

Article

Fluid-Driven Opposed-Piston Pumps for Dense-Phase CO₂ Injection: Direct Force Coupling and Energy Efficiency Analysis

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Abstract

Large-scale dense-phase carbon dioxide (CO₂) injection is an energy-intensive process in the carbon capture, utilization, and storage (CCUS) value chain. To address insufficient utilization of inlet pressure potential energy and sealing/friction losses of conventional reciprocating pumps under high-base-pressure dense-phase CO₂ transport conditions, this study develops a dense-phase CO₂-oriented structural optimization scheme for a hydraulically driven opposed-piston reciprocating pump based on force-coupling. A dynamic model was established to clarify the in situ recovery mechanism by which inlet pressure potential energy is converted into auxiliary thrust, enabling the drive load to shift from absolute pressure to net pressure difference. Simulation results show that under the rated 8 MPa inlet and 25 MPa discharge condition, the optimized opposed-piston configuration reduces peak driving oil pressure by 31.39% compared with the non-opposed reference configuration. Field reliability operation data show an average normalized specific energy consumption of 0.422 kWh/(MPa·m³) during the selected 24 h continuous operating period. The optimized configuration improves inlet-pressure utilization and reduces hydraulic power demand under high-base-pressure dense-phase CO₂ injection conditions, providing theoretical support and engineering reference for low-energy CCUS injection systems.

Keywords: dense-phase CO₂ injection; opposed reciprocating pump; structural optimization; suction-pressure-assisted load compensation; CCUS



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1. Introduction

As the commercialization of large-scale CCUS technology accelerates, high-pressure fluid compression equipment—serving as the core power unit of the entire CCUS chain [1–3]—naturally makes energy efficiency and operational reliability the most critical factors determining the system’s economic viability over its entire lifecycle [4,5]. More importantly, dense-phase CO₂ exhibits several characteristic properties, including low viscosity, compressibility, and thermal sensitivity [6–8]. Consequently, traditional gas injection equipment faces significant challenges regarding flow assurance and operational stability when transporting dense-phase CO₂ over long distances [9–11]. Currently,

research in both academia and industry on this issue has clearly focused on three directions: performance optimization of traditional three-plunger reciprocating pumps, fundamental research on the mechanisms of high-pressure CO₂ fluid transport, and innovative design of new hydraulic drive topologies.

Thanks to their proven manufacturing processes and high-pressure output capabilities, traditional multi-plunger reciprocating pumps have long held a dominant position in the field of pressure-boosted injection [12,13]; however, the inherent kinematic characteristics of the crank-connecting rod mechanism make pressure pulsations the most direct and fundamental cause of pipeline vibration [14]. Aakre et al. [14,15] provided a clear and logical explanation of the mechanism by which multi-cylinder configurations suppress pressure pulsations: although multi-cylinder designs theoretically help offset some higher-order harmonics, the compressibility of the fluid when transporting supercritical CO₂ significantly amplifies the pulsation amplitude; therefore, passive buffering devices are generally required to mitigate this effect.

More importantly, traditional mechanical drive trains exhibit significant friction-related deficiencies when handling high-pressure fluids [16]. Li et al. [17] conducted an extremely thorough and rigorous analysis of the mechanisms of internal friction power loss in plunger pumps, clearly and accurately identifying the periodic lateral forces generated by the connecting rod's angular displacement as the root cause of seal wear and reduced mechanical efficiency. For low-viscosity CO₂ media, this lateral load disrupts the normal formation of the oil film, leading to a sharp increase in boundary friction, which severely compromises both pump lifespan and energy efficiency. Even more concerning is that the unique thermophysical behavior of CO₂ itself causes volumetric efficiency to deteriorate sharply with changes in operating conditions [18–20]. Unlike conventional fluids, the density and viscosity of dense-phase CO₂ are highly sensitive to thermodynamic parameters [21,22]. Ma et al. [23,24] provide a highly intuitive and convincing explanation for this: the high compressibility of CO₂ inevitably results in a pronounced “compression lag” effect, causing the effective displacement to decrease significantly.

Due to the limitations of mechanical drive chains in terms of dynamic characteristics and energy efficiency, opposed-piston and hydraulic-drive configurations have been widely investigated in recent years. Liu et al. [25,26] analyzed the pressure-bearing capacity and energy-efficiency characteristics of opposed-piston reciprocating pumps, showing that opposed layouts can improve load balance and operating stability under suitable conditions. Li et al. [27] designed a novel hydraulic oil-extraction pump unit suitable for oilfield conditions, systematically and thoroughly demonstrating that a symmetrical structural layout facilitates mutual balancing of hydraulic-end loads, thereby offering superior energy efficiency compared to traditional units. More notably, Qi et al. [28] proposed an active control scheme based on hydraulic oil supply regulation to address the issue of pressure mismatch during multistage compression, successfully achieving adaptive balancing of inter-stage pressure ratios and significantly enhancing the system's thermodynamic stability.

Current studies on hydraulic and double-acting reciprocating pumps mainly focus on general-purpose applications, such as flow/pressure pulsation suppression, dynamic response optimization, and load balancing [29]. However, the use of opposed-cylinder force coupling to utilize the 6–8 MPa suction pressure of dense-phase CO₂ as a load-compensation term during high-pressure injection has not been sufficiently clarified. In this scenario, the inlet pressure of pipeline CO₂ can reach 6–8 MPa, while conventional triplex plunger pumps and reciprocating pumps usually suffer from insufficient utilization of inlet pressure, high driving energy consumption, and limited pump efficiency. Therefore, low-energy CO₂ pressurization and injection remain key technical challenges.

To address these issues, this study develops a dense-phase CO₂-oriented structural optimization of a conventional hydraulically driven opposed-piston reciprocating pump. The optimized configuration introduces force-transmission cylinders, vertical CO₂ cylinders, rodless pistons, and an ionic-liquid isolation layer to improve force transmission, sealing performance, and volumetric efficiency under high-base-pressure injection conditions.

In this configuration, the high suction pressure of pipeline CO₂ is introduced into the force balance of the opposed piston assembly and provides auxiliary thrust for the compression process of the opposite cylinder. Consequently, the hydraulic load is transformed from a discharge-pressure-dominated load to an inlet-outlet net-pressure-difference-dominated load, thereby reducing the required hydraulic driving pressure and energy input. The force-transmission cylinders further convert horizontal hydraulic actuation into vertical CO₂ compression, providing the structural basis for stable ionic-liquid coverage above the rodless piston under low-viscosity dense-phase CO₂ conditions.

In summary, this paper presents a structural optimization and performance evaluation of a hydraulically driven opposed-piston reciprocating pump for high-base-pressure dense-phase CO₂ injection. The work focuses on suction-pressure-assisted load compensation, vertical force transmission through force-transmission cylinders, and ionic-liquid-assisted sealing. By combining dynamic modeling, AMESim simulation, and field reliability operation data, this study supplements existing opposed-piston pump research and provides theoretical and engineering references for high-base-pressure fluid injection systems.

- (1) A suction-pressure-assisted load-compensation mechanism is established for high-base-pressure dense-phase CO₂ injection. The high inlet pressure of 6–8 MPa inherent to dense-phase CO₂ in the pipeline is utilized to participate in the force balance of the opposite compression cylinder, thereby reducing the hydraulic load required for discharge.
- (2) A structural optimization scheme is developed by integrating a horizontal hydraulic cylinder, intermediate force-transmission cylinders, vertical CO₂ cylinders, rodless pistons, and ionic-liquid isolation. This layout converts horizontal hydraulic actuation into vertical CO₂ compression, reduces lateral loading on the sealing interface, and provides the structural basis for improved sealing and volumetric efficiency.
- (3) The energy-saving effect of the optimized opposed-piston configuration under high-base-pressure conditions is evaluated by dynamic modeling, AMESim simulation, and field reliability operation data. The results clarify how the suction pressure of dense-phase CO₂ participates in the force balance of the opposite compression cylinder and reduces the hydraulic power demand.

2. Structural Design

2.1. Structural Characteristics of the Opposed Reciprocating Pump

Figure 1 shows the overall structure of a hydraulically driven opposed-type dense-phase CO₂ reciprocating pump. The characteristics of this pump during the transmission and pressurization processes can be summarized as follows:

- (1) During the pressurization process, the hydraulic direct-drive mechanism replaces the traditional crank–connecting rod mechanism. By adapting the inherent force-coupling characteristics of the opposed structure to high-base-pressure dense-phase CO₂ injection, the inlet pressure is introduced as an auxiliary thrust for the opposite compression chamber, thereby improving the utilization of suction-pressure potential energy. The hydraulic transmission also simplifies the mechanical structure and reduces lateral-force-related wear associated with crank–connecting rod mechanisms under high-pressure conditions.

- (2) The design combines a rodless piston with ionic fluid isolation, reliably physically separating the hydraulic oil from the dense-phase CO₂ medium and preventing medium contamination. Meanwhile, the relatively high viscosity of the ionic liquid provides auxiliary sealing and lubrication, helping to mitigate leakage and dry-friction problems caused by the low viscosity of dense-phase CO₂.
- (3) Considering the control characteristics of the hydraulic system and the compressibility of dense-phase CO₂, the piston motion follows a constant-speed and smooth-reversal profile. Simulation results indicate that this hydraulic constant-speed profile is more stable than a conventional sinusoidal motion profile. Combined with the force-coupling mechanism, the hydraulic cylinder mainly compensates for the inlet–outlet net pressure load, which reduces the required driving pressure and supports efficient high-pressure injection.

