed and compared regarding the cycle thermodynamic performance and operational issue control for the MC under off-design conditions. Based on the comparison among the cycle performance with different operation modes, the optimal off-design control scheme is finally recommended.

## 2 | MODEL ESTABLISHMENT

Figure 1 is the schematic of the S-CO$_2$ recompression cycle with an ICMC and the corresponding T-S diagram. The detailed on-design model of this cycle was reported in our previous work.$^{11}$ In comparison to the basic recompression cycle, a higher cycle efficiency ($\eta_{\text{cyc}}$) and a larger temperature differential for thermal input can be achieved with the introduction of an ICMC, which can lead to more power output from the cycle and a lower capital cost of the thermal energy storage (TES) system.$^{13,24}$ The leveled cost of electricity of the entire CSP system can be reduced as a result. Table 1 gives the values of cycle design parameters to initialize the off-design calculation. As shown in Table 1, a 100-megawatt S-CO$_2$ power cycle system is considered in this work. To accommodate this large power production scale, a split shaft configuration is applied for the turbomachinery. A synchronous generator is tied to the turbine and variable-speed drive motors are used for the compressors for the sake of system efficiency and control flexibility.$^{25}$ The main compressor is a two-stage internally geared compressor with an intercooler in between. A CO$_2$-Salt counter-flow shell-and-tube heat exchanger is chosen for the primary heat exchanger (PHX). Counter-flow printed circuit heat exchangers with flow channels that are 5 mm wide and 2.5 mm deep, which is deemed as a representative design for using in S-CO$_2$ cycles,$^{14,26}$ is adopted for the high-temperature recuperator (HTR) and low-temperature recuperator (LTR). The air-cooled heat sink is applied for both the precooler and intercooler. A buffer tank is integrated at the inlet of the MC to impose active control on the inlet pressure of the MC with inventory control.$^{14}$ The inlet pressure of the MC is assumed to remain the on-design value and the split ratio of the stream flowing through the recompressor is also assumed to remain the on-design value with the active valve control.

An object-oriented approach is employed for the off-design model development of each main component in the cycle in MATLAB 2018a. The calculation of the
required thermal properties of the CO$_2$ is achieved by calling REFPROP database from MATLAB.\textsuperscript{27} The inlet pressure of the MC is assumed to remain the design value using the inventory control under off-design conditions\textsuperscript{14} and the split ratio of the stream flowing through the recompressor is also assumed to remain the on-design value with the active valve control. The model for each component is presented as follows.

### 2.1 Turbine

A multi-axial flow type turbine is selected for the cycle for the sake of high efficiency and steady flow considering the 100-megawatt power output.\textsuperscript{28} The off-design turbine inlet pressure is calculated according to the Stodola’s ellipse method assuming the sliding mode and a fixed nozzle area are also assumed for the turbine for the off-design calculation.\textsuperscript{29,30} Applying the Stodola’s ellipse method, the following relationship between the on-design ($\phi_d$) and off-design ($\phi_{od}$) mass flow coefficients is obtained:

$$\frac{\phi_{od}}{\phi_d} = \frac{\sqrt{1 - \left(\frac{p_{out,od}}{p_{in,od}}\right)^2}}{\sqrt{1 - \left(\frac{p_{out,d}}{p_{in,d}}\right)^2}}$$

(1)

where $\phi$ is defined as

$$\phi = \dot{m}_{in} \cdot \sqrt{\frac{T_{in}}{P_{in}}}$$

(2)

$p_{in,od}$ is then obtained as follow by substituting Equations (1) and (2) as
The off-design isentropic efficiency of turbine ($\eta_{T,od}$) is finally obtained as

$$\eta_{T,od} = \eta_{T,d} \cdot \sin \left[ 0.5 \pi \cdot \left( \frac{m_{in,od} \cdot \rho_{in,od}}{m_{in,d} \cdot \rho_{in,d}} \right)^0.1 \right]$$  \hspace{1cm} (4)

where $\rho_{in,d}$ and $\rho_{in,od}$ are the density of CO$_2$ under on-design and off-design, respectively.

### 2.2 Compressor

The off-design performance characterization of the compressors in the cycle is achieved by regressing the experimental data of a prototype compressor at the Sandia National Laboratory with further nondimensionalization on the shaft speed ($N$). The dimensionless flow coefficient ($\phi_C$) and ideal head coefficient ($\psi_C$) of the compressor are first defined as shown in Equations (5) and (6).

$$\phi_C = \frac{m}{\rho_{in} \cdot U_{in} \cdot D_{in}^2}$$  \hspace{1cm} (5)

$$\psi_C = \frac{(h_{out} - h_{in})}{U_{in}^2}$$  \hspace{1cm} (6)

where $U_{in}$ is the tip velocity at the inlet as defined in Equation (7) and $D_{in}$ is the inlet diameter of the compressor. $h_{in}$ and $h_{out}$ stand for the inlet and outlet enthalpies of CO$_2$, respectively.

$$U_{in} = \frac{D_{in}}{2} \cdot N$$  \hspace{1cm} (7)

