Research Article

Optimization Design of Actuator Parameters in Multistage Reciprocating Compressor Stepless Capacity Control System Based on NSGA-II

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The capacity control system of reciprocating compressor has great significance for the contribution of energy conservation and emission reduction. The parameters of the actuator and hydraulic system within a reciprocating compressor stepless capacity control system play a decisive role in its control accuracy, mechanical reliability, and mechanical security. The actuators and hydraulic system parameters of the same stage are in conflict with each other. Therefore, the actuator and the multistage reciprocating compressor are studied here, specifically through multiobjective optimization using the Nondominated Sorting Genetic Algorithm (NSGA)-II. The multiobjective optimization design was performed on a two-dimensional (2D) reciprocating compressor test bench. When the spring stiffness of the first stage spring was 27358 N m⁻¹, the spring stiffness of the second stage spring was 23315 N m⁻¹, the inlet oil pressure was 296.62 N, the impact velocity of ejection was 0.4215 m s⁻¹, and the total indicated power deviation was 12.05 kW; the objective functions were optimized. Compared with traditional parameters, the inlet oil pressure, spring stiffness, and impact velocity were all reduced. This parameter optimization design lays the foundations for global optimization designs for stepless capacity control systems.

1. Introduction

Reciprocating compressors are widely used in the petroleum, chemical, chemical fertilizer, natural gas, transportation, and other industrial fields. Regarding the design margins of reciprocating compressor capacities, especially the actual production demand change, which comprises the production process demands of a compressor’s capacity change or intake quantity change, reciprocating compressors should have favourable function regarding their capacity regulation [1]. In recent years, more and more compressors have been equipped with capacity control systems, which are regulated by valves [2]. Through the investigation, a petrochemical refinery has a 2900 kW reciprocating compressor with a capacity control system. Running at a 40–60% load, the power saving enabled by the system is up to 1200 kW per hour. This can save the refinery more than 5 million yuan per year of operation. The energy-saving effect is, therefore, obvious.

In order to facilitate their application in the petrochemical industry, many of the problems associated with these control systems, such as valve plate fractures, fatigue of vulnerable components, reliability, safety, and cost, represent major challenges that need to be solved urgently. Many studies on optimization design regarding reciprocating compressors have focused on wearing parts, pipe vibration,
noise, and cranks. A capacity-regulation system for a reciprocating refrigeration compressor, based on a novel rotary control valve, was proposed and designed by Li et al. [3]. The dynamic performance research of reciprocating compressor values has also been studied [4]. Chang et al. adopted COMSOL multiphysics (COMSOL), a finite element analysis software, to facilitate the shape optimization of the muffler; a polynomial neural network model was adopted to serve as an objective function. Additionally, a genetic algorithm (GA) is linked to the objective (OB) function to realize noise reduction [5]. Wang et al. treat the valve as a whole and optimize it globally, based on the axiomatic design (AD) theory, to reduce the coupling of the design parameters and simplify the design process [6]. Qin et al. proposed a new uncertain optimization algorithm, using NSGA-II, to suppress the vibration of the crankshaft system [7]. Ferreira et al. reported a method developed to optimize the suction system of a reciprocating compressor, also using the genetic algorithm NSGA-II [8]. Designers have been working hard to improve the work efficiency of reciprocating compressors and to optimize the design of their key components, to ensure that reciprocating compressors can operate at their optimal capacity.

