

Performance Optimization of a CO₂ Energy Storage System Coupled with Flywheel Waste Heat Utilization

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Abstract: In the context of large-scale grid integration of renewable energy, conventional CO₂ energy storage systems face two core technical bottlenecks: first, poor tolerance to electrical power input fluctuations, making it difficult to match the intermittent and volatile output characteristics of renewable sources such as wind and solar power; second, high exergy loss in heat exchangers, which significantly restricts the potential for improving overall system energy efficiency. These issues severely limit scale-up applications in long-duration energy storage. To address the sensitivity of traditional CO₂ energy storage systems to power input fluctuations and the high exergy loss in heat exchangers, this paper proposes a coupled energy storage scheme featuring "Flywheel Dynamic Frequency Modulation + Waste Heat Synergistic Exchange." This approach achieves a synergistic solution to the two core problems by utilizing the flywheel to smooth power fluctuations and utilizing cascading waste heat to reduce exergy loss. An Aspen Plus thermodynamic model of the system was constructed, incorporating a Flywheel Permanent Magnet Synchronous Motor (PMSM) dynamic control module and optimizing the internal helical heat exchange structure of the rotating shaft to analyze and verify system performance for efficiency improvement and exergy loss reduction. The results indicate that under the operating conditions of an energy storage pressure ratio of 7.0 and a flywheel speed of 12,000 rpm, the system Round Trip Efficiency (RTE) reaches 77.3%, an increase of 34.7 percentage points compared to pure CO₂ energy storage. The energy storage density is 12.94 kWh/m³, and heat exchanger exergy loss is reduced by 28.6%. Under conditions of $\pm 20\%$ power fluctuation, the flywheel frequency modulation response time is ≤ 0.5 s, and the system output power fluctuation amplitude is $\leq 5\%$, enabling stable adaptation to the fluctuating output characteristics of renewable energy. This coupled system provides a highly efficient and stable technical solution for long-duration energy storage under renewable energy fluctuation conditions and possesses significant engineering application value.

Keywords: Carbon Dioxide Energy Storage; Flywheel Energy Storage; Waste Heat Recovery; Exergy Analysis; Aspen Plus.

1. Introduction

To address the current energy situation, China has explicitly established the strategic goals of "Carbon Peaking by 2030 and Carbon Neutrality by 2060," promoting the large-scale development of renewable energy[1]. As a key technology to meet this challenge, CO₂ energy storage systems have become a preferred direction for renewable energy storage support due to their high energy storage density, controllable storage duration, and environmental friendliness[1]. However, conventional CO₂ energy storage systems face two core technical bottlenecks in practical applications: First, poor tolerance to electrical input fluctuations; when input power fluctuations exceed 10%, system efficiency decay can exceed 15%, making it difficult to adapt to renewable energy output characteristics. Second, the proportion of exergy loss in heat exchangers remains high (typically exceeding 30% of total exergy loss), primarily stemming from mismatched temperature differences between cold and hot fluids and insufficient heat transfer efficiency, which significantly limits the potential for overall system energy efficiency improvement.

CO₂ energy storage technology can be categorized into transcritical, liquid, and supercritical types. Zheng et al.[3] proposed a two-stage Liquid Carbon Dioxide Energy Storage (LCES) system using a pebble-bed packed bed for thermal storage, achieving an energy storage density of 29.46 kWh/m³ and an exergy efficiency of 93.73%, but did not address dynamic regulation. Li et al.[4] optimized supercritical CO₂