Then, the nondimensionalization on the shaft speed is done for the flow coefficient, ideal head coefficient, and the compressor isentropic efficiency, by introducing the corresponding item relating to the shaft speed to the expressions of these parameters as displayed in Equations (8) through (10). It should be noted that the modified flow coefficient should be within the range of 0.02-0.05 to avoid operational issues. The lower bound and upper bound are deemed as the threshold to trigger surge and choke. Nevertheless, a more accurate performance map is necessary for the purpose of capturing the abnormal operating conditions accurately. Finally, the modified ideal head coefficient ($\psi_C^*$) and modified compressor isentropic efficiency ($\eta_C^*$) are characterized as the function of modified flow coefficient $\phi_C'$ with polynomial regression as presented in Equations (11) and (12).

$$\phi_C = \frac{m}{\rho_{in} \cdot U_{in} \cdot D_{in}^2} \left( \frac{N}{N_d} \right) \in [0.02, 0.05]$$  \hspace{1cm} (8)

$$\psi_C^* = \frac{(h_{out} - h_{in})}{U_{in}^2} \left( \frac{N}{N_d} \right)^{(20\phi_C')^{0.5}}$$  \hspace{1cm} (9)
\[ \eta_C = \eta_C \left( \frac{N_{C,D,0}}{N_{C,D}} \right) \left( \frac{N}{N_d} \right)^{(20\gamma_C)^3} \]  

(10)

\[ \eta_C^* = -0.7069 + 168.66\phi_C^* - 8099\phi_C^{*2} + 182725\phi_C^{*3} - 1638000\phi_C^{*4} \]  

(11)

\[ \psi_C^* = -0.4049 + 54.6\phi_C^* - 2505\phi_C^{*2} + 53224\phi_C^{*3} - 498626\phi_C^{*4} \]  

(12)

### 2.3 Heat exchanger

The off-design model is developed for the heat exchangers in the S-CO\(_2\) cycle system including the HTR, LTR and PHX. For the intercooler and precooler, the off-design models are simplified by assuming a constant cold-end temperature difference of 15 K with the active control of the cooling air flowrate. The off-design model is an one-dimensional counter-flow heat exchanger model divided into several axial nodes as commonly applied for the off-design performance prediction of the heat exchangers in the system-level modeling in previous works.\(^{14,26}\) The developed off-design model considers both the heat transfer and fluid dynamic characteristics of the heat exchanger. For the purpose of capturing the effect of fluid property variation, the models are discretized into 20 sub-nodes. The pressure loss (\(\Delta p\)) and the conductance of heat transfer (\(UA\)) are calculated for each sub-node following the method developed in Patnode’s thesis\(^{31}\) with some customized modifications. The effects of fluid property variation on the off-design performance are considered in different way for each type of heat exchanger. For the HTR and PHX which have insignificant thermal property variations of the CO\(_2\) fluid, the effects of thermal property variations of each sub-node in the heat transfer process are neglected. The conductance of heat transfer (\(UA_{od}\)) and pressure loss (\(\Delta p_{od}\)) under off-design conditions are treated as the functions of the mass flowrate as follows.

\[ \frac{UA_{od}}{UA_d} = \frac{\dot{m}_{od}^{-0.8} + \dot{m}_{hd}^{-0.8}}{\dot{m}_{c,od}^{-0.8} + \dot{m}_{h,od}^{-0.8}} \]  

(13)

\[ \frac{\Delta p_{od}}{\Delta p_d} = \left( \frac{\dot{m}_d}{\dot{m}_{od}} \right)^{7/4} \]  

(14)

The CO\(_2\) fluid flowing through the LTR displays significant and nonlinear variation in fluid properties relating to the heat transfer and hydraulic characteristic due to the vicinity to the critical point. Therefore, the \(UA_{od}\) and \(\Delta p_{od}\) are calculated considering the thermal property variations as shown in Equations (15) and (16).\(^{15}\)

\[ \frac{UA_{od}}{UA_d} = \frac{\alpha_{c,d}^{-1} + \alpha_{h,d}^{-1}}{\alpha_{c,od}^{-1} + \alpha_{h,od}^{-1}} \]  

\[ \frac{\Delta p_{od}}{\Delta p_d} = \left( \frac{\dot{m}_d}{\dot{m}_{od}} \right)^{7/4} \cdot \left( \frac{\mu_d}{\mu_{od}} \right)^{1/4} \cdot \left( \frac{\rho_d}{\rho_{od}} \right)^{-1} \]  

(15)

(16)

where \(c_p\) stands for the specific heat capacity, \(k\) stands for the coefficient of heat conductivity, \(\mu\) stands for the dynamic coefficient of viscosity. \(n\) equals to 0.4 and 0.3 for hot and cooling fluid, respectively.

### 2.4 Model validation

The developed model is validated with the S-CO\(_2\) off-design code developed by Dyreby.\(^{14}\) The results obtained from the developed model are compared to Dyreby’s under both on-design and the off-design conditions as shown in Table 2. It is found the results of the developed model show good coherence with the results obtained with Dyreby’s code under both on-design and off-design conditions with a maximum difference of 0.14%. The accuracy of this model is therefore validated.