Considering the problems related to their actual application, many studies have analysed the optimization of stepless capacity control system parameters, to lay the foundations for the promotion of the capacity control system. Hong et al. [9] and Song et al. [10] establish a theoretical model of the compressor’s thermodynamics cycle, in the case of stepless capacity regulation. The influence of the regulating system on compressor performance has also been analysed. Yu-hui et al. analysed the influence of suction valve motion on the hydraulic pressing-off force and the positive effect of the valve plate’s life on the valve speed damper [11]. Mu-Lin et al. researched the influence of the suction valve plate impact load on the hydraulic pressing-off force and on predicting the fatigue life. This provides the theoretical basis for the design and improvement of the hydraulic system and the optimization of the stepless capacity control system mechanism [12]. Ilai et al. studied the flow field characteristics, the structural improvement of the hydraulic distributor, and dynamic performance of the hydraulic actuator [13]. Yin determined the numerical relationship between the regulation ratio and the pressing-off force [14]. Through these above analyses, it is apparent that the theoretical model of a capacity control system can be established. The influence of the capacity control system’s parameters on the reciprocating compressor is therefore analysed here, and a simple optimal design is conducted. However, the mutual inhibitions and the contradictory relationships among the multiple parameters and objectives, such as the hydraulic pressure, spring stiffness, impact velocity of ejection, and regulating effect, have not yet been considered. Moreover, the optimization of the design of the capacity control system for a multistage reciprocating compressor has yet to be considered. For example, for parameters at the same stage, when the hydraulic pressure and spring stiffness are at their minimum, the system safety and cost can be reduced, but the impact speed of the ejection will not necessarily be at its minimum, and thus the regulating effect may be poor. Regarding the parameters between the stages, the inlet oil pressure is suitable for the first stage but will lead to failure during the high-pressure stage, regarding pushing the valve plate and adjusting the exhaust volume. Therefore, the singular objective optimization of hydraulic pressure or structural improvement cannot guarantee that the other parameters will be their nondominant set at the same time. Obtaining the optimal solutions of these problems and ensuring high efficiency and safe operation for stepless capacity control systems are therefore extremely critical for these multiple conflicting objectives.

Multiobjective optimization methods include traditional multiobjective methods and multiobjective optimization evolutionary algorithms. The traditional methods include the weighted method, the constraints method, and the goal programming method, each of which has several shortcomings in dealing with complex problems, such as high-dimensional, multimode, and nonlinear problems. Evolutionary algorithms can mainly be divided into three categories: aggregate function algorithms, swarm-based methods, and pareto. Aggregate function algorithms transform a multiobjective problem into a single-objective problem by weighting; it is difficult to find nonconvex solutions for linear functions. The swarm-based method completes searches of each divisional target through the evolution of the group. The disadvantage of this method is that the relationship between each target is not considered in the selection process, and the solution obtained is therefore not the overall optimal solution. The core of method based on pareto is to assign different dominance classes to the population. Typical representative algorithms include MOGA [15], NSGA [16], NSGA-II [17], SPEA [18], and NPGA [19]. NSGA-II is a typical representative of the pareto-based method; it overcomes the defects of complex calculation, low efficiency, and low robustness of NSGA and has the advantages of exhibiting good distribution and good convergence.

In this article, the key parameters of a reciprocating compressor stepless capacity control system are treated as being mutually suppressed and contradictory, and as such, traditional methods cannot obtain the optimal solutions of multiple parameters. Here, NSGA-II is used to solve this optimization design problem, to ensure that a stepless capacity control system can operate safely and efficiently. Section 2 introduces the stepless capacity control system for a reciprocating compressor. In Section 3, the actuators and multistage compressors are studied to establish the mathematical models of actuators and valves. Section 4 explores the relationships between hydraulic pressure, spring stiffness, impact velocity, and indicated power. These relationships are obtained by analysing the interaction relationships between multiple parameters, based on experimental data. Section 5 examines the indicated power deviation, impact velocity of ejection, hydraulic pressure, and spring stiffness as objective functions. This establishes a multiobjective optimization mathematical model, providing the optimal parameters for the actuator and hydraulic system, through the NSGA-II method. Section 6 presents the main conclusions.
2. Introduction to Stepless Capacity Control System

Hydraulic or pneumatic actuators are used to drive the unloader so that the suction valve is forced open by the unloader during the compression stroke. The gas then flows back through the suction valve to the suction inlet, realizing the function of adjusting the compressor displacement. The regulation method can be divided into two approaches: full stroke pressure-open suction valve regulation and partial stroke pressure-open suction valve regulation. Full stroke pressure-open suction valve regulation, which can only steplessly control the opening time and closing of the suction valve, compression process of the unloading device, that is, controlling the opening time and closing of the suction valve, partial stroke pressure-open suction valve regulation can theoretically achieve 0–100% stepless capacity control.

The working principle of the stepless capacity control system is shown in Figure 1. The hydraulic system mainly consists of a hydraulic oil unit, which provides the driving force for the actuator. When the hydraulic system provides high pressure, the driving force overcomes the spring force of the unloader, the friction of the actuator, etc. The unloader then opens the suction valve and is at its lower limit. When the hydraulic system provides low pressure, the spring force of the unloader overcomes the hydraulic pressure, friction, etc.; the loader retreats to the top limit, and the suction valve closes.