fluid properties by matching critical temperature ranges, increasing the system round-trip efficiency to 69.32%, yet heat exchanger exergy loss still accounted for 28.64% of the total exergy loss. Current carbon dioxide energy storage technologies are constrained by limitations in specific subsystems, limiting improvements in storage density and efficiency. Consequently, introducing other novel energy storage technologies for coupling to enhance parameters and operating efficiency has become a new direction of exploration. The application of flywheel energy storage in grid frequency modulation is relatively mature: Cimuca et al. [5] coupled flywheels with variable-speed wind turbines, shortening frequency modulation response time to 0.8 s; Xiao et al.[6] proposed a distributed flywheel cooperative control strategy, enhancing grid stability. In terms of motor technology, Li et al. [7] designed a multi-axis flux PMSM motor, increasing flywheel power density by 20% after optimizing the winding structure; Wang Cheng et al. [8] utilized a water-cooling structure design to control the flywheel motor temperature below 65 °C, ensuring reliability at high rotational speeds. However, existing research has not addressed the "Dynamic Frequency Modulation + Waste Heat Utilization" synergistic coupling of flywheels with CO₂ energy storage, and relevant models and performance analyses remain a gap.

Flywheel energy storage technology has demonstrated unique advantages in the field of grid dynamic regulation due to characteristics such as short-time high-power response, fast frequency modulation speed (millisecond level), and long

cycle life (15~25 years) [9]. Notably, the friction loss and electromagnetic loss generated during the high-speed rotation of the flywheel can be recovered and converted into thermal energy, providing a potential pathway for cascading energy utilization. Based on this, coupling flywheel energy storage with a CO₂ energy storage system to construct a "Kinetic Energy - Thermal Energy - Pressure Energy" multi-energy synergistic conversion mechanism can rapidly smooth input power fluctuations through flywheel dynamic frequency modulation and utilize recovered waste heat to reduce the heat exchanger thermal load. This offers an innovative approach to overcoming the core bottlenecks of traditional CO₂ energy storage systems.

Therefore, this paper proposes a "Flywheel Dynamic Frequency Modulation + Waste Heat Synergistic Exchange" coupling mechanism to resolve the fluctuation sensitivity and high exergy loss issues of pure CO₂ energy storage.

2. System Description

This study proposes a hybrid energy storage scheme where, under normal operating conditions of the carbon dioxide energy storage system, flywheel energy storage is introduced for optimization, and the flywheel structure is improved.

The operating principle establishes the flywheel energy storage as the "power regulation hub," balancing frequency modulation stability and input efficiency maximization goals. Therefore, the flywheel energy storage system is selected as the electrical energy input terminal to achieve synergistic enhancement between the flywheel and carbon dioxide energy storage. During operation, input electrical energy passes uniformly through the flywheel energy storage system, which supplies energy to the carbon dioxide energy storage system. Addressing complex electrical input conditions, the flywheel receives electrical energy and outputs stable power to directly drive the expansion turbine equipment of the carbon dioxide system. During this period, thermal energy generated by flywheel rotation is transported to the carbon dioxide energy storage system via heat exchange through an improved pipeline structure within the rotating shaft, ultimately achieving efficient energy utilization. The flywheel module employs a carbon fiber composite rotor Permanent Magnet Synchronous Motor (PMSM), with a spiral heat exchange pipeline installed inside the rotating shaft to recover friction and electromagnetic losses generated during flywheel operation. The control module utilizes a PI controller to match flywheel speed with CO₂ cycle power, smoothing grid input fluctuations.