### 3 CONSIDERATIONS FOR THE OFF-DESIGN CHARACTERISTIC STUDY

The S-CO\(_2\) cycle system employed in a CSP plant frequently operates under off-design conditions due to the implementation of CSP dispatch and variation of ambient conditions. The CSP dispatch leads to the variation of thermal input to the cycle in the hot end, which may change the mass flowrate (\(\dot{m}_{salt}\)) and inlet temperature (\(T_{salt,in}\)) of the molten salt at the inlet of the primary heat exchanger. This may then affect the performance of turbine and other cycle components. The variation of ambient temperature causes the change of CO\(_2\) temperature at the outlets of precooler and intercooler, which in turn lead to the change in inlet temperature of the MC (\(T_{MC,in}\)). The variations in the MC inlet temperature tend to cause drastic changes in the density of CO\(_2\) at the inlet of the main compressor, especially when the pressure of CO\(_2\) is close to the critical pressure. Surge/choke may occur to the MC as the volume flowrate drastically fluctuates down/up with density if no measure is taken. Therefore, regarding the control of the potential operational
issue to the MC under varied conditions of inlet temperature, two configurations of the MC, namely, the basic configuration and the modified configuration, are evaluated and compared in the following discussion. The simplified diagrams of the two MC configurations are presented in Figure 2. The shaft speed is the only control variable for the MC with the basic configuration for operational issue prevention, whereas the MC with the modified configuration is further configured with an anti-surge valve and an additional parallel compressor for each compressor stage besides using the shaft speed control to deal with the potential operational issues. The prevention of the surge condition is achieved by the adjustments of both shaft speed and the recirculation flows in two stages ($m_{rec,1}$ and $m_{rec,2}$). The choke control is achieved by the adjustment of shaft speed and the introduction of the second parallel compressor configured for both stages if necessary. These modifications are expected to expand the applicable

### TABLE 2  Validation of the developed model

| Parameter                | Unit  | On-design condition | Off-design condition |
|--------------------------|-------|---------------------|----------------------|
|                          |       | Present | Dyreby | Error (%) | Present | Dyreby | Error (%) |
| Input heat               | [MW]  | 10      | 10     | Input     | 5.22    | 5.22   | −0.04     |
| $p_{min}$                | [kPa] | 9000    | 9000   | Input     | 9000    | 9000   | Input     |
| $p_{max}$                | [kPa] | 25 000  | 25 000 | Input     | 18 933.28 | 18 993.20 | 0.32     |
| Split ratio              | [—]   | 0.3     | 0.3    | Input     | 0.3     | 0.3    | Input     |
| $\text{eff}_\text{HRTR}$| [—]   | 0.95    | 0.95   | Input     | 0.94    | 0.94   | −0.05     |
| $\text{eff}_\text{LTR}$  | [—]   | 0.95    | 0.95   | Input     | 0.90    | 0.90   | −0.03     |
| Cycle efficiency         | [%]   | 44.67   | 44.67  | 0.00      | 40.46   | 40.46  | 0.02      |
| Mass flowrate            | [kg/s]| 51.76   | 51.76  | 0.00      | 35.90   | 35.90  | 0.00      |
| $t_1$ ($t_2, t_3$)       | [:C]  | 41      | 41     | Input     | 50      | 50     | Input     |
| $t_4$                    | [:C]  | 93.10   | 93.10  | 0.00      | 106.82  | 106.82 | 0.00      |
| $t_5'$                   | [:C]  | 210.47  | 210.88 | 0.20      | 239.85  | 240.19 | 0.14      |
| $t_5''$                  | [:C]  | 197.40  | 197.40 | 0.00      | 196.08  | 196.09 | 0.00      |
| $t_5$                    | [:C]  | 206.51  | 206.80 | 0.14      | 226.41  | 226.65 | 0.10      |
| $t_6$                    | [:C]  | 395.19  | 395.19 | 0.00      | 431.77  | 431.72 | −0.01     |
| $t_7$                    | [:C]  | 550     | 550    | Input     | 550     | 550    | Input     |
| $t_8$                    | [:C]  | 431.12  | 431.12 | 0.00      | 461.20  | 461.20 | 0.00      |
| $t_9$                    | [:C]  | 217.67  | 218.03 | 0.17      | 240.91  | 241.24 | 0.14      |
| $t_10$                   | [:C]  | 98.35   | 98.36  | 0.00      | 118.82  | 118.87 | 0.04      |

**FIGURE 2** Two configurations of the main compressor. (A) basic configuration, (B) modified configuration
FIGURE 3  Four operation schemes under off-design conditions. (A) Basic main compressor configuration with fixed cycle maximum pressure (B-FP), (B) modified main compressor configuration with fixed cycle maximum pressure (M-FP), (C) basic main compressor configuration with variable cycle maximum pressure (B-VP); (D) modified main compressor configuration with variable cycle maximum pressure (M-VP)
conditions during off-design operation. Besides, two cycle maximum pressure modes, namely, fixed pressure mode and variable pressure mode, are evaluated and compared with each other. The main compressor outlet pressure ($p_{MC,\text{out}}$) is always controlled at 25 MPa in the fixed pressure mode, whereas in the variable pressure mode, $p_{MC,\text{out}}$ is not controlled unless it would cross the maximum bound of 30 MPa. Four operation schemes are thereby presented for the analysis, namely. (1) Basic main compressor configuration with Fixed cycle maximum Pressure mode (B-FP); (2) modified main compressor configuration with Fixed cycle maximum Pressure (M-FP); (3) basic main compressor configuration with Variable cycle maximum pressure (B-VP); (4) modified main compressor configuration with Variable cycle maximum Pressure (M-VP). Figure 3A-D presents the realization of these four schemes.