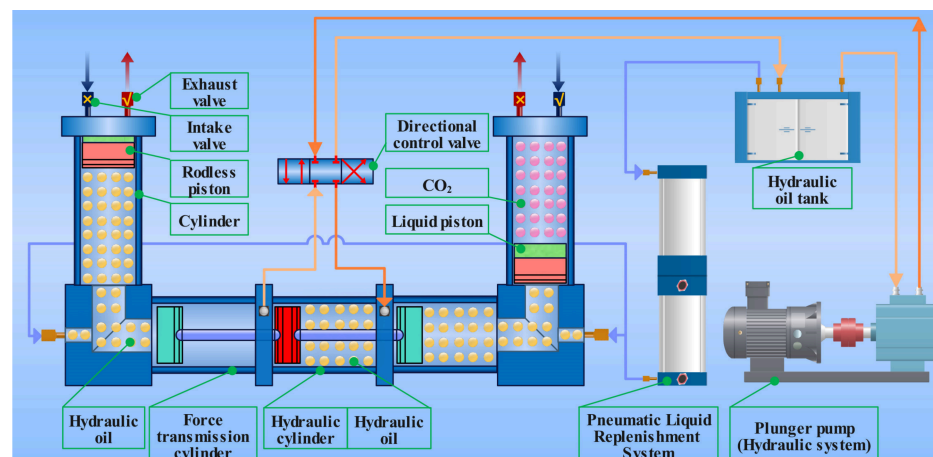


Figure 1. Schematic diagram of the opposed reciprocating pump.

2.2. Detailed Structural Design

Based on the conventional opposed-piston layout, a single-stage dense-phase CO₂ injection pump structure was optimized by introducing force-transmission cylinders, vertical CO₂ cylinders, rodless pistons, and an ionic-liquid isolation layer. Figure 1 shows the detailed structure of the core unit of the injection pump, which mainly consists of two parts: a hydraulic drive unit and an opposed pressurization structure. The entire pump body adopts a modular design to meet the delivery requirements of high-pressure dense-phase fluids. The pressurization section of the core unit includes a main hydraulic cylinder, a force transmission cylinder, a CO₂ cylinder, a rodless piston assembly, and corresponding sealing assemblies.

Taking a single reciprocating motion as an example, the pressurization process is driven by the double-acting hydraulic cylinder in the middle. When the piston rod of the hydraulic cylinder moves to one side, it pushes the hydraulic oil in the force transmission cylinder on that side into the bottom of the vertical CO₂ cylinder, thereby driving the rodless piston upward to compress the dense-phase CO₂. At the same time, the opposite CO₂ cylinder is in the suction stroke, and the high-pressure intake gas pushes the piston downward, assisting the retraction of the main hydraulic cylinder through hydraulic transmission, thus achieving direct force coupling.

As shown in Figure 1, the overall structure of the drive system is a hydraulically direct-driven opposed mechanism. The middle hydraulic cylinder is connected to the force-transfer pistons on both sides through a rigid piston rod, realizing bidirectional power transmission. This arrangement reduces lateral loading, improves force balance, and maintains a compact structural layout.

The force-transmission cylinder is the key component that links the horizontal hydraulic cylinder and the vertical CO₂ cylinder. During operation, the central hydraulic cylinder drives the piston in the force-transmission cylinder, and the hydraulic oil in this cylinder transmits pressure to the bottom of the rodless piston in the vertical CO₂ cylinder. In this way, the horizontal hydraulic drive is converted into the vertical compression motion of CO₂ without using a crank-link mechanism. This arrangement allows the CO₂ cylinder to be installed vertically, providing the structural basis for maintaining a stable ionic-liquid layer above the rodless piston.

During normal operation, to address the problems of low viscosity and poor lubricity of dense-phase CO₂, a layer of ionic liquid is arranged above the rodless piston in the vertical cylinder. As a liquid seal and lubricant, the ionic liquid helps suppress gas leakage and improve the friction condition between the piston and the cylinder wall. Meanwhile, the chemical stability of the ionic liquid must be ensured to prevent reaction or emulsification with high-pressure CO₂.

Considering the possible dissolution of CO₂ in ionic liquids under high pressure [30], the ionic liquid layer should not be regarded as an independent energy-saving mechanism in this study. Its function is limited to sealing, lubrication, and medium isolation. During cyclic pressure variation, CO₂ desorption may disturb the gas–liquid interface and increase the risk of droplet entrainment. Therefore, the ionic liquid should be selected with low CO₂ solubility, high viscosity, low volatility, and good chemical compatibility with dense-phase CO₂. In addition, the vertical cylinder arrangement, sufficient liquid-layer height, and discharge-side gas–liquid separation are required to suppress liquid carryover through the discharge valve.

Figure 2 shows the schematic diagram of the hydraulic control system for the injection pump. To ensure smooth piston reversal at each stroke and improve the utilization of suction-pressure potential energy, the hydraulic system is designed with inlet–outlet pressure-difference sensing and commutation logic.

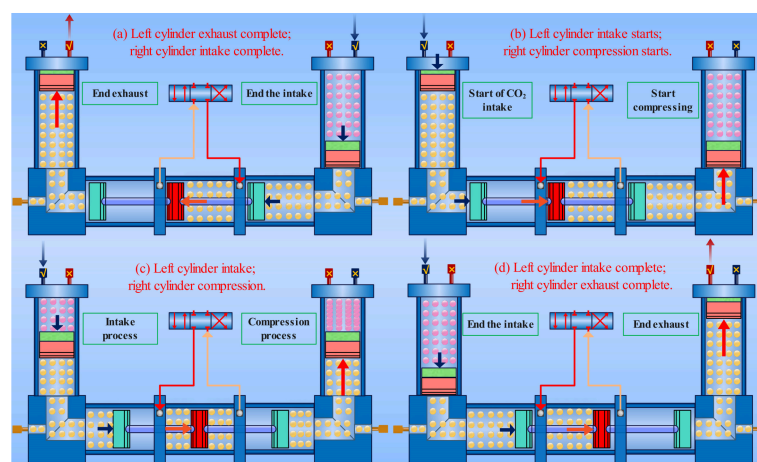


Figure 2. Schematic diagram of the injection pump operating process. (a) Left cylinder exhaust complete; right cylinder intake complete, (b) Left cylinder intake starts; right cylinder compression starts, (c) Left cylinder intake; right cylinder compression, (d) Left cylinder intake complete; right cylinder exhaust complete.

Figure 2a shows that the left cylinder has just completed the discharge process, and the pressurization of the right cylinder begins. As shown in Figure 2b, the hydraulic power unit starts, and high-pressure oil enters the left chamber of the middle hydraulic cylinder, pushing the piston rod of the hydraulic cylinder to move rightward. At the same time, the intake valve of the left cylinder opens to start the intake process.

As shown in Figure 2c, the rodless piston in the left vertical cylinder moves downward under the intake pressure of 6–8 MPa. Through the hydraulic oil medium below the piston, the force is applied to the piston of the left force transmission cylinder, which couples with the thrust provided by the hydraulic oil of the hydraulic cylinder to jointly drive the hydraulic cylinder piston to move rightward. The oil in the right force transmission cylinder is squeezed into the right vertical cylinder, pushing the rodless piston upward to compress the CO₂ in the cylinder.

As shown in Figure 2d, the piston of the left cylinder reaches the bottom dead center to complete the intake process, while the piston of the right cylinder reaches the top dead center to complete the discharge process synchronously. The piston of the hydraulic cylinder and the pistons of both force transmission cylinders all reach the rightmost ends of the corresponding cylinders, triggering the position sensor at the right end of the hydraulic cylinder. At this moment, the directional control valve starts to operate, and the flow direction of the hydraulic oil is switched to “outlet left, inlet right”, initiating the compression process of the left cylinder and the intake process of the right cylinder. The specific technical parameters are listed in Table 1.

Table 1. Main technical parameters of the hydraulically driven opposed reciprocating pump.

Parameter	Value	Unit
Cylinder stroke frequency	10.19	cpm
Rated Inlet Pressure	6.00~8.00	MPa
Rated Discharge Pressure	25.00	MPa
Design Flow Rate	15.00	m ³ /h
Motor Frequency	50.00	Hz
Rated Motor Power	110.00	kW
Driving Medium	Hydraulic Oil	/
Working Fluid	Dense-phase CO ₂	/

For high-base-pressure dense-phase CO₂ injection, the essential energy-saving feature of the optimized structure is that the suction stroke of one CO₂ cylinder is synchronized with the compression stroke of the opposite cylinder. The 6–8 MPa inlet pressure acts on the intake-side piston and is transmitted through the force-transmission cylinder and coupled piston assembly, thereby providing auxiliary thrust for the opposite compression process. As a result, the hydraulic actuator mainly compensates for the inlet-outlet net pressure load rather than the full discharge-pressure load, which reduces the required driving pressure and hydraulic power input.

To prevent the rodless piston from striking the cylinder head or cylinder bottom at high speed at the end of the stroke, a cushion circuit is integrated into the hydraulic system. An air-controlled oil replenishment system is installed on both sides of the force transmission cylinder (see Figure 1), which can monitor and supplement the hydraulic oil pressure in the force transmission cylinder in real time.