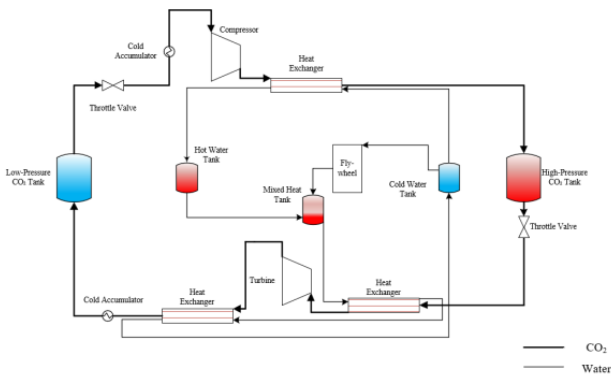


Fig 1. Schematic diagram

(1) Boundary Conditions and Assumptions

To construct the aforementioned system utilizing existing thermodynamic analysis logic for carbon dioxide energy storage systems, the following assumptions are made to simplify the theoretical model during thermodynamic analysis: The system operates in a steady state, ignoring pressure drops within pipes and changes in kinetic and potential energy. The pressure drop flowing through the heat exchanger is set to a minimal value (this paper adopts 1%). The fluid flow processes within the system are regarded as isothermal processes, and compression and expansion processes are regarded as adiabatic. The duration of the energy storage process is set equal to the energy release process. The fluid is assumed to have a constant mass flow rate. Based on these assumptions, the system is constructed as shown in Figure 1.

For the vapor-liquid equilibrium and thermodynamic properties (pressure, volume, enthalpy, entropy, etc.) calculation of CO₂ as a weakly polar fluid, the PENG-ROB model is adopted. Its core formulas and parameter definitions are as follows (Equations 1-3):

$$P = \frac{RT}{V_m - b} - \frac{a\alpha}{V_m^2 + 2bV_m - b^2} \quad (1)$$

Where: P is the absolute pressure of the fluid (Pa); R is the universal gas constant ($8.314 \text{ J} \cdot \text{mol}^{-1} \cdot \text{K}^{-1}$); T is the absolute temperature of the fluid (K); V_m is the molar volume of the fluid ($\text{m}^3 \cdot \text{mol}^{-1}$); a is the pure component parameter related to intermolecular attraction ($\text{Pa} \cdot \text{m}^6 \cdot \text{mol}^{-2}$); b is the pure component parameter related to molecular volume ($\text{m}^3 \cdot \text{mol}^{-1}$); α is the temperature correction factor, related to fluid critical properties and the acentric factor.

$$\alpha = [1 + \kappa(1 - \sqrt{T_r})]^2 \quad (2)$$

$$\kappa = 0.37464 + 1.54226\omega - 0.26992\omega^2 \quad (3)$$

Where: T_r is the reduced temperature (dimensionless), $T_r = T/T_c$; ω is the fluid acentric factor. For CO₂, $\omega = 0.228$ is obtained from the NIST database.

The system parameter settings involved in this paper are shown in Table 2:

Table 1. System Initial Parameter Settings

Parameter	Unit	Value
Ambient Temperature	K	298.15
Ambient Pressure	kPa	101.13
Compressor Inlet Temperature	K	305.15
SCO ₂ Mass Flow Rate	kg/s	10
High-Pressure CO ₂ Tank Temperature	K	308.15
Compressor Isentropic Efficiency	%	88
Turbine Isentropic Efficiency	%	93
Compressor/Turbine Mechanical Efficiency	%	98

(2) Mathematical Model Construction

The compressor assumes adiabatic compression, with adiabatic efficiency defined by Equation (4):

$$\eta_c = \frac{h_{c,s}^{out} - h_c^{in}}{h_c^{out} - h_c^{in}} \quad (4)$$

Where: h_c^{in} and h_c^{out} are compressor inlet and outlet

enthalpies (kJ/kg); $h_{C,S}^{out}$ is the theoretical outlet enthalpy under isentropic compression.

The heat exchanger uses a shell-and-tube model, with heat load defined by Equation (5):

$$Q_i = m_{hot}(h_{hot,in} - h_{hot,out}) = m_{cold}(h_{cold,out} - h_{cold,in}) \quad (5)$$

Where: hot and cold represent hot and cold heat exchange fluids, respectively.

Flywheel energy storage and dynamic frequency modulation formulas are given by Equations (6-8):

$$E = \frac{1}{2} J n^2 \quad (6)$$

$$J = \frac{1}{2} M (R_2^2 - R_1^2) \quad (7)$$

$$\Delta n = \sqrt{\frac{2 \Delta P \cdot \Delta t}{J}} \quad (8)$$

Where: E is flywheel energy storage (kJ); J is the moment of inertia (kg·m²); n is flywheel speed (rad/s); M is rotor mass (50 kg); R₁, R₂ are rotor inner and outer radii (0.2 m, 0.5 m); ΔP is grid power fluctuation value (kW); Δt is response time (0.1 s).