## RESULTS AND DISCUSSION

The cycle off-design performance is evaluated with a sensitivity analysis in this section. Three design points of the main compressor inlet temperature ($T_{MC,\text{in,d}}$), i.e., 32°C, 41°C and 50°C, are selected to investigate the effects of the $T_{MC,\text{in,d}}$ on the off-design performance. The design variables are optimized for three design points prior to the off-design analysis. The values of the relevant parameters at the three corresponding on-design points are presented in Table 3. The thermodynamic states of the cycle at these three design points are presented in Table 4. The parametric analyses are performed for $T_{\text{salt,in}}$ and $m_{\text{salt}}$ to consider the off-design changes of the thermal input and for $T_{MC,\text{in}}$ to consider the effect of the ambient temperature. The information regarding the off-design sensitivity

| Parameter | Design value of $T_{MC,\text{in}}$ [°C] | 32 | 41 | 50 |
|-----------|--------------------------------------|----|----|----|
| Inlet pressure of the MC | MPa | 6.63 | 8.06 | 9.82 |
| Split ratio | — | 0.62 | 0.64 | 0.70 |
| Inlet rotor diameter of the first MC stage | m | 0.50 | 0.49 | 0.49 |
| Inlet rotor diameter of the second MC stage | m | 0.41 | 0.42 | 0.43 |
| Shaft speed of the first MC stage | rpm | 2361 | 2562 | 2131 |
| Shaft speed of the second MC stage | rpm | 12000 | 10329 | 8786 |
| Inlet rotor diameter of the first RC stage | m | 0.88 | 0.84 | 0.88 |
| Inlet rotor diameter of the second RC stage | m | 0.35 | 0.41 | 0.48 |
| Shaft speed of the RC | rpm | 11977 | 11208 | 10215 |
| Cycle mass flow rate | kg/s | 859.18 | 986.49 | 1164.39 |
| Design-point cycle efficiency | % | 48.00 | 46.44 | 44.81 |

| State point | $T_{MC,\text{in,d}} = 32^\circ\text{C}$ | $T_{MC,\text{in,d}} = 41^\circ\text{C}$ | $T_{MC,\text{in,d}} = 50^\circ\text{C}$ |
|-------------|--------------------------------------|--------------------------------------|--------------------------------------|
| $t$ [°C] | $p$ [kPa] | $t$ [°C] | $p$ [kPa] | $t$ [°C] | $p$ [kPa] |
| 1 | 32 | 6627.80 | 41 | 8060.06 | 50 | 9817.88 |
| 2 | 45.25 | 7909.19 | 54.90 | 9804.57 | 59.95 | 11475.54 |
| 3 | 32 | 7889.19 | 41 | 9784.57 | 50 | 11455.54 |
| 4 | 64.97 | 25000 | 75.42 | 25000 | 84.43041 | 25000 |
| 5 | 201.89 | 24960 | 196.47 | 24960 | 191.011 | 24960 |
| 5' | 202.05 | 24960 | 198.96 | 24960 | 195.672 | 24960 |
| 5' | 201.62 | 24960 | 191.72 | 24960 | 180.6101 | 24960 |
| 6 | 355.47 | 24930 | 374.73 | 24930 | 396.002 | 24930 |
| 7 | 550 | 24880 | 550 | 24880 | 550 | 24880 |
| 8 | 392.26 | 6727.80 | 413.57 | 8160.06 | 435.78 | 9917.88 |
| 9 | 211.52 | 6667.80 | 207.35 | 8100.06 | 203.08 | 9857.88 |
| 10 | 72.97 | 6647.80 | 82.62 | 8080.06 | 91.73 | 9837.88 |
analysis is shown in Table 5. It should be noted that the upper limit of $T_{\text{salt,in}}$ selected here is selected only for the sake of parametric analysis. The actual operating $T_{\text{salt,in}}$ for the CSP using solar salt as storage media is recommended to remain lower than 580°C considering the chemical stability of the solar salt.32

### 4.1 Analysis on thermal input

Figure 4 presents the results of the variables relating to the cycle control as the function of $T_{\text{salt,in}}$. No operational issues occur to the MC in the entire studied range of $T_{\text{salt,in}}$. The MC configuration is found to have no effects on the system control. The cycle maximum pressure is slightly different in two cycle maximum pressure mode as can be found by comparing the results of FP and VP cases. As $T_{\text{salt,in}}$ rises, the cycle maximum pressure increases linearly under the VP mode and the shaft speed decreases linearly under the FP mode. The maximum variation of the cycle maximum pressure is within ±0.15 MPa and the variation of the shaft speed is within ±0.3% as $T_{\text{salt,in}}$ deviates the design point by ±30°C. The value of $T_{\text{MC,in,d}}$ does not show apparent effects on the cycle control.