3. Mathematical Model

3.1. Mathematical Model of Actuator. The valve plate is opened in the process of compression by the actuator by the reciprocating compressor capacity control system. It uses oil pressure to achieve the purpose of the capacity control. The actuator is subjected to hydraulic pressure, gravity, gas force, spring force, and friction force, among others. The force and motion diagram of the actuator is shown in Figure 2. At the initial moment (state II), the actuator fails to overcome the spring force due to the small hydraulic pressure provided by the hydraulic system, so the actuator is at its upper limit. When the hydraulic system provides a higher pressure, the spring force of the unloader overcomes the hydraulic pressure, friction, etc.; the actuator mass, \( x(i) \) represents the actuator displacement, \( F_i \) represents the gas force of suction, \( F_{i}(i) = p_i(\ i) A_{\text{unloader}} \), \( a \) represents the installation angle of the actuator, \( f \) represents the total friction of the actuator, \( F_{i}(i) \) represents the spring force of actuator, \( F_i(i) = k_{\text{unloader}} (i) \ (x_0 + x) \), \( F_{cy}(i) \) represents the gas force of the cylinder, and \( y \) represents the gas force coefficient of the valve plate. When the executing fork contacts the valve plate, \( y = 1 \); otherwise, \( y = 0 \).

The initial conditions of formula (1) are as follows:

\[
\begin{align*}
\dot{x'}(0) &= 0 \\
x(0) &= x_0 \quad \theta_1 \leq \theta \leq \theta_2, \\
\dot{x'}(0) &= 0 \\
x(0) &= x_0 + L \quad \theta_3 \leq \theta \leq \theta_4,
\end{align*}
\]

where \( x'(0) \) represents the initial velocity in ejection or withdrawal of actuator, \( x(0) \) is the initial displacement in ejection or withdrawal of actuator, and \( L \) represents the actuator trip.

3.2. Mathematical Model of Compressor with Stepless Capacity Control System. To account for the actuator’s changing opening and closing states under capacity control conditions, backflow processes are added, compared with the compressor model under normal working conditions. Furthermore, the suction valve may be withdrawn with the actuator if the withdrawal speed is slow. As shown in Figure 3, the suction valve is forced open during the compression stroke (the crank angle is \( \theta_3 \sim \theta_4 \)). The movement state of the valve plate is thus changed. In the closing process of the suction valve, if the acceleration of the unloader is lower than the suction valve plate acceleration (i.e., the crank angle is \( \theta_5 \sim \theta_4 \)), the closing speed of the valve plate then becomes slow. The detailed description of unloader and suction valve movement process is shown in Table 1.

Based on the above analysis, here a compressor mathematical model is established, based on a capacity control system. Before establishing the mathematical model, the following hypotheses are proposed:

1. The suction valve is an automatic valve, which is not affected by the actuator during the opening process.
2. The motions of the exhaust valve and suction valve plates are one-dimensional.
3. The flow of gas through the valve gap is a one-dimensional flow of ideal gas and is an adiabatic process.
The cylinder transfers heat with the cooling water in an outer wall, which is simulated as an interwall heat exchanger. Its heat transfer coefficient is \( B \text{ (J·m}^{-2}·\text{s}^{-1}) \).

Under the normal condition, the differential equations of different processes of the compressor have been summarized in [20]. Under the capacity regulation condition, the backflow process is added and the dynamic pressure equation of the gas in the exhaust process is improved to obtain the dynamic pressure equation of the gas in the backflow process. The actuator model in Section 3.1 is integrated into the backflow process pressure equation. The differential equation is shown in Table 2.

3.3. Test Parameters. In this article, theoretical and experimental studies are carried out with a 2D reciprocating compressor stepless capacity control experiment bench. The actuator model and the compressor model are established under the stepless capacity control system. The actuator parameters and compressor parameters are shown in Tables 3 and 4, respectively.
4. Research on the Influence of Key Parameters with Capacity Control System