Waste Heat Generation Model: Flywheel waste heat primarily originates from friction loss and electromagnetic loss, calculated by Equations (9-11):

$$P_{friction} = \mu M g \cdot \frac{n}{60} \cdot 2\pi R_2 \quad (9)$$

$$P_{em} = I^2 R \quad (10)$$

$$Q_{FW} = P_{elec} - P_{FW} - P_{friction} - P_{em} \quad (11)$$

Where: $P_{friction}$ is friction loss (kW); μ is the friction coefficient (0.001); g is gravitational acceleration (9.8m/s²); P_{em} is electromagnetic loss (kW); I is motor current (A); R is winding resistance (0.5Ω); P_{elec} is grid input energy (kW); P_{FW} is flywheel output power (kW); Q_{FW} is total waste heat (kW).

Waste Heat Exchange Model: Utilizing spiral pipe heat exchange, the recovered waste heat is calculated by Equation (12):

$$Q_{recover} = K A (\Delta t_m) \quad (12)$$

Where: $Q_{recover}$ is recovered waste heat (kW); A is heat exchange area (0.5m²); Δt_m is the logarithmic mean temperature difference (K).

Regarding control strategy, collaborative control theory is adopted. The flywheel energy storage system, CO₂ energy storage subsystem, and thermal energy recovery subsystem are treated as interconnected sub-modules, achieving synergistic enhancement through information interaction and parameter coordination.

The PI controller continuous control law is given by Equation (13):

$$u(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau \quad (13)$$

Where: $u(t)$ is the controller output regulation variable, K_p is the proportional coefficient, K_i is the integral coefficient, $e(t)$ is the deviation between the controlled variable and the setpoint, and τ is the integral variable.

Power balance analysis during charge/discharge phases is given by Equations (14-15):

$$P_{in} = P_{f_charge} + P_{loss} \quad (14)$$

$$P_{f_discharge} = P_t + P_{heat_loss} \quad (15)$$

Where: P_{in} is grid input power, P_{f_charge} is flywheel charging power, P_{loss} is total system loss, $P_{f_discharge}$ is flywheel discharging power, P_t is CO₂ turbine input power, and P_{heat_loss} is heat exchange pipeline thermal loss.

(3) Evaluation Criteria

Round Trip Efficiency (RTE): The ratio of net output power during the release phase to the input power during the storage phase. It is a critical indicator for evaluating energy storage system efficiency performance, calculated by Equation (16):

$$RTE = \frac{P_t \times P_f}{P_c} \quad (16)$$

Where: P_t is expander output power (MW); P_f is flywheel total consumed power (MW); P_c is compressor total consumed power (MW).

Energy Storage Efficiency: Intuitively reflects the performance of the storage system. As one of the important evaluation indicators, it is calculated by Equation (17):

$$\eta = \frac{\int_0^{t_{dis}} W_T dt}{\int_0^{t_{ch}} W_C dt + \sum E_{PCHE,heat} t_{dis}} \quad (17)$$

Where: Subscript PCHE, heat represents the heater during the energy release process; subscripts ch and dis represent energy storage and release processes respectively; t represents time (s).

System Exergy Efficiency: Calculated by Equation (18):

$$\eta_E = 1 - \frac{\sum \Delta E}{E_{electricity} + E_{thermal}} \quad (18)$$

Where: η_E is the system exergy efficiency; $E_{electricity}$ and $E_{thermal}$ are total electrical exergy and total thermal exergy inputs to the system (kW). In this paper, neglecting thermal loss from the flywheel system to the outside, the total thermal exergy of the heater is the system input total thermal exergy.