Figures 5 and 6 display the results of variables relating to the cycle thermodynamic performance. As shown in Figures 5 and 6, the variation of $T_{\text{salt,in}}$ has relatively mild effects on the cycle thermodynamic performance. The maximum variations of $\eta_{\text{en,cycle}}$ and $\dot{W}_{\text{net}}$ are both within ±5.5% as $T_{\text{salt,in}}$ deviates the design point by ±30°C. The choice of control schemes has limited effects on the thermodynamic characteristics under off-design conditions of $T_{\text{salt,in}}$ as evidenced by the close variation trends of the four control scheme cases. The MC configuration is

| Parameter | Unit | Design point value | Range for sensitivity analysis |
|-----------|------|--------------------|--------------------------------|
| $T_{\text{salt,in}}$ | °C | 565 | 535-595 |
| $\dot{m}_{\text{salt}}/\dot{m}_{\text{salt,d}}$ | \ | 1 | 0.6-1.2 |
| $T_{\text{MC,in}}$ | °C | 32/41/50 | 32-50 |

**FIGURE 4** The variations of relative shaft speed ($RN = N/N_d$) and main compressor outlet pressure ($p_{\text{MC,out}}$) with molten salt temperature ($T_{\text{salt,in}}$) under three design points of main compressor inlet temperature ($T_{\text{MC,in,d}}$). (A) the case of $T_{\text{MC,in,d}} = 32°C$, (B) the case of $T_{\text{MC,in,d}} = 41°C$, (C) the case of $T_{\text{MC,in,d}} = 50°C$ [Colour figure can be viewed at wileyonlinelibrary.com]
FIGURE 5  The variations of net output power (\(W_{\text{net}}\)) and cycle energetic efficiency (\(\eta_{\text{en,cycle}}\)) with molten salt temperature (\(T_{\text{salt,in}}\)) under three design points of main compressor inlet temperature (\(T_{\text{MC,in,d}}\)). (A) The case of \(T_{\text{MC,in,d}} = 32^\circ\text{C}\), (B) the case of \(T_{\text{MC,in,d}} = 41^\circ\text{C}\), (C) the case of \(T_{\text{MC,in,d}} = 50^\circ\text{C}\) [Colour figure can be viewed at wileyonlinelibrary.com]

FIGURE 6  The variations of outlet temperature of the molten salt (\(T_{\text{salt,out}}\)), turbine inlet temperature (\(T_{\text{in}}\)) and mass flowrate of cycle working fluid CO\(_2\) (\(m_{\text{CO2}}\)) with molten salt temperature (\(T_{\text{salt,in}}\)) under three design points of main compressor inlet temperature (\(T_{\text{MC,in,d}}\)). (A) The case of \(T_{\text{MC,in,d}} = 32^\circ\text{C}\), (B) the case of \(T_{\text{MC,in,d}} = 41^\circ\text{C}\), (C) the case of \(T_{\text{MC,in,d}} = 50^\circ\text{C}\) [Colour figure can be viewed at wileyonlinelibrary.com]
found to have no effects here according to the comparison between the results of different MC configurations. The cycle maximum pressure control has slight effects on the results, specifically, the variation rate of $m_{\text{CO}_2}$ and $W_{\text{net}}$ with the change of $T_{\text{salt,in}}$. The cycle with the VP mode has slightly lower variation rate of $m_{\text{CO}_2}$ than that with the FP mode due to the alleviation from the change of maximum pressure with $T_{\text{salt,in}}$. Correspondingly, the $W_{\text{net}}$ of the cycle with the VP mode has slightly higher variation rate than that with the FP mode. The cycle with lower $T_{\text{MC,in,d}}$ exhibits higher $\eta_{\text{en,cycle}}$ and lower $m_{\text{CO}_2}$ as expected. The effect of $T_{\text{MC,in,d}}$ appears to be independent of the variation of $T_{\text{salt,in}}$ since the studied variables have similar variation tendencies and rates as the $T_{\text{salt,in}}$ deviates from the design point.

Figure 7 presents the sensitivity analysis results on the variables relating to the cycle control as the function of $m_{\text{salt}}/m_{\text{salt,d}}$. No operational compressor issues occur as $m_{\text{salt}}$ deviates from the design point. The main compressor configuration does not affect the system control under the off-design conditions of $m_{\text{salt}}$. The cycle maximum pressure is slightly different in two cycle maximum pressure modes according to the comparison between the results of FP and VP cases. As $m_{\text{salt}}$ changes from the 60% to 120% of the design value, the cycle maximum pressure increases under the variable pressure mode and the shaft speed decreases linearly under the FP mode, both at a decreasing rate. The cycle maximum pressure is decreased by around 0.35-0.55 MPa for the cycle with the VP mode and the shaft speed is increased by 1% -1.5% as the $m_{\text{salt}}$ decreases to 60% of the design value, while these two variables only increases and decreases by less than 0.04 MPa and 0.1%, respectively, as the $m_{\text{salt}}$ rises to 120% of the design value. The reduction in $T_{\text{MC,in,d}}$ slightly increases the sensitivity of the control variable to the $m_{\text{salt}}$ variation.