2.3. Structural Differences from the Non-Opposed Reference Pump

A conventional non-opposed reciprocating pump is shown in Figure 3. Compared with the non-opposed reference pump, the optimized opposed-piston configuration offers the following advantages under dense-phase CO₂ injection conditions:

- (1) Conventional non-opposed hydraulic reciprocating pumps typically consist of a single hydraulic cylinder and a single CO₂ cylinder connected in series. Although this design is simpler than a mechanical crank mechanism, it has low space utilization, discharges gas only during half a working cycle, and exhibits significant output pulsation from the single cylinder. The optimized opposed-piston structure retains the alternating two-cylinder operation of conventional opposed pumps, while introducing force-

- transmission cylinders and vertical CO₂ cylinders to adapt the layout to dense-phase CO₂ injection. This configuration increases the effective displacement, improves space utilization, and enhances the continuity of high-pressure dense-phase CO₂ discharge.
- (2) Although both designs employ hydraulic drive to eliminate the lateral friction found in traditional mechanical pumps, the non-opposed configuration faces significant load fluctuations during the compression and suction phases. The hydraulic transmission system must frequently withstand unidirectional high-pressure shocks, which can easily induce low-frequency vibrations and accelerate fatigue wear of the sealing components. The optimized structure uses the opposed layout to balance alternating cylinder loads, while the force-transmission cylinders convert the horizontal hydraulic drive into vertical CO₂ compression and reduce lateral loading on the sealing interface. This structure helps reduce inertial shocks and improve the smoothness of linear piston motion, which is beneficial for reducing wear of the sealing components.
 - (3) In a non-opposed single-cylinder design, the inlet pressure potential energy associated with the 6–8 MPa high-base-pressure CO₂ stream cannot be effectively converted into compression assistance. During the compression stroke, the hydraulic main pump overcomes absolute exhaust back pressure. The opposed reciprocating pump utilizes a force-coupling mechanism to synchronize the suction process on one side with the compression process on the other. The inlet pressure of the high-base-pressure CO₂ stream is introduced into the coupled force balance and acts as an auxiliary thrust for the opposite compression stroke. This reduces the energy input required by the main pump and lowers the system's energy consumption.

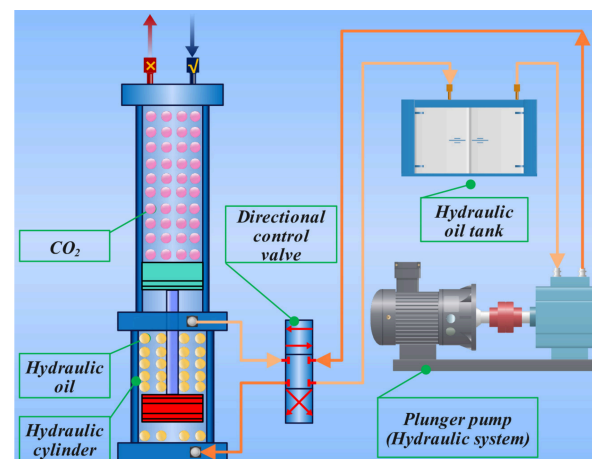


Figure 3. Schematic diagram of a non-opposed reciprocating pump.

3. Working Principle and Theoretical Analysis of the Injection Pump

3.1. Theoretical Mass Flow Rate Model

The discharge capacity of the injection pump is determined by the cylinder geometry and the piston reciprocation frequency. Dense-phase CO₂ in the pressure range of 6–25 MPa exhibits liquid-like high-density characteristics, while its compressibility remains sensitive to pressure and temperature variations. Therefore, to avoid numerical oscillations caused by nonlinear fluid properties and ensure good convergence of the model, this paper reasonably neglects the second-order nonlinear variation of density and treats the medium as a linearly compressible fluid based on the effective bulk modulus K_{eff} . Accordingly, the established theoretical mass flow rate model m_{th} still strictly complies with the basic principles of positive displacement pumps [31]:

$$m_{th} = \rho_{in} \cdot A_{gas} \cdot S \cdot f \cdot n \quad (1)$$

Considering the volumetric loss, the actual discharge flow rate m_{act} needs to be corrected by the volumetric efficiency η_v [31]:

$$\eta_v = 1 - \varepsilon \left(\frac{\rho_{out}}{\rho_{in}} - 1 \right) \quad (2)$$

ρ_{in} and ρ_{out} denote the suction and discharge thermodynamic-state densities of CO₂, respectively, which are determined by the corresponding pressure-temperature conditions. They are not instantaneous densities at a specific moment during the discharge process. The influence of residual CO₂ expansion and compression hysteresis in the clearance volume is accounted for by the relative clearance volume coefficient (ε).

For the ionic-liquid isolation structure proposed in this study, CO₂ remains the working medium in the volumetric-efficiency model; therefore, the form of Equation (2) remains applicable. The ionic liquid mainly modifies the effective gas-side clearance volume and leakage boundary, and ε is therefore interpreted as the effective clearance volume coefficient after considering the ionic-liquid isolation layer.

3.2. Energy Efficiency Evolution Mechanism and Energy Flow Model

The core of the proposed optimization lies in adapting the force-coupling mechanism of the opposed configuration to utilize the suction pressure potential of high-base-pressure dense-phase CO₂. Due to the rigidly coupled opposed configuration, the intake-side pressure acts as a load-compensation term in the force balance of the opposite compression cylinder. The instantaneous power $P_{oil}(t)$ required by the hydraulic system is given by:

$$P_{oil}(t) = \frac{[p_{out}(t) - p_{in}] \cdot A_{gas} \cdot v(t)}{\eta_h} \quad (3)$$

In Equation (3), η_h denotes the overall hydraulic transmission efficiency from the hydraulic pump outlet to the actuator. It accounts for the combined influence of pump mechanical loss, volumetric loss, valve throttling loss, and pipeline pressure loss.

The normalized specific energy consumption (e_s) of the hydraulic system is defined as the ratio of the steady-state peak hydraulic output power to the product of the inlet–outlet net pressure difference and the volumetric flow rate:

$$e_s = \frac{P_{oil}(t)}{Q_v \cdot (p_{out} - p_{in})} \quad (4)$$

When electrical energy is converted into the output power of a reciprocating pump drive system, the energy transmission efficiency of the motor and the oil pump must be taken into account. The actual active power input to the drive motor is determined by the hydraulic power and the mechanical/volumetric transmission efficiencies at each stage:

$$P_{in}(t) = \frac{P_{oil}(t)}{\eta_m \times \eta_p} \quad (5)$$

Equations (4) and (5) can be used to derive the normalized specific energy consumption (e_n) of a reciprocating pump system. This parameter quantifies the actual electrical energy required to compress 1 m³ of fluid and overcome a pressure difference of 1 MPa.

$$e_n = \frac{P_{in}(t)}{Q_v \cdot (p_{out} - p_{in})} \quad (6)$$

3.3. Dynamic Characteristic Analysis of the Piston Assembly

In this design, the piston assembly consists of a central hydraulic piston, a force-transmitting piston rod, and rodless pistons located in the cylinders on either side. To quantify the energy-saving mechanism of the optimized opposed-piston configuration under a 6–8 MPa high-base-pressure suction condition, this section derives the dynamic equilibrium equations.

The optimized opposed-cylinder configuration retains the inherent force-coupling feature of opposed pumps and adapts it to high-base-pressure dense-phase CO₂ injection. Because the suction cylinder and compression cylinder are coupled through the piston assembly and the force-transmission cylinders, the 6–8 MPa pressure on the suction side is introduced into the force balance of the opposite compression cylinder. The suction-side pressure acts as a load-compensation term rather than an independent energy source.

Therefore, the hydraulic pressure (p_{oil}) required by the actuator is governed by both the inlet-outlet pressure difference and the area ratio between the CO₂ cylinder and the hydraulic cylinder.

For the opposed configuration, when the piston assembly moves rightward, the force balance can be written as:

$$M\ddot{x} = p_{oil}A_{hyd} + p_{in}A_{gas} - p_{out}A_{gas} - F_f \quad (7)$$

where F_f is treated as the lumped friction resistance of the moving assembly, including the sliding friction in the hydraulic cylinder, the force-transmission cylinder, and the CO₂ cylinder sealing interface.

Thus, the hydraulic oil pressure required by the opposed configuration is obtained as:

$$p_{oil,op} = \frac{(p_{out} - p_{in})A_{gas} + F_f + M\ddot{x}}{A_{hyd}} \quad (8)$$

For a non-opposed hydraulic reciprocating pump with the same CO₂ cylinder and hydraulic cylinder dimensions, the suction-side compensation term does not exist. The required hydraulic oil pressure can therefore be expressed as:

$$p_{oil,no} = \frac{p_{out}A_{gas} + F_f + M\ddot{x}}{A_{hyd}} \quad (9)$$

Equations (8) and (9) indicate that the maximum required hydraulic pressure is not determined by the discharge pressure alone, but by the gas-side pressure load transmitted through the effective area ratio (A_{gas}/A_{hyd}). For the same CO₂ cylinder and hydraulic cylinder dimensions, the opposed configuration introduces the suction-pressure term ($p_{in} \cdot A_{gas}$) into the coupled force balance, whereas this term is absent in the non-opposed reference pump. Therefore, the suction pressure acts as a load-compensation term under a specified cylinder-area ratio, rather than as an independent energy source. As a result, the hydraulic load is transformed from a discharge-pressure-dominated load to an inlet-outlet net-pressure-difference-dominated load.

4. Results and Discussions

4.1. Simulation Model Establishment and Parameter Setting

In this study, a full-system model of the injection pump was constructed using the hydraulic and mechanical libraries within the AMESim 2020.1 simulation environment. Given that this study focuses on the force transmission mechanism of the opposed-cylinder configuration, the model employs an equivalent fluid modeling approach. In the simulation,

a hydraulic fluid with equivalent elastic modulus properties was used to replace the dense-phase CO₂ medium, with the bulk modulus set to match the average modulus of CO₂ within the operating temperature and pressure range. The core components of the model include a variable-displacement hydraulic pump, a three-position four-way electro-hydraulic directional control valve, a central double-acting hydraulic cylinder, a force-transmitting cylinder, and two opposing cylinders on either side. Each cylinder is equipped with an intake and exhaust check valve, and the opening and closing of these valves are controlled by pressure. The main simulation boundary conditions are set as shown in Table 2, and the simulation model is illustrated in Figure 4.