Energy Storage Density: Reflects the energy storage capability of the working fluid, calculated by Equations (19-21):

$$\sigma = \frac{\int_0^{t_{dis}} W_T dt}{V_{HPV} + V_{LPV}} \quad (19)$$

$$V_{HPV} = \frac{m_C t_{ch}}{\rho_{HPV}} \quad (20)$$

$$V_{LPV} = \frac{m_C t_{ch}}{\rho_{LPV}} \quad (21)$$

Where: Subscripts HPV and LPV represent high-pressure and low-pressure storage tanks respectively; V is storage tank volume (m³); ρ is the density of CO₂ in the corresponding tank (kg/m³).

3. Results and Discussion

(1) Model Validation Results

To ensure the precision of subsequent research results, the accuracy of system components was verified. Based on relevant results in Reference[10], steady-state verification was conducted for the compressor and turbine. The specific validation results are shown in Table 1. During verification, parameters such as inlet temperature, inlet pressure, and outlet pressure were inputs, while the working fluid outlet temperature was the output parameter. The results show that the maximum relative error of the aforementioned

components is only 1.1%, confirming that the established compressor and turbine models possess reliable accuracy. The primary cause of the error is that under the verification conditions, the turbine operating temperature is relatively high, leading to a deviation between the isentropic efficiency set in this model and the actual operating conditions.

Table 2. Parameter Comparison

Parameter	Experimental Data[10]	Model Calculation Data	Error (%)
Cooler Outlet Temp	323.15K	316.7K	2
Compressor Power	60MW	58.9MW	1.8
Turbine Power	290MW	306.8MW	5.8

(2) Influence of Key Parameters on System Performance

The energy storage pressure ratio refers to the ratio relationship between specific pressure parameters in the context of the energy storage system. This ratio reflects the impact of pressure changes on the energy storage effect during energy conversion and storage. A higher energy storage pressure ratio implies that more energy can be stored within the same storage space, but it also imposes higher requirements on equipment pressure resistance.

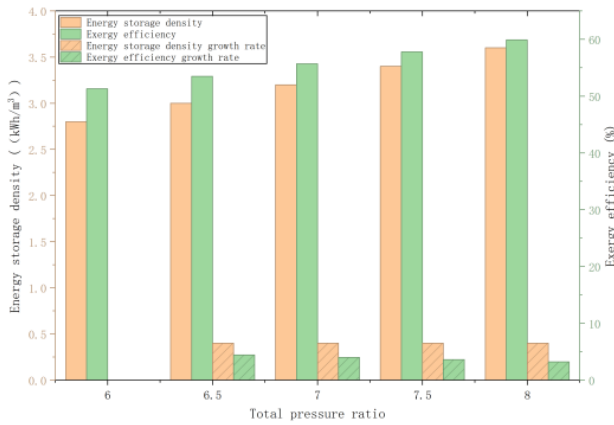


Fig 2. Effect of Energy Storage Pressure Ratio on Energy Storage Efficiency

From a system perspective, the energy storage pressure ratio aids in assessing the energy conversion efficiency, operational stability, and compatibility with other system components of the entire storage device.

In the total pressure ratio range of 6~8, the energy storage density of the CO₂ storage system increases linearly from 2.8kWh/m³ to 3.6kWh/m³, with the growth rate stabilizing at 0.4kWh/(m³unit pressure ratio). Exergy efficiency increases from 51.3% to 59.9%, with the growth rate decreasing gently from 4.4%/unit pressure ratio to 3.2%/unit pressure ratio, exhibiting characteristics of "uniform growth in storage density, slowly decreasing growth in exergy efficiency." Balancing "Efficiency - Storage Density - Cost," a total pressure ratio of 7.0 is the optimal operating point. At this point, storage density reaches 3.2kWh/m³, exergy efficiency is 55.7%, and the composite index (Storage Density × Exergy Efficiency) reaches 178.24. Furthermore, the match between equipment investment (high-pressure components, tanks) and performance gain is optimal. If higher storage density is required, a total pressure ratio of 7.5 is the suboptimal choice, where storage density and exergy efficiency increase to 3.4kWh/m³ and 57.8% respectively, but this necessitates approximately 15% additional equipment investment.