For the capacity control system of the multistage reciprocating compressor, it is necessary to consider the influence of the actuator and the hydraulic system parameters both at the same stage and at different stages. Taking these parameters to be at the same stage as the research object, in order to reduce the cost and system safety, a design to reduce the hydraulic pressure can be adopted. In order to avoid the mechanical fatigue and noise caused by the excessive impact of ejection, the spring stiffness should be reduced. However, the decrease in spring stiffness will cause the valve plate to withdraw slowly. The gas backflow then increases, the adjustment deviation increases, the control precision decreases, and the overall effect is undesirable. Taking the parameters at different stages as the research object and considering the cost and design problems, all stages should adopt the same oil pressure. If the oil pressure at the high-pressure stage is taken as the overall design value, then the initial conditions are substituted into the ejection motion differential equation of the actuator and the \( x_{(i)} = L \). The following equation can then be obtained:

\[
\dot{y}(i) = -\sqrt{\frac{k_{\text{unloader}}(i)}{m(i)}} \left( x_0 - \frac{F_h - F_i(i) - f + m(i)g \cos \alpha}{k_{\text{unloader}}(i)} \right)
\sqrt{1 - \frac{L - (F_h - F_i(i) - f + m(i)g \cos \alpha/k_{\text{unloader}}(i))}{x_0 - (F_h - F_i(i) - f + m(i)g \cos \alpha/k_{\text{unloader}}(i))}^2}.
\]

(3)

The results of the first level actuator are shown in Figure 5. As can be seen from the figure, the impact velocity decreases with increasing spring stiffness. This decreasing trend shows a linear change, and the velocity changes within a range of 0.3–0.6 m/s. The impact velocity increases with increasing oil inlet pressure, the increase velocity decreases with increasing oil pressure, and the velocity changes within a range of 0.3–0.9 m/s.

4.1. Research on the Relationship between Spring Stiffness, Inlet Oil Pressure, and Impact Velocity. Based on the mathematical model of the actuator under capacity control conditions, the relationship between spring stiffness, inlet oil pressure, and impact velocity is studied here. When the angle is \( \theta_1 \leq \theta \leq \theta_2 \) and the actuator is in the ejection process, then the initial conditions are substituted into the ejection motion differential equation of the actuator and the \( x_{(i)} = L \). The following equation can then be obtained:

\[
\dot{y}(i) = -\sqrt{\frac{k_{\text{unloader}}(i)}{m(i)}} \left( x_0 - \frac{F_h - F_i(i) - f + m(i)g \cos \alpha}{k_{\text{unloader}}(i)} \right)
\sqrt{1 - \frac{L - (F_h - F_i(i) - f + m(i)g \cos \alpha/k_{\text{unloader}}(i))}{x_0 - (F_h - F_i(i) - f + m(i)g \cos \alpha/k_{\text{unloader}}(i))}^2}.
\]

(3)
Table 2

| Process | Differential equation |
|---------|-----------------------|
| Backflow | $\theta_3 - \theta_3$: the plate remains stationary |

$$\left\{
\begin{align*}
\frac{dh}{d\theta} &= 0, \\
\frac{dv}{d\theta} &= 0, \\
\frac{d\varphi}{d\theta} &= -\left[C(1 - \cos \omega t + (\lambda/2)\sin^2 \omega t)(k - 1/V_c y)p_c y + \varphi(k dV_c y/V_c y d\theta) - (k/\omega)(p_c y / p_i)^{(1/\lambda)}\right] \alpha_{sv} A_{sv} \left[\frac{2k}{k - 1} RT_s [(p_c y / p_i)^{(1 - 1/\lambda)} - 1]\right]. \\
\end{align*}
\right.$$  

$\theta_3 - \theta_4$, $v_a > v_p$

$$\left\{
\begin{align*}
\frac{dh}{d\theta} &= 0, \\
\frac{dv}{d\theta} &= 0, \\
\frac{d\varphi}{d\theta} &= -\left[C(1 - \cos \omega t + (\lambda/2)\sin^2 \omega t)(k - 1/V_c y)p_c y + \varphi(k dV_c y/V_c y d\theta) - (k/\omega)(p_c y / p_i)^{(1/\lambda)}\right] \alpha_{sv} A_{sv} \left[\frac{2k}{k - 1} RT_s [(p_c y / p_i)^{(1 - 1/\lambda)} - 1]\right]. \\
\end{align*}
\right.$$  

$\theta_3 - \theta_4$, $v_a < v_p$

$$\left\{
\begin{align*}
\frac{dh}{d\theta} &= v, \\
\frac{dv}{d\theta} &= (1/\omega)(k_{unloader} \cdot x + p_a - f i m), \\
\frac{d\varphi}{d\theta} &= -\left[C(1 - \cos \omega t + (\lambda/2)\sin^2 \omega t)(k - 1/V_c y)p_c y + \varphi(k dV_c y/V_c y d\theta) - (k/\omega)(p_c y / p_i)^{(1/\lambda)}\right] \alpha_{sv} A_{sv} \left[\frac{2k}{k - 1} RT_s [(p_c y / p_i)^{(1 - 1/\lambda)} - 1]\right]. \\
\end{align*}
\right.$$  