Table 2. Key parameters and boundary conditions for AMESim dynamic simulation.

Parameter	Value	Description
CO ₂ cylinder bore	250.00 mm	
CO ₂ cylinder stroke	500.00 mm	
Force cylinder bore	300.00 mm	
Force cylinder stroke	350.00 mm	
Hydraulic cylinder bore	300.00 mm	
Hydraulic cylinder stroke	350.00 mm	
Cylinder piston speed	0.102 m/s	Constant speed phase
Simulation step	10 ^{−2} s	Fixed time step
Oil bulk modulus	1700.00 MPa	
CO ₂ bulk modulus	120.00~300.00 MPa	
Inlet pressure	6.00~8.00 MPa	
Outlet pressure	25.00 MPa	

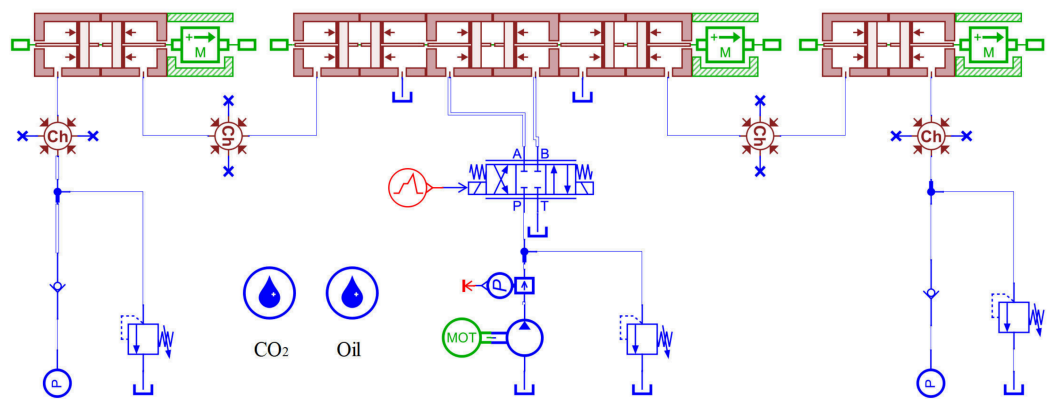


Figure 4. AMESim simulation model of the hydraulically driven opposed injection pump.

4.2. Dynamic Response and Pulsation Characteristics

4.2.1. Piston Motion and Operational Stability

In the reciprocating pump described in this paper, the constant-flow characteristics of the hydraulic system ensure that the cylinder piston moves steadily and at a uniform speed throughout the entire main compression stroke, resulting in a trapezoidal distribution of piston displacement. This long-stroke constant-speed characteristic effectively suppresses the violent fluid fluctuations caused by changes in the volume of the compressible medium.

Figure 5a,b illustrates the kinematic characteristics of a complete reciprocating cycle in an opposed-piston reciprocating pump. The piston’s uniform motion provides stable kinematic conditions for pressure transfer and force coupling during the reciprocating cycle. The piston’s uniform main power stroke lasts 4.8 s, and the directional control valve’s response time is 0.2 s. A slight displacement lag effectively prevents high-pressure hydraulic shock.

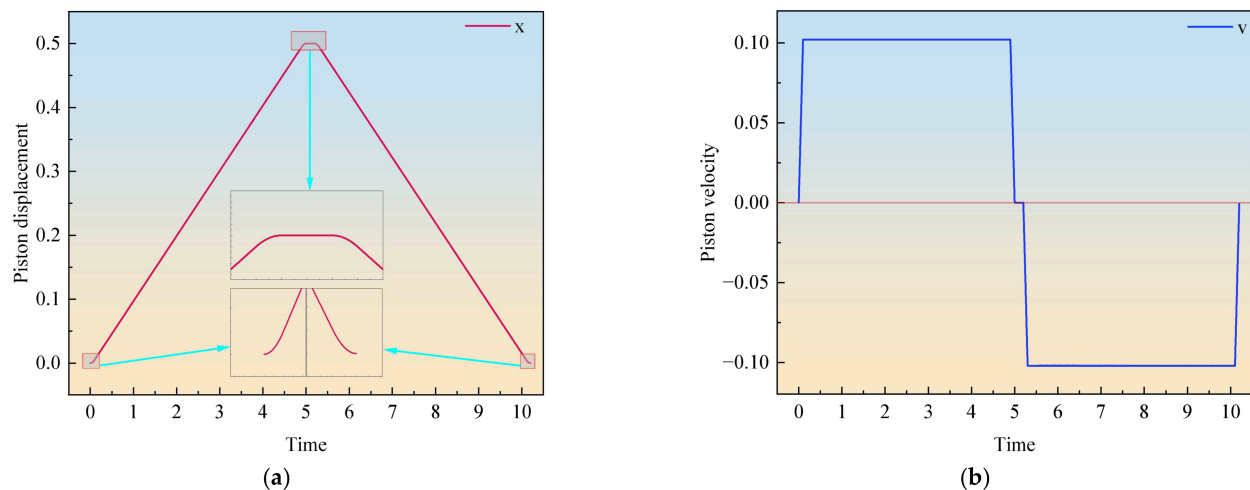


Figure 5. Cylinder piston motion curves: (a) Cylinder piston displacement curve. (b) Cylinder piston velocity curve.

4.2.2. Pressure and Discharge Flow Characteristics

To further characterize the complete-cycle dynamic response of the optimized opposed-piston pump, the time-domain pressure and discharge-flow responses under the rated 8 MPa inlet condition were compared with those of a non-opposed single-cylinder reference pump.

Figure 6 compares the cylinder pressure responses of the opposed-piston pump and the non-opposed single-cylinder reference pump under an inlet pressure of 8 MPa. In the opposed-piston pump, Cylinder B first undergoes compression and discharge. Its pressure rises from the suction-pressure level to approximately 25.11 MPa and remains at the discharge-pressure plateau from about 0.46 s to 5.20 s. During this period, Cylinder A is in the suction stroke, and its pressure is maintained near the inlet pressure of 8.03 MPa. After hydraulic reversal, Cylinder B completes the discharge stroke, enters the suction stroke, and maintains the inlet-pressure level of approximately 8 MPa. Meanwhile, Cylinder A is compressed to approximately 25.12 MPa and enters the subsequent discharge phase.

This alternating pressure phase reflects the force-coupling mechanism of the opposed-cylinder configuration. When Cylinder B performs compression and discharge, Cylinder A simultaneously executes the suction stroke. The high-base suction pressure acting on Cylinder A provides auxiliary thrust to the piston assembly, partially offsetting the discharge load of Cylinder B. After reversal, the working states of the two cylinders are exchanged, and the same force-coupling effect is produced in the opposite direction. Therefore, the gas load transmitted to the hydraulic actuator is mainly determined by the pressure difference between the discharge-side cylinder and the suction-side cylinder, rather than by the full discharge pressure. In contrast, the non-opposed single-cylinder reference pump exhibits an intermittent pressure response: the cylinder pressure rises to approximately 25.11 MPa during the compression/discharge stroke and then returns to the suction-pressure level during the return stroke. In this case, the 8 MPa inlet pressure acts only as the suction boundary and does not participate in load compensation.

Figure 7 shows the discharge flow-rate responses under the 8 MPa inlet condition. For the opposed pump, Cylinders A and B discharge alternately, with a stable instantaneous flow-rate plateau of approximately 286 L/min in each discharge stroke. Cylinder A discharges during the first half-cycle, while Cylinder B discharges after hydraulic reversal. Therefore, the total outlet flow of the opposed pump is obtained by superposing the discharge flow rates of the two cylinders at the same time instant.

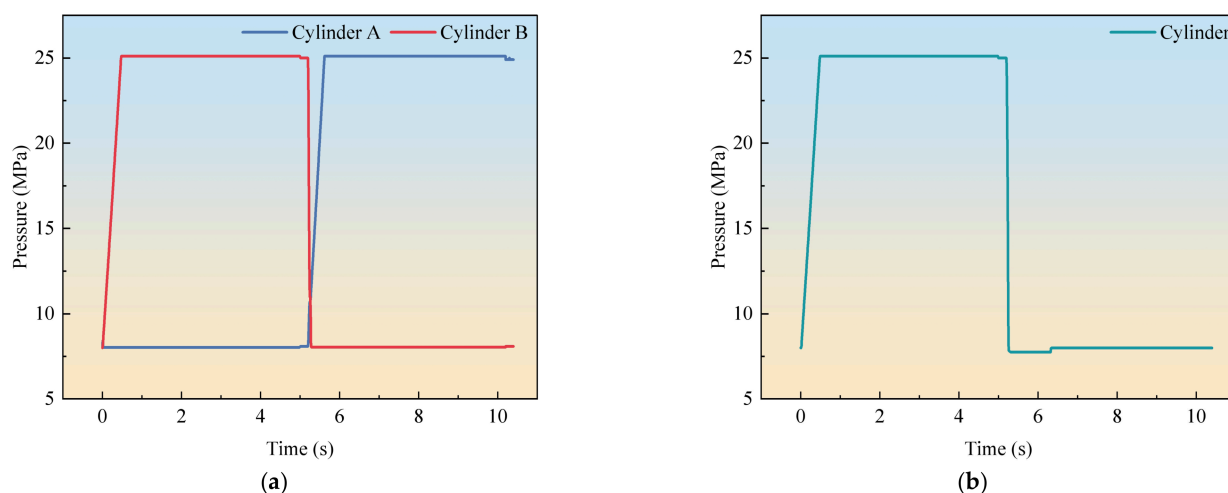


Figure 6. Comparison of CO₂ cylinder pressure responses under the 8 MPa inlet condition: (a) Opposed reciprocating pump. (b) Non-opposed reciprocating pump.