After the pressure ratio exceeds 7.0, the growth trend of RTE slows (from 7.0 → 8.0, RTE increases from 77.3% to 79.1%), and the marginal gain in storage density decreases (density growth drops from 32% to 5% per 1.0 pressure ratio increase). The fundamental reason is that CO₂ approaches a supercritical state under high pressure, where density changes are gradual, while compressor adiabatic compression work surges. The growth rate of input work (20%/1.0 pressure ratio) exceeds that of turbine output work (12%/1.0 pressure ratio).

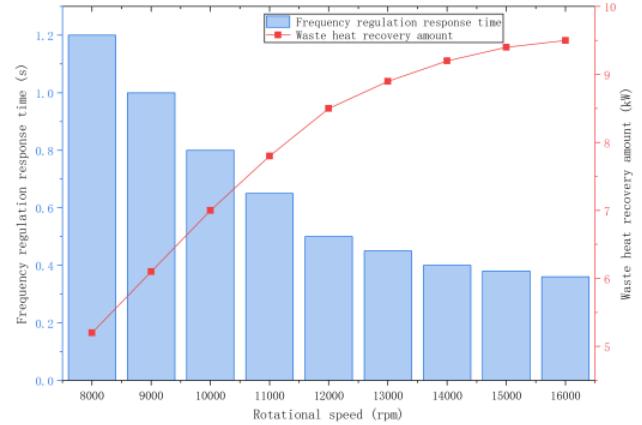


Fig 3. Influence of Flywheel Speed on Frequency Modulation Response Time and Waste Heat Recovery

As illustrated in Figure 7, when the rotational speed increases from 8,000 rpm to 12,000 rpm, the frequency modulation response time decreases from 1.2 s to 0.5 s, and the waste heat recovery amount increases from 5.2 kW to 8.5 kW. This is because flywheel energy storage is proportional to the square of the speed; thus, energy regulation capability and loss recovery amount increase synchronously at high speeds. When the speed exceeds 12,000 rpm, the response time only shortens from 0.5 s to 0.4 s, and waste heat recovery increases from 8.5 kW to 9.2 kW. However, motor winding losses increase by 35%, resulting in a net system efficiency improvement of less than 1%. Therefore, 12,000 rpm is the optimal flywheel speed, balancing frequency modulation performance and energy consumption costs.

(3) Exergy Loss Analysis and Optimization

Using Aspen Plus, the exergy loss of each part of the energy storage system was calculated for a given pressure ratio of 7.0. Partial results of input and output segment information selected from each component are shown in Table 3.

Table 3. System Partial Thermodynamic Data

No.	Air	Temp T/K	Pressure p/MPa	Enthalpy h/kJ•kg ⁻¹	Entropy s/kJ•(kg•K) ⁻¹
1	CO ₂	305.15	8.00	-8952.049	-0.513
2	CO ₂	343.64	9.16	-8922.551	-0.500
3	CO ₂	307.01	9.13	-8962.455	-0.620
4	CO ₂	348.79	10.07	-8934.043	-0.608
5	CO ₂	308.22	9.89	-8986.227	-0.765

Integrating and calculating the results, as seen in Figure 9, the component with the highest exergy loss in the system is the heat exchanger, with a value of 346.17kW. This is mainly due to the large temperature glide between cold and hot fluids and the influence of physical properties on heat exchange effectiveness, leading to excessively high hot-side exhaust temperatures. Additionally, the exergy loss of the turbine is also relatively high, primarily because the working fluid

temperature differs significantly from the ambient temperature; its exergy loss is mainly utilized to improve equipment efficiency. Under this operating condition, the calculated total exergy loss is 544.72 kW, input exergy is 2119.24 kW, resulting in a system exergy efficiency of 74.2%.