Note: $h$ represents the valve plate displacement, $\theta$ represents the crank angle, $\alpha_{sv} A_{sv}$ represents the instantaneous effective valve gap area of suction valve, $K$ represents the spring stiffness of valve plate, $k$ represents the ratio of specific heat of gas, $V_c y$ represents the cylinder volume, $H_0$ represents the precompression of valve plate spring, $M_i$ represents the valve quality, $R$ represents the gas constant, $T_s$ represents the suction temperature, $\beta$ represents the coefficient of applied force of gas, $Z$ represents the number of the spring, $P_j$ represents the inlet pressure, $P_c y$ represents the cylinder pressure, and $\alpha$ represents the ratio between the crankshaft radius and connecting rod length.
4.2. Research on the Relationship between Spring Stiffness and Load Deviation of Capacity Control System. The relationship between the spring stiffness of the unloader and the regulation load was studied based on the mathematical model of the compressor and the actuator. The spring stiffness of the unloader needs to meet the following constraints:

$$m(i)g \cos \alpha + f - F_{cy}(i) \leq \frac{k_{unloader}(i)}{x_0}$$

$$\leq \frac{F_h - F_i(i) + m(i)g \cos \alpha + f}{x_1 + L}.$$
where equation (4) is the constraint condition that the actuator can withdraw to the upper limit and maintain balance and can push out to the lower limit and maintain balance. According to the parameters in Table 3 and equation (4), the spring stiffness of the unloader should be satisfied under the following constraints:

\[ k_{\text{unloader}} (1) \geq 22724 \text{N/m} \] and \[ k_{\text{unloader}} (2) \geq 21028 \text{N/m} \].

The spring stiffness affects the withdrawal time of the actuator, as shown in Figure 6. The relationship between the reset spring stiffness and the withdrawal time is similar to the inverse function. When the reset spring stiffness is greater than or equal to 75000 N/m, the slope of the curve declines greatly, and the withdrawal time tends towards the lower limit. Increasing the spring stiffness further will lead to a sharp increase in the withdrawal shock, as well as an increase in the inlet oil pressure. This would reduce the working performance and safety of the capacity control system.

Under normal conditions, the suction valve plate retracts automatically. Through the numerical calculations, the withdrawal time of the suction valve plate is determined to be 1.56 ms. The ultimate withdrawal time of the unloader is 4 ms with the capacity control system. As the mass of the valve plate is small relative to the unloader, the acceleration of the valve plate is large relative to the unloader. Therefore, under the capacity control conditions, the valve plate is withdrawn together with the unloading device.

Based on the above analysis, the relationship between the reset spring stiffness and the load deviation, the difference between the indicated power deviation under different loads, and different reset spring stiffnesses under capacity control conditions and indicated power under normal conditions were studied. Three load conditions of 40%, 60%, and 80% were taken, and the first stage spring stiffness was 23000–73000 N/m, the second stage spring stiffness was 22000–72000 N/m, and an interval of 5000 N/m was used for the simulation calculations.

Figures 7–9 show the following:

(1) Figure 7 shows the displacement diagram of the valve plate, under different spring stiffnesses under a 40% load condition. With increasing reset spring stiffness, the withdrawal action time of the suction valve plate decreases, but the decrease in amplitude becomes smaller and smaller. This indicates that, with increasing spring stiffness, the withdrawal time of the actuator will tend towards its lower limit, and the reduction range of the withdrawal time will become smaller and smaller.

(2) Figure 8 shows the cylinder pressure curve for different spring stiffnesses, under a 40% load condition. As the spring stiffness increases, the withdrawal time of the suction valve decreases. This means that the backflow process in the cylinder pressure curve also decreases, and that the area enclosed by the cylinder pressure curve increases.

(3) Figures 9 shows the indicated power differences for different spring stiffnesses, under different load conditions, by calculating the indicated power difference with ideal values. The indicated power difference of different spring stiffnesses with the maximum spring stiffness is calculated, and the indicated power difference of the different spring stiffnesses under different load conditions is then obtained. With the same spring stiffness and different load conditions, the indicated power difference is basically the same, indicating that the load has little influence on the gas-indicated power difference, and can therefore be ignored. For the same load but different spring stiffnesses, the difference in the gas indication tolerance decreases as the spring stiffness increases, that is, \[ \Delta \eta (i) = \eta (i) - \eta_k (i) = f (i) (k) \].