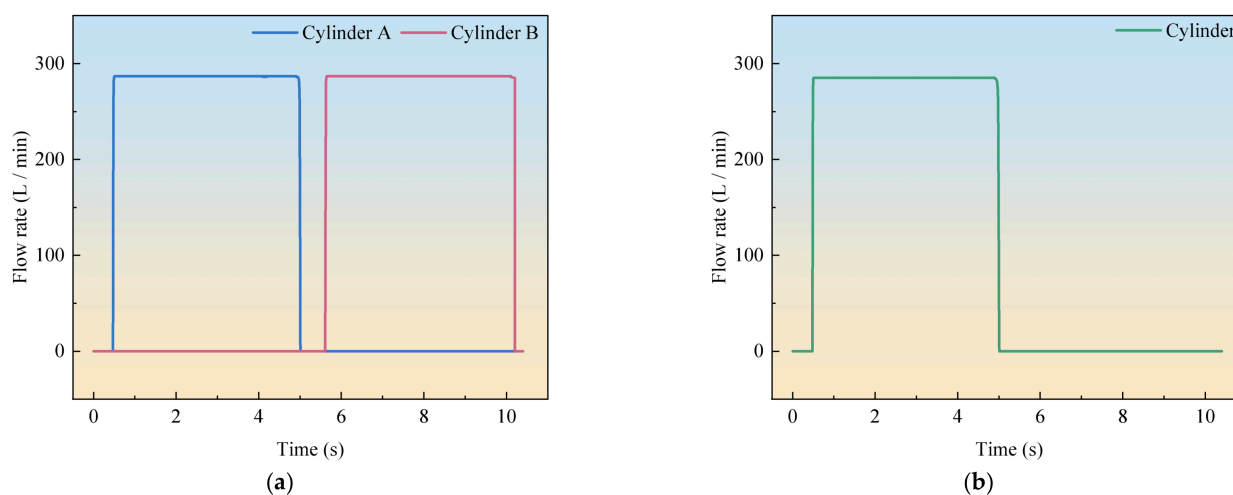


Figure 7. Comparison of discharge flow-rate responses under the 8 MPa inlet condition: (a) Opposed reciprocating pump; (b) Non-opposed reciprocating pump.

In contrast, the non-opposed single-cylinder reference pump only produces outlet flow during the discharge stroke. During the suction stroke, the discharge valve remains closed, and the outlet flow rate is zero because no discharge process occurs. Although the instantaneous discharge-flow plateaus of the two configurations are similar, the opposed pump provides two alternating discharge stages within one reciprocating cycle, thereby extending the effective discharge duration and shortening the zero-flow interval. Based on the complete-cycle superposition of the two cylinder discharge flows, the cycle-averaged outlet flow rate of the opposed pump is approximately 250 L/min, corresponding to 15 m³/h, which agrees well with the designed flow rate of 15 m³/h. This confirms that the phase-shifted discharge of the two cylinders weakens outlet-flow intermittency and improves discharge continuity.

4.3. Force-Coupling Mechanism and Hydraulic-Pressure Reduction

4.3.1. Pressure Response Under Different Inlet Pressures

To verify the force coupling characteristics of the opposed-cylinder configuration in this reciprocating pump, real-time response curves for the hydraulic drive pressure, intake and discharge pressures, and the intake-discharge pressure differential were extracted from the model over a single complete stroke.

Figure 8a–e illustrates the transient pressure evolution of a reciprocating pump operating under a constant discharge backpressure of 25 MPa as the inlet pressure increases from 6 MPa to 8 MPa. Within the steady-state discharge range, fluctuations in the actual hydraulic oil pressure on the drive side consistently track changes in the net pressure difference between the cylinder's intake and exhaust ports and remain below the discharge pressure at all times. This pressure inversion phenomenon confirms that the high potential energy of dense-phase CO₂ can be continuously converted into mechanical thrust and applied to the rear surface of the cylinder piston.

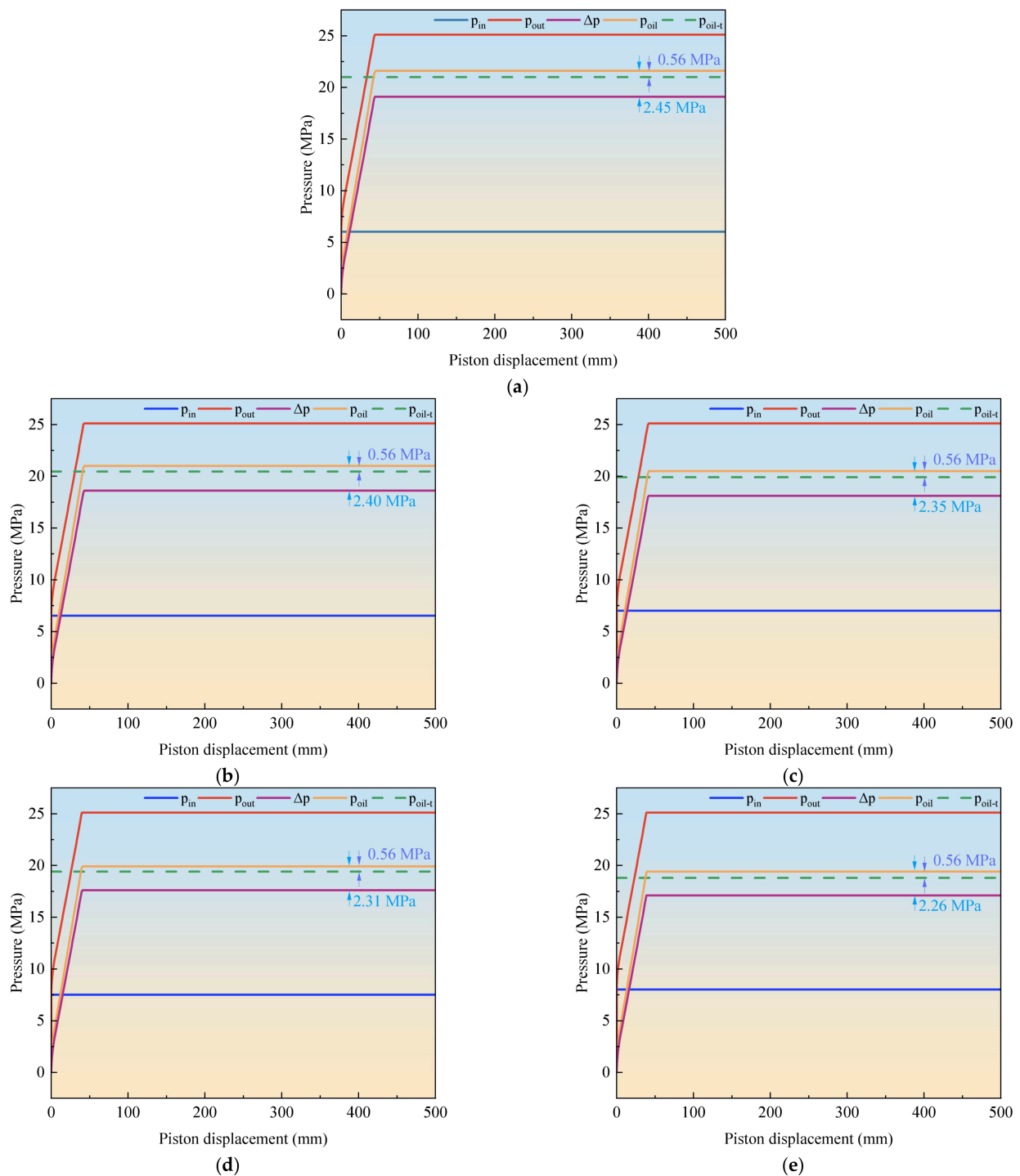


Figure 8. Comparison of cylinder pressures under different intake conditions: (a) 6.0 MPa inlet conditions; (b) 6.5 MPa inlet conditions; (c) 7.0 MPa inlet conditions; (d) 7.5 MPa inlet conditions; (e) 8.0 MPa inlet conditions.

When the inlet pressure is 6 MPa, 6.5 MPa, 7 MPa, 7.5 MPa, and 8 MPa, respectively, the peak operating pressure of the hydraulic pump required to maintain system exhaust is 21.56 MPa, 21.01 MPa, 20.46 MPa, 19.91 MPa, and 19.37 MPa, respectively. The peak hydraulic pressure of the drive system gradually decreases as the inlet CO₂ pressure increases.

During the discharge strokes under these five operating conditions, the actual hydraulic oil pressure curve consistently maintains a pressure difference of approximately 0.56 MPa relative to the theoretical curve. This discrepancy is primarily attributed to the system overcoming high-pressure friction in the main sealing assembly, pressure drop due to throttling in the directional control valve, and fluid transient compression hysteresis.

4.3.2. Comparison of Driving Pressures Between Opposed and Non-Opposed Configurations

This section further compares the hydraulic oil pressure response characteristics of non-opposed and opposed configurations under different supply pressures. Figure 9a,b illustrate the pressure variation patterns of the hydraulic cylinders for both configurations under a constant exhaust load of 25 MPa.