Figures 8 and 9 display the sensitivity analysis results on the variables relating to the cycle thermodynamic performance as the functions of $m_{\text{salt}}/m_{\text{salt,d}}$. As shown in Figures 8 and 9, the thermodynamic performance is more sensitive to the decrease in $m_{\text{salt}}$ than to its increase. The $\eta_{\text{en,cycle}}$ and $W_{\text{net}}$ decrease by 15%-22% as $m_{\text{salt}}$ decreases to 60% of the design point. The control scheme has overall limited effects on the thermodynamic characteristics as $m_{\text{salt}}$ deviates. The MC configuration is found to have no effects here according to the comparison between the results of the cases with different MC configurations. The cycle maximum pressure mode has slight effects on the variation rate of $m_{\text{CO}_2}$ and $W_{\text{net}}$ with the change of $m_{\text{salt}}$, especially as $m_{\text{salt}}$ decreases. The VP mode leads
FIGURE 8  The variations of net output power ($W_{\text{net}}$) and cycle energetic efficiency ($\eta_{\text{en,cycle}}$) with the molten salt mass flow rate fraction ($m_{\text{salt}}/m_{\text{salt,d}}$) under three design points of main compressor inlet temperature ($T_{\text{MC,in,d}}$). (A) the case of $T_{\text{MC,in,d}} = 32^\circ \text{C}$, (B) the case of $T_{\text{MC,in,d}} = 41^\circ \text{C}$, (C) the case of $T_{\text{MC,in,d}} = 50^\circ \text{C}$ [Colour figure can be viewed at wileyonlinelibrary.com]

FIGURE 9  The variations of outlet temperature of the molten salt ($T_{\text{salt,out}}$), turbine inlet temperature ($T_{\text{T,in}}$) and mass flowrate of cycle working fluid CO$_2$ ($m_{\text{CO}_2}$) with the molten salt mass flow rate fraction ($m_{\text{salt}}/m_{\text{salt,d}}$) under three design points of main compressor inlet temperature ($T_{\text{MC,in,d}}$). (A) the case of $T_{\text{MC,in,d}} = 32^\circ \text{C}$, (B) the case of $T_{\text{MC,in,d}} = 41^\circ \text{C}$, (C) the case of $T_{\text{MC,in,d}} = 50^\circ \text{C}$ [Colour figure can be viewed at wileyonlinelibrary.com]
to slightly higher $\dot{W}_{\text{net}}$ than the FP mode does under the low $\dot{m}_{\text{salt}}$ conditions. The reason for this is also due to the variation of the $\dot{m}_{\text{CO}_2}$ as explained above in the $T_{\text{salt,in}}$ cases. The cycle with a lower $T_{\text{MC,in,d}}$ exhibits higher $\eta_{\text{en,cycle}}$ and lower $\dot{m}_{\text{CO}_2}$ as expected. It is also found that the decrease of $T_{\text{MC,in,d}}$ leads to more drastic changes in the cycle variables with $\dot{m}_{\text{salt}}$, especially as the $\dot{m}_{\text{salt}}$ has a significant reduction relative to the design value.

### 4.2 Analysis on ambient temperature

Figure 10 presents the results of the variables relating to the cycle control as the function of $T_{\text{MC,in}}$. In comparison to the thermal input variation, the variation in ambient temperature has more significant effects on the cycle control. The cycle at different design points of the $T_{\text{MC,in,d}}$ exhibit different off-design characteristics. The results of different $T_{\text{MC,in,d}}$ are therefore discussed separately. No operational compressor issues occur as $T_{\text{MC,in}}$ deviates from the design point for the cycle with a $T_{\text{MC,in,d}}$ of 32°C, and the MC configuration does not affect the system control in this case. The two maximum pressure modes lead to significantly different results in the cycle maximum pressure and shaft speed. When $T_{\text{MC,in}}$ increases to 50°C, the cycle maximum pressure is reduced to around 12 MPa in the VP mode and the shaft speed is increased by around 50% in the FP mode to keep the cycle maximum pressure at 25 MPa. In the cases with a $T_{\text{MC,in,d}}$ of 41 or 50°C, compressor surge occurs to the MC with basic configuration when the $T_{\text{MC,in}}$ reduces to be lower than a certain value as shown in Figure 10B,C. For the cycle with the modified main compressor configuration, the recirculation system is activated when the compressor surge is approaching. The decrease in $T_{\text{MC,in}}$ entails more recirculating flow to prevent the potential surge condition. For the cycle with the FP mode, the shaft speed decreases as $T_{\text{MC,in}}$ decreases. For the cycle with the VP mode, the cycle maximum pressure increases as $T_{\text{MC,in}}$ decreases until it reaches the upper limit of the maximum pressure, the further decrease in $T_{\text{MC,in}}$ leads to the decrease in shaft speed. Besides, the cycle with the VP mode is less demanding on the prevention of compressor surge as the cycle with the VP mode can operate in a larger range of $T_{\text{MC,in}}$ without the use of recirculation system. The design under a high $T_{\text{MC,in,d}}$ entails more demanding control actions for compressor