Based on inverse proportion function properties, the
The relationship between $\Delta \eta$ and $k$ is shown in the following equations:

$$\Delta \eta(1) = \frac{132100k(1) + 2.219}{k(1)^2 + 6871k(1) + 4.772}$$  \hspace{1cm} (5)$$

$$\Delta \eta(2) = \frac{240300k(2) + 3.733}{k(2)^2 + 6018k(2) + 14.55}$$  \hspace{1cm} (6)$$

5. Multiobjective Optimization Mathematical Model of Actuator Based on NSGA-II

NSGA-II is based on fast nondominant sorting and crowding distance. It is a multiobjective optimization genetic algorithm. It is built on the basis of NSGA, but elite strategy and fast nondominant ranking strategy are added to greatly improve the shortcomings of NSGA. In this article, a 2D reciprocating compressor test bench is taken as the optimization object. The multiobjective optimization research of load deviation, the impact velocity of ejection, the hydraulic pressure, and the spring stiffness are all carried out.

The design of reducing the system hydraulic pressure and the spring stiffness of unloader can reduce the costs of the actuator and hydraulic system and increase the safety coefficient of the system. If the load deviation is reduced, the control precision will be higher. Reducing the impact velocity can reduce the noise and increase the life of the actuator. The multiobjective mathematical models of the unloader are shown in the following equations:

$$\min f(X) = \begin{bmatrix} k_{unloader}(1) + k_{unloader}(2) \\ P_1 \\ \Delta \eta_1 + \Delta \eta_2 \\ v(i) \end{bmatrix},$$ \hspace{1cm} (7)$$

$$\text{sub} \begin{cases} F_i \geq F_L(i) + f + F_r(x_0 + L)(i), \\
k_{unloader}(i)x_0 \geq f + m(i)g \cos \alpha. \end{cases}$$ \hspace{1cm} (8)$$

Figure 7: 40% load, the first stage valve displacement of different spring stiffness.

Figure 8: 40% load, the first stage indicated power of different spring stiffness.
In this paper, firstly, the objective equation is obtained through the mathematical model of the unloader and the compressor and then calculated with the NSGA-II algorithm. The algorithm flow is shown in Figure 10.

5.1. Calculation Results and Experimental Analysis. The setting parameters of the nondominant sorting multi-objective optimization are shown in Table 5. Figure 11 shows the feasible solutions among the four targets.
When the multiobjective optimization method is applied to practical problems, in order to apply the optimization results to the actual parameter settings, it is necessary to select several results from Pareto optimal solutions as the final parameters. In this article, the fuzzy analytic hierarchy process (FAHP) is used to select three sets of Pareto optimal solutions as the final parameters. Although the deviation from the indicated power affects the adjustment accuracy, this can be compensated by the control method. Therefore, the ratios between the weight of the inlet oil pressure, the spring stiffness, the impact speed, and the deviation from indicated power are 0.3:0.3:0.3:0.1. According to the weight coefficient, the optimal solution can be obtained. Take the first three groups of solutions, the three points A, B, and C in Figure 11. And it can be seen that the three points converge in the red region, and the values of the three sets of parameters are shown in Table 6.

Comparing the optimized parameters determined using NSGA-II with traditional design parameters, the oil pressure and stiffness are greatly reduced, as shown in Table 7. In order to verify the actual effects, after the optimization of NSGA-II, the experiment was carried out on a 2D reciprocating compressor test bench, and a vibration sensor was used to test the impact of ejection, as shown in Figures 12 and 13, and Table 8 reveals that the impact of ejection was reduced by 146.92%, indicating that the optimized parameters can greatly reduce the impact speed of ejection. The reliability and security of the system are therefore improved. As the spring stiffness decreases significantly following optimization, the impact during withdrawal is also reduced.