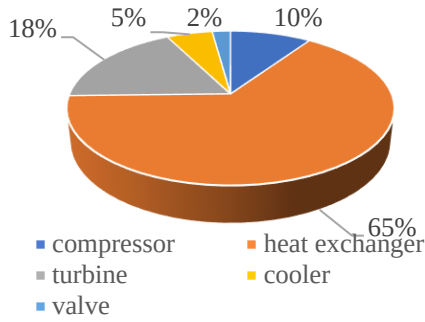


Fig 4. Statistical Analysis of Main Equipment Exergy Loss Distribution

By introducing the flywheel structure, energy is supplied from the flywheel to the carbon dioxide energy storage system. During this period, thermal energy generated by flywheel rotation can be transported to the carbon dioxide energy storage system via the improved pipeline structure for heat exchange, thereby increasing heat exchanger efficiency. Calculating the carbon dioxide energy storage system with the introduced flywheel structure at a storage pressure ratio of 7, and analyzing the exergy loss of each component, the results are shown in the figure below. Under the same working fluid input, total exergy loss decreased from 544.72 kW to 473.41 kW. This optimization is mainly attributed to the utilization of flywheel rotation waste heat. From the perspective of system exergy efficiency, after introducing the flywheel structure, the input exergy is 1986.53 kW, meaning energy input is reduced to a certain extent, thereby increasing system exergy efficiency to 76.2%.

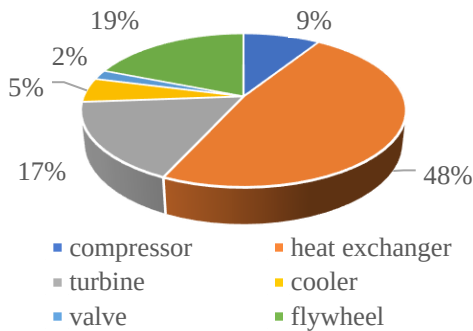


Fig 5. Statistical Analysis of Main Equipment Exergy Loss Distribution (with Flywheel Structure)

Based on the above analysis, the recuperator in the system has significant room for improvement. By adding a secondary compressor for split-flow compression and dividing the recuperator into a pre-recuperator and a re-recuperator, the split ratio is analyzed when heating temperatures are 513.15 K, 543.15 K, and 573.15 K, as shown in Figure 13, to enhance the heat exchange effect in this section.

Energy storage efficiency initially increases and then decreases with the increase of the split ratio. However, the optimal split ratio is related to the heating temperature. For 573.15 K, the energy storage efficiency is highest at 75.4% when the split ratio is 0.15. Simultaneously, it is observed that

after adding the re-recuperation mechanism, the system exhibits higher energy storage efficiency at any split ratio compared to the corresponding original system. The reason is that a smaller split ratio can increase the temperature of the working fluid recovered in the pre-recuperator, thereby reducing the heating load and input thermal exergy. Conversely, when the split ratio is larger, the increased power consumption of the compressor due to high-temperature compression has a more significant impact on energy storage efficiency.

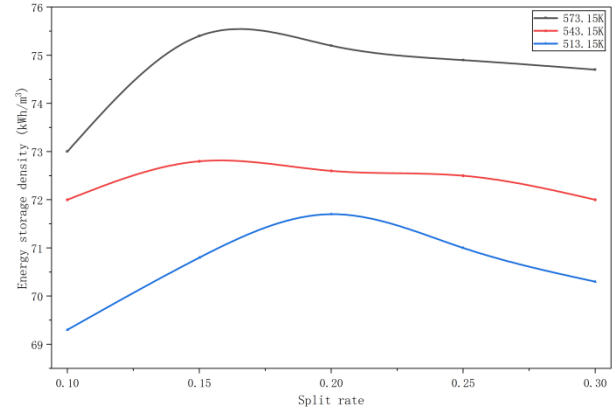


Fig 6. Effect of Split Ratio on Energy Storage Efficiency of System at Different Temperature

Calculating the exergy loss distribution statistics for each piece of equipment in the system given an energy storage pressure ratio of 7.0, a heating temperature of 573.15 K, and a split ratio of 0.15, it is evident that the total exergy loss of the improved system is significantly reduced. This is mainly due to the improvement of the recuperator. The exergy loss of the pre-recuperator is 23.27 kW, and the re-recuperator is 239.15 kW. The system exergy efficiency is improved by reducing energy input.