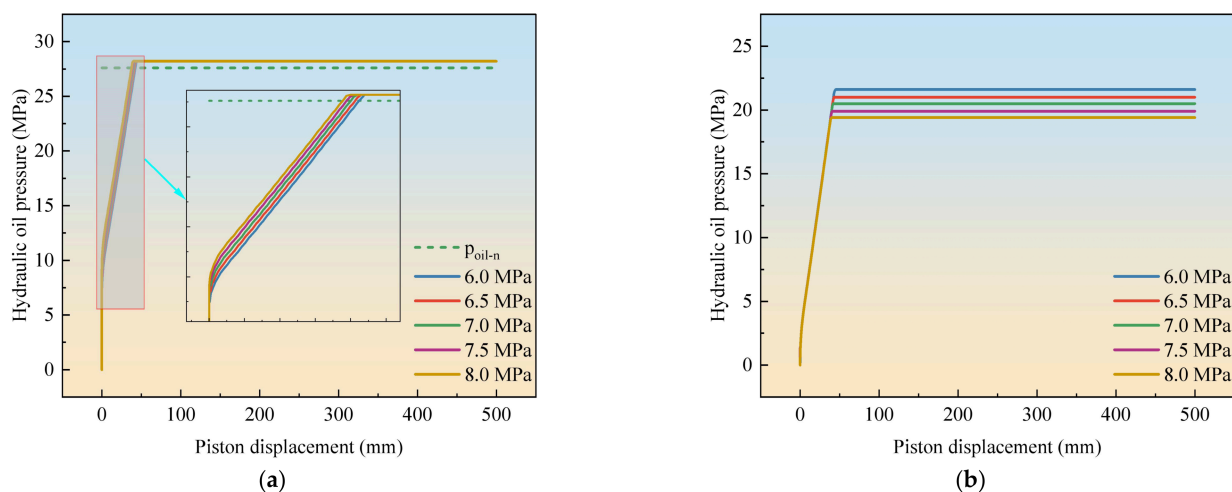


Figure 9. Comparison of hydraulic cylinder pressures under 6–8 MPa intake conditions: (a) Without an opposed structure. (b) With an opposed structure.

The simulation curve for the non-opposed structure shown in Figure 9a indicates that as the inlet CO₂ pressure increases from 6 MPa to 8 MPa, the peak pressure of the hydraulic system remains consistently around 28.23 MPa. However, as the inlet pressure rises, the initial displacement point at which the pressure begins to rise shifts, resulting in inconsistent growth curves during the compression phase, although the rate of increase remains the same.

In traditional non-opposed configurations, during the fluid displacement process, hydraulic oil from the rear of the piston flows to the low-pressure return line, and the main hydraulic pump must overcome an exhaust absolute pressure of 25 MPa as well as system friction. Therefore, regardless of how the inlet pressure changes, the load faced by the hydraulic pump remains constant at 25 MPa.

The trend of drive pressure in the opposed-cylinder configuration shown in Figure 9b differs from that of the non-opposed configuration. The pressure rise curves for the hydraulic oil remain consistent across different intake pressures, but the peak pressure achieved gradually decreases as the CO₂ source pressure increases. When the intake pressure is increased from 6 MPa to 8 MPa, the actual peak oil pressure decreases from 21.56 MPa to 19.37 MPa. This phenomenon reflects the inherent nature of direct force coupling in the opposed configuration. Through the hydraulic oil in the intermediate force-transmitting cylinder and the rigid connecting rods, the suction process of one cylinder is

synchronized with the compression process of the other, allowing the high-pressure fluid at the intake end to act directly on the back of the compression piston, converting it into forward auxiliary thrust.

This structural difference results in a shift in the driving load. While traditional designs are subjected to absolute loads, opposed-piston designs are subjected to net pressure differential loads. The opposed-piston design enables on-site recovery of potential energy; for every 0.5 MPa increase in intake pressure, the driving-end load decreases by approximately 0.55 MPa. Under the rated simulation condition of 8 MPa inlet pressure and 25 MPa discharge pressure, the opposed-piston configuration reduced the peak driving oil pressure by 31.39% compared with the non-opposed reference pump with the same CO₂ cylinder and hydraulic cylinder dimensions.

4.4. Comparative Analysis of Energy Efficiency Characteristics for Different Configurations

4.4.1. Output Power Characteristics of Non-Opposed Reciprocating Pump Drive Systems

Using AMESim's sensor module, the output power of the hydraulic pump in a reciprocating pump drive system was monitored to investigate the evolution of the drive system's transient output power under different inlet CO₂ pressures ranging from 6 to 8 MPa.

As shown in Figure 10, under different intake pressures, the output power curves of traditional non-opposed reciprocating pumps exhibit consistent peaks during the discharge stroke, but show differences in power growth during the compression stroke. Simulation data show that under intake conditions ranging from 6 MPa to 8 MPa, the peak output power during the exhaust phase is approximately 120.62 kW. Since the hydraulic system cannot obtain force assistance from a high-pressure CO₂ stream, the main hydraulic pump must continuously bear a constant back pressure of 25 MPa during the liquid displacement and exhaust phases.

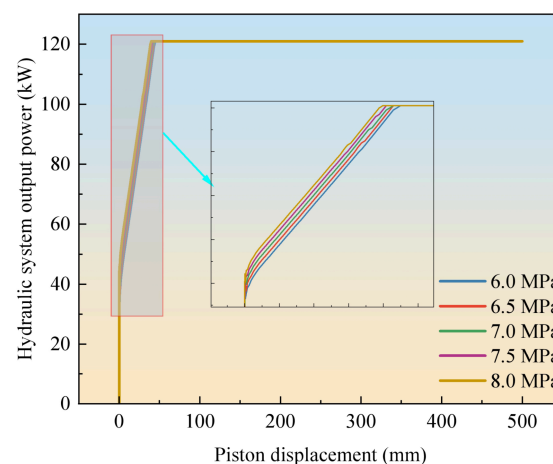


Figure 10. Output Power Curve of Hydraulic System for Non-opposed Reciprocating Pump (6~8 MPa).

During the compression stroke, as the intake base pressure increases from 6 MPa to 8 MPa, the system's pressure ratio decreases from 4.17 to 3.125. At higher initial intake pressures, the density of CO₂ inside the cylinder increases, causing the piston to encounter greater fluid resistance at the same displacement point. Differences in the compressibility characteristics of the fluids result in inconsistent power increase rates in the hydraulic system during the early compression phase; therefore, the higher the intake pressure, the more dramatic the increase in transient power.

The difference in hydraulic system output power between the 8 MPa and 6 MPa operating conditions reached a maximum of 10.62 kW at a piston displacement of approximately

0.47 mm. Furthermore, the non-opposed configuration causes the power load to increase sharply during the early compression phase, exacerbating the hydraulic transient impact.

4.4.2. Output Power Characteristics of the Drive System for an Opposed-Piston Reciprocating Pump

Figure 11 reveals obvious differences in the output power characteristics between non-opposed reciprocating pumps and opposed reciprocating pumps. During the compression stroke, the growth trajectories and rates of the power curves under different intake conditions are nearly identical. However, during the steady-state discharge phase, the peak output power gradually decreases as the intake pressure increases. Data shows that under operating conditions ranging from 6 MPa to 8 MPa, the maximum difference in transient power during the early compression phase is only 0.93 kW.

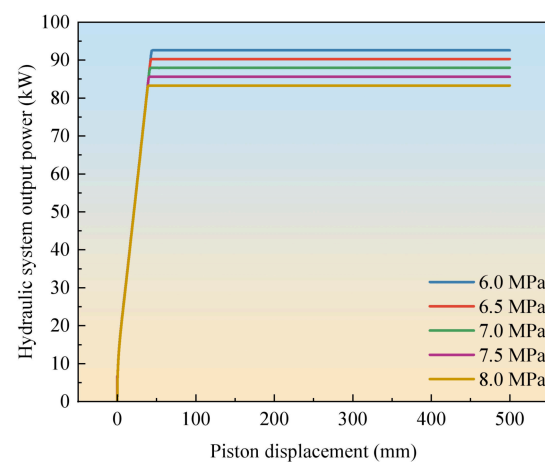


Figure 11. Output Power Curve of Hydraulic System for Opposed Reciprocating Pump (6–8 MPa).

However, when the system enters the 25 MPa steady-state discharge phase, as the inlet pressure increases from 6 MPa to 8 MPa, the peak output power of the hydraulic main pump decreases from 92.59 kW to 83.31 kW. For every 0.5 MPa increase in inlet pressure, the system's steady-state peak power decreases by approximately 2.32 kW.

This power output behavior is mainly caused by the suction-pressure-assisted load compensation of the opposed-piston configuration. At the beginning of each stroke, the same suction-pressure level exists on the intake side. Therefore, the initial hydrostatic load is largely balanced in the coupled opposed-piston assembly. As the discharge-side pressure increases during compression, the suction-side pressure continues to act as a load-compensation term. Consequently, the hydraulic power demand is mainly governed by the inlet–outlet net pressure difference rather than the full discharge pressure.

As the compression process progresses, the pressure inside the cylinder gradually rises, and the compensatory power required from the hydraulic side varies with the net pressure difference, ultimately resulting in a monotonically decreasing trend in steady-state peak power.

4.4.3. Relationship Between Net Pressure Difference and Hydraulic Output Power Under Different Inlet Pressures

Figure 12 compares the inlet–outlet net pressure difference and the steady-state peak hydraulic power of the opposed and non-opposed pumps under different inlet pressures. As the inlet pressure increases from 6 MPa to 8 MPa, the net pressure difference decreases from approximately 19.11 MPa to 17.11 MPa. Correspondingly, the peak hydraulic power of the opposed pump decreases from approximately 92.59 kW to 83.31 kW, indicating that the increase in suction pressure effectively reduces the hydraulic load required for CO₂ pressurization.