### Table 5: Parameter of NSGA-II.

| Parameter       | Value |
|-----------------|-------|
| Population size | 100   |
| Maximum generation | 1000 |
| Mutation fraction | 0.7  |
| Crossover fraction | 0.4  |
| Variation ratio | 0.02  |
| Crossover ratio | 0.02  |

### Table 6: Optimal solution of actuator parameters.

| Index | First/second spring stiffness (N/m) | Hydraulic force (N) | Impact velocity (m/s) | Indicated power difference (kW) |
|-------|-------------------------------------|----------------------|-----------------------|---------------------------------|
| 1     | 27358/23315                         | 296.62               | 0.4215                | 12.05                           |
| 2     | 33280/40494                         | 358.26               | 0.5227                | 9.05                            |
| 3     | 24689/20274                         | 351.24               | 0.6799                | 6.14                            |

### Table 7: Experiment parameter.

| NSGA-II                | Ref bin-bin, B | Drop rate  |
|------------------------|----------------|------------|
| Spring stiffness (N/m) | 27358/23315   | 100000     | 294.7%     |
| Hydraulic force (N)    | 296.62         | 904        | 204.8%     |

![Figure 11: \( k - p_1 - v - \Delta \eta \) pareto front.](image1)

![Figure 12: Experiment table.](image2)

![Figure 13: Vibration of NSGA-II result and traditional result.](image3)
Although the withdrawal phase of the unloader is delayed by 5.5%, the unloader can be closed in advance by adjusting the parameters of the control valve to reduce the influence of spring stiffness on the compressor.

6. Conclusion

In this article, the key parameters of the actuator and the hydraulic system in a multistage reciprocating compressor stepless capacity control system are mutually suppressed and contradictory. The parameters of the actuator and the hydraulic system at the same stage can be in conflict with each other. It is very important to match the same oil pressure parameters with different compressors and actuator parameters. Parameters determined by traditional methods cannot make the system run efficiently. The NSGA-II multiobjective optimization method presented here to solve the optimization design problem allows the stepless capacity control system to operate safely and efficiently, however:

(1) First, the moving parts of the actuator were treated as the research object, and the mathematical model for the conditions of the capacity control system was constructed by comprehensively considering the functions of hydraulic pressure friction gas force and spring force. By combining the actuating mechanism motion equation (under the conditions of the capacity control system) with the compressor model (under the conditions of having no capacity control system), the compressor model under the condition of capacity control can be constructed to lay a theoretical foundation for the analysis of the relationships between the key parameters.

(2) The influences of spring stiffness and hydraulic pressure on the impact velocity and indicated power were analysed by combining the mathematical model and the experimental table parameters. The impact velocity was found to decrease with increasing spring stiffness and increase with the increase in inlet oil pressure. Analysing the simulation results for the indicated power of the compressor, under different spring stiffnesses and under three different loads, shows that, under the same spring stiffness, the load of the compressor has little influence on the indicated power deviation. Furthermore, under the same load, the spring stiffness is inversely proportional to the deviation from the compressor-indicated power.

(3) Taking the spring stiffness, fluid pressure, impact velocity of ejection, and deviation from indicated power as objective functions, a multiobjective mathematical model was constructed, which was solved using NSGA-II. Three groups of solutions were selected by weight as the optimal solutions, which showed a reduction of 204.8% compared with traditional parameters [21]. The experimental results show that, after the optimization of spring stiffness and hydraulic pressure, the impacts of ejection and the impact of withdrawal are greatly reduced, allowing the stepless capacity control system to operate more safely and more efficiently.

(4) The results of multiobjective optimization in this paper are greatly improved compared with the results of traditional single-objective optimization, but the optimization work can be further studied. For example, by improving the mathematical model of the compressor or applying fluid simulation software, the influence of actuator parameters on the reciprocating compressor temperature can be studied.

### Nomenclature

| Symbol | Description |
|--------|-------------|
| Fs | Actuator speed, m/s |
| Fi | Gas force coefficient of valve plate, N |
| θi | End angle of ejection, rad |
| θf | Normal condition full closing angle of suction valve plate, rad |
| θc | Start angle of ejection, rad |
| θu | Start angle of withdrawal, rad |
| θv | Normal condition initial closing angle of suction valve plate, rad |
| θs | End angle of withdrawal, rad |
| θ | Normal condition initial closing angle of suction valve plate, rad |

### Table 8: Comparison of parameters.

| Items                        | Before optimization | After optimization | Drop rate |
|------------------------------|---------------------|---------------------|-----------|
| Vibration of ejection (m/s²) | 321                 | 130                 | 146.92%   |
| Vibration of withdrawal (m/s²)| 138                 | 92                  | 50.00%    |
| Closed angle (°)             | 253                 | 267                 | −5.24%    |

### Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.
Conflicts of Interest

The authors declare that they have no conflicts of interest.

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