![FIGURE 10](https://wileyonlinelibrary.com/image)
FIGURE 11  The variations of net output power ($\dot{W}_{\text{net}}$) and cycle energetic efficiency ($\eta_{\text{en,cycle}}$) with the main compressor inlet temperature ($T_{\text{MC,in}}$) under three design points of main compressor inlet temperature ($T_{\text{MC,in,d}}$). (A) the case of $T_{\text{MC,in,d}} = 32^\circ$C, (B) the case of $T_{\text{MC,in,d}} = 41^\circ$C, (C) the case of $T_{\text{MC,in,d}} = 50^\circ$C [Colour figure can be viewed at wileyonlinelibrary.com]

FIGURE 12  The variations of outlet temperature of the molten salt ($T_{\text{salt,out}}$), turbine inlet temperature ($T_{\text{T,in}}$) and mass flowrate of cycle working fluid CO$_2$ ($\dot{m}_{\text{CO}_2}$) with the main compressor inlet temperature ($T_{\text{MC,in}}$) under three design points of main compressor inlet temperature ($T_{\text{MC,in,d}}$). (A) The case of $T_{\text{MC,in,d}} = 32^\circ$C, (B) the case of $T_{\text{MC,in,d}} = 41^\circ$C, (C) the case of $T_{\text{MC,in,d}} = 50^\circ$C [Colour figure can be viewed at wileyonlinelibrary.com]
surge prevention as can be observed by comparing the results for the cycle with different $T_{MC,in,d}$.

Figures 11 and 12 display the results of the variables relating to the cycle thermodynamic performance as the functions of $T_{MC,in}$. As shown in Figures 11 and 12, the effects of $T_{MC,in}$ on the cycle thermodynamic performance are significantly different for the cycle with different $T_{MC,in,d}$. The $\dot{W}_{net}$ reduces as $T_{MC,in}$ increases for all three $T_{MC,in,d}$ cases, with more significant reduction observed with lower $T_{MC,in,d}$. The cycle maximum pressure mode has significant effects on the thermodynamic characteristics as $T_{MC,in}$ deviates from the design value. The FP mode can lead to much more stable $\eta_{en,cycle}$ and $\dot{W}_{net}$. A main reason for this is that the $\dot{m}_{CO2}$ remains constant for the cycle with a fixed maximum pressure. It is found that the cycle performance exhibits lower sensitivity to the $T_{MC,in}$ with a higher $T_{MC,in,d}$ according to the comparison among the results of the cycle with different $T_{MC,in,d}$.

5 CONCLUSION

This article develops an off-design model for a recompression S-CO$_2$ cycle with an ICMC with an emphasis on CSP application. The off-design characteristics with respect to the thermodynamic performance and operational issue prevention are highlighted. Four control schemes with different cycle maximum pressure mode or main compressor configuration are evaluated and compared. The effects of the off-design changes in cycle thermal input and ambient temperature are investigated through the sensitivity analyses on $T_{salt,in}$, $m_{salt}$ and $T_{MC,in}$. Three different $T_{MC,in,d}$ are selected to investigate the effects of the choice of $T_{MC,in,d}$ on the cycle off-design performance. The following conclusions are drawn:

- The compressor surge may occur to the main compressor with basic configuration when the $T_{MC,in}$ attains a certain value lower than the corresponding $T_{MC,in,d}$. By contrast, the surge risk can be prevented with the modified MC configuration by activating the recirculation system and can thus operate normally in the entire off-design range of $T_{MC,in}$. The choke condition does not occur to the cycle under all the off-design conditions, the parallel compressor is not activated for either stage of the main compressor. This indicate that the risk of choke appears controllable with shaft speed control. A customized performance map is still required to predict the abnormal conditions and exert the prevention control for the main compressor with better accuracy.

- The off-design change in thermal input has relatively mild effects on the cycle system control. No operational compressor issues occur when the thermal input deviates from the on-design conditions in the studied ranges regardless of the main compressor configuration. The cycle maximum pressure mode has slight effects on the cycle thermodynamic performance under off-design condition. The cycle in flexible maximum pressure mode is less sensitive to the thermal input variation in $\dot{W}_{net}$ due to the counteract of $\dot{m}_{CO2}$ variation. The maximum variations of $\eta_{en,cycle}$ and $\dot{W}_{net}$ are both within ±5.5% as $T_{salt,in}$ deviates the design point by ±30°C. The $\eta_{en,cycle}$ and $\dot{W}_{net}$ decrease by 15%-22% as $m_{salt}$ decreases to 60% of the design point and increase by 1.0%-1.2% and 1.4%-1.9% as $m_{salt}$ increases to 120% of the design point.

- The selection of $T_{MC,in,d}$ significantly affect the off-design characteristics of the cycle under varied conditions of $T_{MC,in}$. A low $T_{MC,in,d}$ leads to less demanding control task regarding the operational issue prevention as no surge/choke is reported at the design point of $T_{MC,in,d} = 32°C$. The cycle with a high $T_{MC,in,d}$ is likely to have stable thermodynamic performance under off-design conditions. The maximum reductions in $\eta_{en,cycle}$ and $\dot{W}_{net}$ are 12.1% and 17.2% at the design point of $T_{MC,in,d} = 50°C$ and 25.6% and 84.8% at the design point of $T_{MC,in,d} = 32°C$.