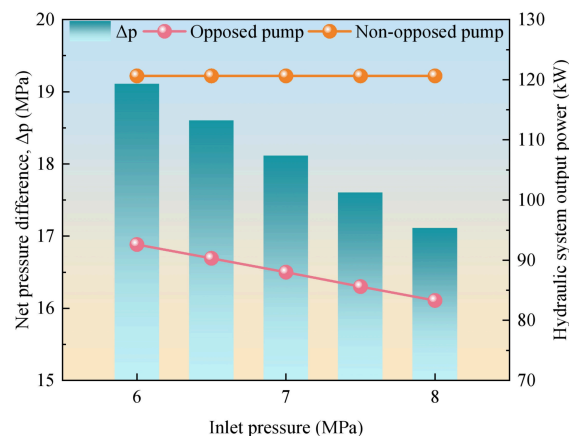


Figure 12. Variations in net pressure difference and hydraulic system output power under different inlet pressures.

In contrast, the non-opposed pump maintains an almost constant peak hydraulic power of approximately 120.62 kW over the same inlet-pressure range. This indicates that the inlet pressure in the non-opposed configuration acts only as a suction boundary and does not contribute to compression-load compensation. Compared with the non-opposed pump, the opposed pump reduces the peak hydraulic power by approximately 28.03 kW at 6 MPa and 37.31 kW at 8 MPa, corresponding to reductions of about 23.2% and 30.9%, respectively. Therefore, the energy-saving advantage of the opposed configuration becomes more pronounced as the inlet pressure increases.

5. Field Testing and Performance Comparison

5.1. Test System Setup

To verify the practical operating performance and model prediction accuracy of the optimized opposed-piston reciprocating pump, a field reliability operation test was carried out under actual dense-phase CO_2 injection conditions. Considering the need for a direct comparison with the simulation results, a continuous 24 h stable operating dataset was selected for detailed analysis. The selected dataset was used to evaluate the agreement between theoretical prediction, AMESim simulation, and field measurement, including hydraulic oil pressure, effective flow rate, and energy-consumption fluctuation.

The test was performed at an actual CCUS injection site, with the discharge pressure based on the on-site wellhead pressure. CO_2 from the pipeline entered the injection pump via a front-end throttling and separation skid, with an inlet pressure range of 6–8 MPa. The test system is shown in Figure 13, and the field test setup is shown in Figure 14.

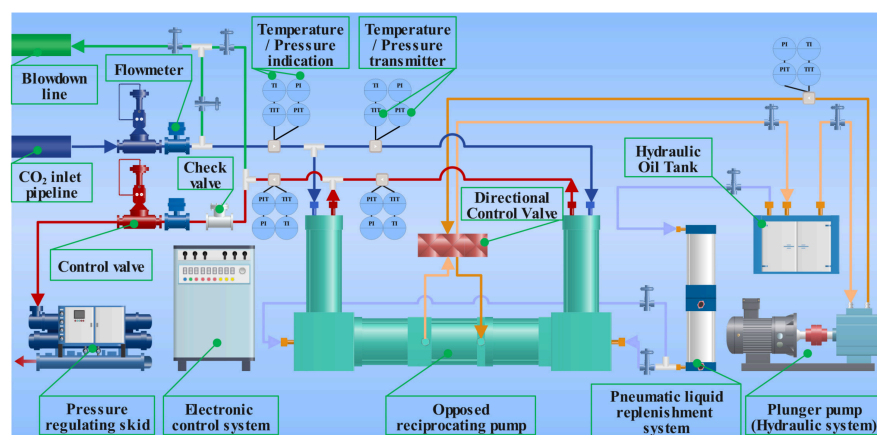


Figure 13. Schematic diagram of a field test of an opposed-piston reciprocating pump.

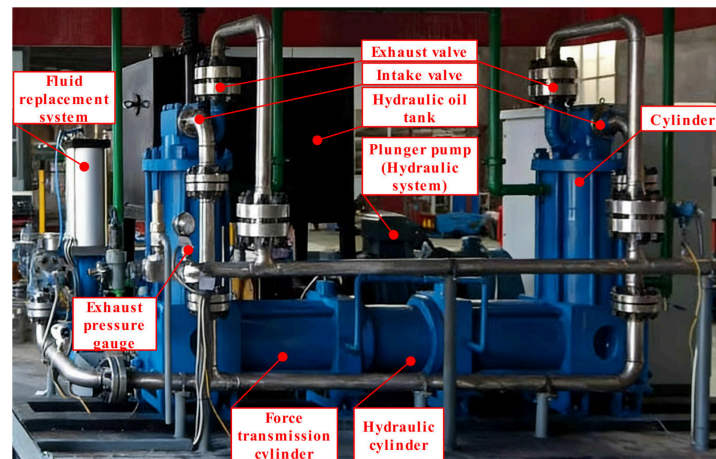


Figure 14. On-site test setup for an opposed-piston reciprocating pump.

To ensure the reliability of the field test data, the test system performed high-precision monitoring of key thermodynamic parameters. The main technical parameters of the field test are shown in Table 3.

Table 3. Table of key technical parameters for field tests.

Category	Parameter	Value	Description
Fluid & Thermodynamic	Rated inlet pressure	6.00~8.00 MPa	
	Rated discharge pressure	20.00~25.00 MPa	
	Ambient temperature	28.00 °C	
	Design Flow Rate	15.00 m ³ /h	
	Working Fluid	Dense-phase CO ₂	
Hydraulic & Control	Max hydraulic pressure	31.5 MPa	
	Directional valve response time	0.2 s	
	Motor Rated Power	110 kW	
Measurement & Sensors	High-frequency pressure transmitters	0~40 MPa	±0.1% FS
	RTD temperature transmitters	−20~80 °C	Class A accuracy
	Shock-proof pressure gauges	0~40 MPa	Class 1.6
	Bimetallic thermometers	−20~80 °C	Class 1.5
	Coriolis mass flowmeter	--	±0.2%

5.2. Field Reliability Operation and Simulation Validation

5.2.1. Comparison of Hydraulic Pressure Responses Between Simulation and Field Operation

To validate the dynamic model under practical operating conditions, the hydraulic pressure response obtained from the selected 24 h field reliability dataset was compared with the theoretical and AMESim simulation results. Due to limitations imposed by the on-site CO₂ feed conditions and the actual backpressure at the injection wellhead, the inlet pressure during the selected test period fluctuated mainly between 6.18 and 6.56 MPa, while the discharge pressure remained within 21.3–21.5 MPa.

Figure 15a,b show the simulated and measured pressure curves for the inlet, discharge, and hydraulic oil pressures of the optimized opposed-piston pump system, respectively. A comparison of the simulated data and field measurements for the actual drive pressure within the hydraulic cylinder reveals that both sets of data exhibit a pressure inversion phenomenon. During the full-load fluid displacement stroke, the actual hydraulic oil pressure remained below the absolute exhaust load of 21.4 MPa, fluctuating above the net pressure difference curve between intake and exhaust, which consistently maintained a value of approximately 15 MPa.

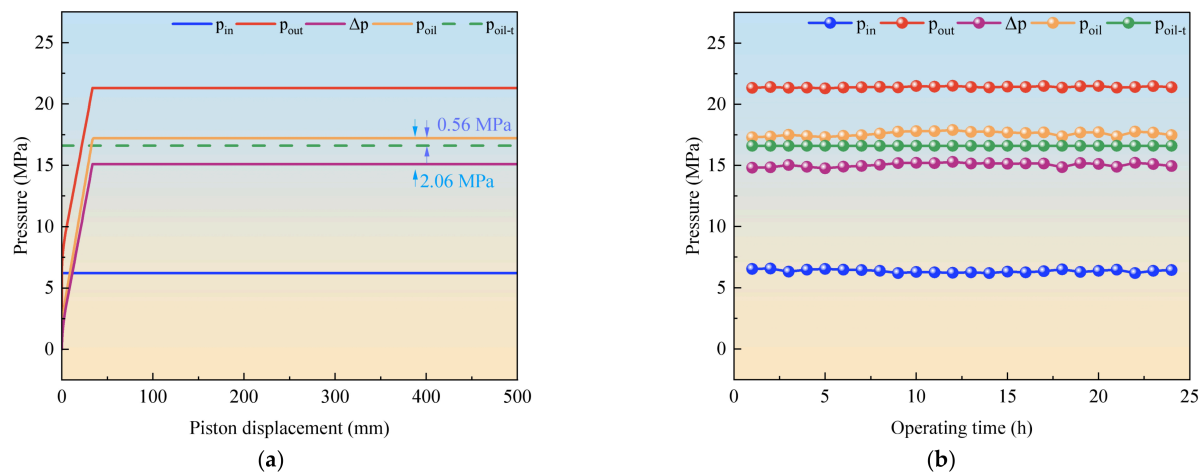


Figure 15. Pressure responses under field conditions: (a) Simulated pressure responses. (b) Measured pressure responses.

This phenomenon demonstrates that, during continuous and fluctuating actual engineering operations, the potential energy of the high-pressure gas source is transferred through the opposing mechanism, continuously providing forward thrust to the rear of the compression piston, thereby reducing the system's working load from the absolute discharge pressure to the net pressure difference.

Table 4 compares the theoretical, simulated, and experimental data for the oil pressure of this opposed-piston reciprocating pump. Based on Pascal's principle, the theoretical reference oil pressure (p_{oil-t}) required to maintain the system's static equilibrium under these operating conditions is 16.60 MPa. After incorporating friction damping for each piston in the AMESim simulation, the model output a steady-state peak oil pressure of 17.16 MPa; meanwhile, the actual operating peak oil pressure captured by the on-site high-pressure sensor fluctuated slightly between 17.29 MPa and 17.84 MPa. The data indicates that the relative error between the theoretical value, the simulated prediction, and the on-site measured value of the hydraulic oil pressure is within 4%.