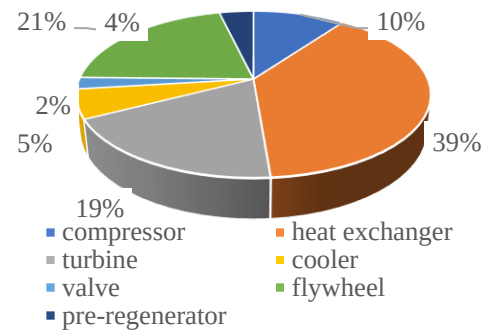


Fig 7. Statistical Analysis of Main Equipment Exergy Loss Distribution after Improvement

In addition, Printed Circuit Heat Exchangers (PCHE) are adopted to replace traditional shell-and-tube heat exchangers. By utilizing micro-channel flow paths (diameter 1~2 mm) to reduce the temperature pinch point, it is estimated that heat exchanger exergy loss can be reduced by 15%~20%. Optimizing the flow layout of cold and hot fluids improves the match of the T-Q curves between CO₂ and the thermal storage medium, narrowing the heat exchange temperature difference to within 10 °C, further reducing irreversible losses.

(4) Performance Comparison with Similar Systems

Common carbon dioxide energy storage strategies and their parameters affecting energy storage capability (RTE, Storage Density, Frequency Modulation Response Time, Exergy

Efficiency) were selected for comparison. The results are shown in Table 4.

Table 4. Comparison Results

Coupling Strategy	RTE/%	Storage Density /(kWh · m ⁻³)	FM Response Time /s	Exergy Efficiency /%
CO ₂ +Organic Rankine Cycle	56.67	36.12	-	-
Liquid CO ₂ Storage	61.54	21.81	-	-
CO ₂ +Two-Stage Cold/Heat Storage	56.12	29.46	-	93.73
No Coupling	57.4	12.94	-	74.2
CO ₂ +Flywheel	77.3	12.94	0.5	76.2

The RTE of the system in this paper (77.3%) is significantly higher than existing similar systems, and it possesses millisecond-level frequency modulation response capability, offering distinct advantages in adaptability to fluctuating conditions. Although the energy storage density is lower than that of liquid CO₂ energy storage systems, the comprehensive efficiency and stability are superior, making it suitable for renewable energy supporting storage scenarios.

4. Conclusion

Aiming to improve operating efficiency and energy storage density, this paper introduced a flywheel structure to the conventional carbon dioxide energy storage system and performed parameter optimization on the overall system. The influence laws of main system performance parameters and evaluation indicators were analyzed from a thermodynamic perspective, leading to the following conclusions:

(1) A "Flywheel Dynamic Frequency Modulation + Waste Heat Synergistic Exchange" coupled energy storage system was proposed. Under selected optimal conditions (energy storage pressure ratio 7.0, flywheel speed 12,000 rpm, split ratio 0.15), the system RTE reached 77.3%, an increase of 34.7% compared to pure CO₂ energy storage; energy storage density was 12.94 kWh/m³, and exergy efficiency was 76.2%. This effectively mitigating the issues of sensitivity to input fluctuations and high heat exchanger exergy loss in traditional CO₂ energy storage.

(2) The introduction of the flywheel module endows the system with excellent frequency modulation performance: under ±20% power fluctuation conditions, response time is ≤0.5 s, and output power fluctuation amplitude is ≤5%, meeting grid frequency modulation technical requirements. Simultaneously, flywheel waste heat recovery reduces heat exchanger exergy loss by 28.6% and total exergy loss by 13.1%.

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