- Despite the relatively demanding control task for compressor surge prevention, M-FP appears to be the most satisfactory control scheme for the consequent steady $\eta_{en,cycle}$ and $\dot{W}_{net}$ as well as the effective prevention of operational issue of the main compressor. However, the cycle performance may be further improved when the real-time parametric optimization is applied for the cycle under off-design conditions.

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NOMENCLATURE

Symbol

| Symbol | Description |
|--------|-------------|
| $A$ | area of heat exchanger, m$^2$ |
| $C_p$ | specific heat at constant pressure, kJ kg$^{-1}$ K$^{-1}$ |
| CSP | concentrated solar power |
| $D$ | diameter, m |
| $h$ | specific enthalpy, kJ kg$^{-1}$ |
| $H$ | height, m |

\( T \) relating to the cycle thermodynamic performance as the activity to the is found that the cycle performance exhibits lower sensitivity for the cycle with a fixed maximum pressure. It comparison among the results of the cycle with different characteristics as sure mode has significant effects on the thermodynamic through the sensitivity analyses on thermal input and ambient temperature are investigated compared. The effects of the off-design changes in cycle or/and main compressor configuration are evaluated and operational issue prevention are highlighted. Four control schemes with different cycle maximum pressure mode with respect to the thermodynamic performance and operational issue prevention are highlighted. Four control schemes with different cycle maximum pressure mode or/main compressor configuration are evaluated and compared. The effects of the off-design changes in cycle thermal input and ambient temperature are investigated through the sensitivity analyses on $T_{salt,in}$, $m_{salt}$ and $T_{MC,in}$. Three different $T_{MC,in,d}$ are selected to investigate the effects of the choice of $T_{MC,in,d}$ on the cycle off-design performance. The following conclusions are drawn:

- The compressor surge may occur to the main compressor with basic configuration when the $T_{MC,in}$ attains a certain value lower than the corresponding $T_{MC,in,d}$. By contrast, the surge risk can be prevented with the modified MC configuration by activating the recirculation system and can thus operate normally in the entire off-design range of $T_{MC,in}$. The choke condition does not occur to the cycle under all the off-design conditions, the parallel compressor is not activated for either stage of the main compressor. This indicate that the risk of choke appears controllable with shaft speed control. A customized performance map is still required to predict the abnormal conditions and exert the prevention control for the main compressor with better accuracy.

- The off-design change in thermal input has relatively mild effects on the cycle system control. No operational compressor issues occur when the thermal input deviates from the on-design conditions in the studied ranges regardless of the main compressor configuration. The cycle maximum pressure mode has slight effects on the cycle thermodynamic performance under off-design condition. The cycle in flexible maximum pressure mode is less sensitive to the thermal input variation in $\dot{W}_{net}$ due to the counteract of $\dot{m}_{CO2}$ variation. The maximum variations of $\eta_{en,cycle}$ and $\dot{W}_{net}$ are both within ±5.5% as $T_{salt,in}$ deviates the design point by ±30°C. The $\eta_{en,cycle}$ and $\dot{W}_{net}$ decrease by 15%-22% as $m_{salt}$ decreases to 60% of the design point and increase by 1.0%-1.2% and 1.4%-1.9% as $m_{salt}$ increases to 120% of the design point.

- The selection of $T_{MC,in,d}$ significantly affect the off-design characteristics of the cycle under varied conditions of $T_{MC,in}$. A low $T_{MC,in,d}$ leads to less demanding control task regarding the operational issue prevention as no surge/choke is reported at the design point of $T_{MC,in,d} = 32°C$. The cycle with a high $T_{MC,in,d}$ is likely to have stable thermodynamic performance under off-design conditions. The maximum reductions in $\eta_{en,cycle}$ and $\dot{W}_{net}$ are 12.1% and 17.2% at the design point of $T_{MC,in,d} = 50°C$ and 25.6% and 84.8% at the design point of $T_{MC,in,d} = 32°C$.

- Despite the relatively demanding control task for compressor surge prevention, M-FP appears to be the most satisfactory control scheme for the consequent steady $\eta_{en,cycle}$ and $\dot{W}_{net}$ as well as the effective prevention of operational issue of the main compressor. However, the cycle performance may be further improved when the real-time parametric optimization is applied for the cycle under off-design conditions.

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mass flow rate, kg s^{-1}
N shaft speed, —
p pressure, kPa
Q heat transfer rate, kW
S-CO_2 supercritical carbon dioxide
SR split ratio
T temperature, K/°C
U tip velocity
UA heat transfer conductance, MW/K
V volume, m^3
W work, kW

Greek symbols
k coefficient of heat conductivity, W m^{-1} K^{-1}
Δp pressure drop, kPa
ΔT temperature difference, K/°C
η isentropic efficiency
μ dynamic coefficient of viscosity, kg m^{-1} s^{-1}
ρ density, kg/m^3
ϕ mass flow coefficient, —
ϕ* mass flow coefficient, —
ψ ideal head coefficient, —
ψ* ideal head coefficient, —

Subscript
1,2... state point
cyc cycle
d design point
in inlet
od off-design
out outlet
rec recirculation
S1 stage 1
S2 stage 2
salt molten salt

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