Table 4. Relative errors in hydraulic oil pressure between theoretical, simulated, and experimental values.

Time (h)	Theoretical Value (MPa)	Simulation Value (MPa)	Relative Error (%)	Experimental Value (MPa)	Relative Error (%)
1	16.60	17.17	3.40	17.29	0.75
2	16.60	17.17	3.40	17.36	1.12
3	16.60	17.17	3.40	17.50	1.97
4	16.60	17.17	3.40	17.40	1.38
5	16.60	17.17	3.40	17.32	0.88
6	16.60	17.17	3.40	17.42	1.46
7	16.60	17.17	3.40	17.47	1.77
8	16.60	17.17	3.40	17.60	2.53
9	16.60	17.17	3.40	17.76	3.44
10	16.60	17.17	3.40	17.80	3.67
11	16.60	17.17	3.40	17.79	3.66
12	16.60	17.17	3.40	17.84	3.92
13	16.60	17.17	3.40	17.73	3.32
14	16.60	17.17	3.40	17.77	3.55
15	16.60	17.17	3.40	17.69	3.05
16	16.60	17.17	3.40	17.64	2.77
17	16.60	17.17	3.40	17.69	3.08
18	16.60	17.17	3.40	17.38	1.22
19	16.60	17.17	3.40	17.69	3.06
20	16.60	17.17	3.40	17.71	3.18
21	16.60	17.17	3.40	17.37	1.20
22	16.60	17.17	3.40	17.76	3.49
23	16.60	17.17	3.40	17.68	2.99
24	16.60	17.17	3.40	17.48	1.81

The simulated values were approximately 0.56 MPa higher than the theoretical values, primarily due to volumetric hysteresis caused by fluid compressibility, frictional resistance along the pipeline, and high-pressure dry friction at the main seal. The on-site measured values were 1% to 3.96% higher than the simulated values. This is primarily due to unavoidable interference factors at the project site, such as localized friction spikes caused by assembly tolerances, microscopic cavitation distortions during high-frequency fluid direction changes, and the measurement tolerance of the load cell itself.

It should be noted that the 31.39% hydraulic-pressure reduction was obtained from the rated 8 MPa/25 MPa comparative simulation rather than directly from the field test. Due to the limitations of the on-site CO₂ supply pressure and injection-well backpressure, the field test was conducted under an inlet pressure of 6.18–6.56 MPa and a discharge pressure of 21.3–21.5 MPa. Therefore, the field test was used to validate the force-balance-based hydraulic pressure prediction under practical operating conditions. The agreement among the theoretical, simulated, and measured hydraulic pressures within 4% confirms the applicability of the force-balance model for actual dense-phase CO₂ injection.

5.2.2. Analysis of Normalized Specific Energy Consumption During Field Operation

Based on the selected 24 h continuous operating data, the rated flow rate of the opposed-piston reciprocating pump was 15 m³/h, while the measured effective flow rate remained approximately 14.7 m³/h. The corresponding volumetric efficiency was about 98%, indicating that the pump maintained stable displacement output during field reliability operation.

Figure 16 shows the variation in the normalized specific energy consumption of the field-test system during the selected 24 h continuous operation. The average normalized specific energy consumption was 0.422 kWh/(MPa·m³). During most of the test period, the energy-consumption level remained relatively stable, indicating that the opposed-piston pump maintained steady energy-conversion performance under actual dense-phase CO₂ injection conditions.

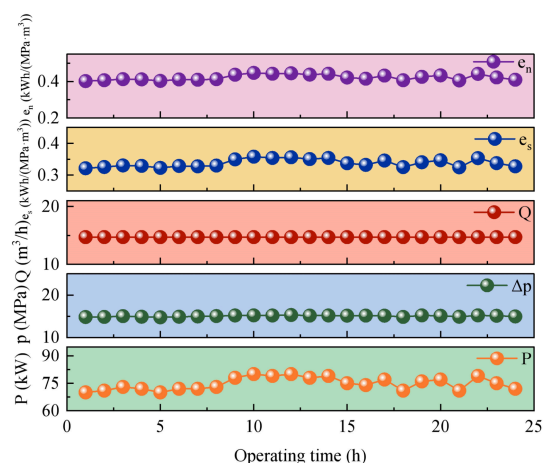


Figure 16. Normalized specific energy consumption of the field test system.

A moderate increase in normalized specific energy consumption was observed during approximately 10:00–14:00. This period corresponded to the strongest solar radiation and the highest environmental heat load at the test site. The increase in ambient temperature affected the thermophysical properties of the dense-phase CO₂ stream. As the feed-fluid temperature increased, the CO₂ density decreased and its compressibility increased, which increased the elastic compression work required for pressurization. In addition, the reciprocating pump is a positive-displacement machine, and the delivered volume per stroke

is essentially fixed. Therefore, when the CO₂ density decreases, a higher reciprocating frequency is required to maintain the rated mass delivery. This increases the average hydraulic power of the drive system and consequently leads to a temporary rise in normalized specific energy consumption.

After this high-temperature period, the normalized specific energy consumption gradually returned to a lower level, demonstrating that the system retained good adaptability to field operating-condition fluctuations.

Although the selected field dataset covers 24 h rather than a full lifetime test, it provides a preliminary assessment of the short-term operational stability of the optimized opposed-piston pump under practical dense-phase CO₂ injection conditions. During the selected test period, no continuous upward drift in hydraulic oil pressure, no obvious decrease in effective flow rate, and no progressive increase in normalized specific energy consumption were observed. These results indicate that the hydraulic drive system maintained stable pressure regulation during continuous operation.

In addition, the rodless piston, vertical CO₂ cylinder, and ionic-liquid isolation layer reduce direct contact between dense-phase CO₂ and the mechanical sealing interface, which is beneficial for suppressing seal wear and leakage. The observed energy-consumption fluctuation was mainly related to external operating-condition variations, such as ambient heat load and CO₂ property variation, rather than progressive deterioration of the pump itself.

6. Conclusions

To reduce the high energy consumption of large-scale dense-phase CO₂ pressurized injection in CCUS applications, this study developed a structural optimization scheme based on a conventional hydraulically driven opposed-piston reciprocating pump under high-base-pressure operating conditions. By establishing a dynamic model, conducting AMESim simulations, and performing a field reliability operation test, this study clarified the suction-pressure-assisted load-compensation mechanism and evaluated the operating performance of the optimized pump under both rated simulation and practical field conditions. The main conclusions are as follows:

- (1) A dense-phase CO₂-oriented structural optimization was developed on the basis of a conventional hydraulically driven opposed-piston reciprocating pump. By introducing the 6–8 MPa suction pressure into the force balance of the opposite compression cylinder, the hydraulic load was transformed from a discharge-pressure-dominated load to an inlet–outlet net-pressure-difference-dominated load.
- (2) Under the rated simulation condition of 8 MPa inlet pressure and 25 MPa discharge pressure, the optimized opposed configuration reduced the peak driving oil pressure by 31.39% compared with the non-opposed reference configuration. The driving load decreased by approximately 0.55 MPa for every 0.5 MPa increase in inlet pressure, confirming the suction-pressure-assisted load-compensation effect.
- (3) The introduced force-transmission cylinder enabled vertical CO₂-cylinder arrangement and converted horizontal hydraulic actuation into vertical CO₂ compression, reducing lateral loading on the sealing interface. The rodless piston and ionic-liquid isolation layer improved sealing and lubrication under low-viscosity dense-phase CO₂ conditions, contributing to an effective flow rate of approximately 14.7 m³/h and a volumetric efficiency of about 98% in the selected 24 h field dataset.
- (4) The selected 24 h field reliability dataset verified the applicability of the theoretical and AMESim models under practical operating conditions. The measured hydraulic oil pressure agreed with the predicted values within 4%, and the average normalized specific energy consumption was 0.422 kWh/(MPa·m³). The optimized pump showed stable short-term field performance and adaptability to operating-condition fluctua-

tions. The suction-pressure-assisted load-compensation principle may also provide a reference for other high-base-pressure fluid pressurization systems.

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Nomenclature

ρ_{in}	CO ₂ density at the suction thermodynamic state	ρ_{out}	CO ₂ density at the discharge thermodynamic state
A_{gas}	Cylinder piston area	A_{hyd}	Annular area of the hydraulic cylinder piston
$v(t)$	Piston velocity	p_{in}	Inlet pressure
$p_{out}(t)$	Instantaneous discharge pressure	Δp	Inlet–outlet net pressure difference
m_{th}	Theoretical mass flow rate	m_{act}	Actual mass flow rate
M	Equivalent mass of the moving piston assembly	$\ddot{x}$	Acceleration of the piston assembly
x	Displacement of the piston assembly	F_f	Lumped friction resistance of the piston assembly and sealing interfaces
p_{oil-t}	Theoretical hydraulic oil pressure	p_{oil}	Hydraulic oil pressure
$P_{in}(t)$	Instantaneous power of the reciprocating pump system	$P_{oil}(t)$	Instantaneous power of the hydraulic system
ε	Relative clearance volume coefficient	Q_v	Volumetric flow rate
S	Cylinder piston effective stroke	f	Reciprocation frequency
η_v	Volumetric efficiency	n	Work times per cycle
e_s	The hydraulic system normalized specific energy consumption.	η_h	Overall hydraulic transmission efficiency from the hydraulic pump to the actuator
η_m	Motor efficiency	η_p	Hydraulic pump efficiency
K_{eff}	Effective bulk modulus of dense-phase CO ₂	e_n	System normalized specific energy consumption.

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