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RESEARCH ON INTEGRATED OPTIMIZATION METHOD FOR POWER LOSS AND ENERGY EFFICIENCY OF HYDRAULIC SYSTEMS

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ДОСЛІДЖЕННЯ КОМПЛЕКСНОГО МЕТОДУ ОПТИМІЗАЦІЇ ВТРАТ ПОТУЖНОСТІ ТА ЕНЕРГОЕФЕКТИВНОСТІ ГІДРАВЛІЧНИХ СИСТЕМ

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Abstract. *Hydraulic systems are widely used in construction machinery and industrial manufacturing due to their high power density and rapid response, but high power loss and low energy efficiency hinder their green low-carbon development. To solve this problem, this paper proposes an integrated optimization method for power loss and energy efficiency of hydraulic systems. Firstly, three core loss sources (throttling, volumetric, mechanical) are analyzed, and a full-dimensional loss model considering their coupling relationships is established. Secondly, a multi-objective optimization model targeting maximum energy efficiency is constructed, with constraints including system dynamic performance (actuator speed, step response time), component operating parameters (rated pressure / flow of pumps / valves, cylinder thrust / stroke), and hydraulic oil characteristics (10–60 °C temperature, viscosity). An improved particle swarm optimization algorithm, featuring adaptive inertia weight, dynamic learning factors, and random boundary resetting, is adopted for solution. Finally, simulations via AMESim (variable pump-multi-way valve-double-acting hydraulic cylinder model) and experiments are conducted on an excavator boom hydraulic system. Results show the method increases comprehensive energy efficiency by 15.3 % and reduces total power loss by 21.7 %, balancing energy saving and operational stability, and providing theoretical and technical support for efficient hydraulic system operation.*

Keywords: hydraulic system, power loss, energy efficiency optimization, integrated method, improved particle swarm optimization algorithm.

Анотація. *Гідравлічні системи широко застосовують у будівельній техніці, спеціальних транспортних засобах і промисловому обладнанні виробництв завдяки високій щільності потужності та швидкій реакції, але великі втрати потужності та низька енергетична ефективність обмежують їхній зелений низьковуглецевий розвиток. Щоб розв'язати цю проблему, у статті запропоновано інтегрований метод оптимізації втрат потужності та енергетичної ефективності гідравлічних систем будівельних машин. По-перше, проаналізовано три головні джерела втрат (лінійні та місцеві, об'ємні, механічні), а також побудовано повномірну модель втрат, що враховує взаємозв'язок між ними. По-друге, у роботі створена багатометрична модель оптимізації за критерієм досягнення максимуму енергетичної ефективності гідравлічного привода з обмеженнями, зокрема динамічною характеристикою системи (швидкість виконавчої частини, час стрибкового відгуку), робочими параметрами компонентів (номінальний тиск / подача насосів, пропускна спроможність і витоки клапанів, тяга / хід гідравлічного циліндра) і характеристиками*

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гідравлічної рідини (температура 10–60 °C, в'язкість номінальна). Для розв'язання моделі застосовано покращений алгоритм оптимізації роя частинок з адаптивною інерційною вагою, динамічними навчальними факторами і стратегією випадкового скидання меж. Проведено симуляції в середовищі AMESim (модель «змінний насос – багатоходовий клапан – гідравлічний циліндр двосторонньої дії»). Модель верифікована експериментально на прикладі гідравлічної системи стріли однокінвшевого екскаватора. Результати демонструють, що запропонований метод оптимізації втрат підвищує загальну енергетичну ефективність на 15,3 % і зменшує загальні втрати потужності на 21,7 %, забезпечуючи баланс між енергозбереженням і робочою стабільністю гідравлічної системи, а також надає теоретичну і технічну підтримку для подальшого розвитку та ефективного функціонування гідравлічних систем будівельних машин і спеціальних транспортних засобів.

Ключові слова: *гідравлічна система, втрати потужності, оптимізація енергетичної ефективності, інтегрований метод, покращений алгоритм оптимізації роя частинок.*

1. Introduction

1.1. Research Background and Significance

Hydraulic transmission technology occupies a core position in heavy machinery and industrial equipment such as excavators, cranes, CNC machine tools and metallurgical equipment due to its high power density, strong load-bearing capacity and stable transmission characteristics [1]. According to statistics, the energy consumption of hydraulic systems in the global industrial field accounts for 18~25 % of the total industrial energy consumption, while the energy efficiency of hydraulic systems in the construction machinery field is generally only 40~60 %, and a large amount of energy is lost through power loss in the form of heat energy [2]. With the intensification of the global energy crisis and the popularization of the concept of green and low-carbon development, reducing power loss and improving energy efficiency of hydraulic systems have become one of the core directions of hydraulic technology development.

The power loss of hydraulic systems is widely distributed, involving multiple core components such as pumps, valves, cylinders and motors, and there are coupling relationships between various losses. Traditional optimization methods mostly focus on local optimization of a single component or a single type of loss, such as optimizing the throttling structure of relief valves to reduce throttling loss and improving the sealing performance of hydraulic pumps to reduce volumetric loss [3]. Although such methods can reduce local losses to a certain extent, they are likely to lead to an increase in other losses, making it difficult to achieve the optimal improvement of the overall energy efficiency of the system. Therefore, carrying out research on the integrated optimization of power loss and energy efficiency of hydraulic systems, considering the coupling relationships between various components and loss types, and realizing global optimization at the system level, is of great theoretical significance and engineering application value for promoting the green upgrading of hydraulic systems and reducing industrial energy consumption [9].

1.2. Research Status at Home and Abroad

Foreign research on energy efficiency optimization of hydraulic systems started early, and a series of theoretical and technical achievements have been formed. In terms of power loss modeling, German scholar Manring proposed a calculation model for volumetric loss and mechanical loss of hydraulic pumps, and verified the accuracy of the model through experiments [4]; the NASA laboratory in the United States established a throttling loss model considering temperature effects for aviation hydraulic systems, providing a basis for energy efficiency optimization of high-altitude hydraulic systems [5]. In terms of optimization methods, Japanese scholars used genetic algorithms to optimize the parameters of valve-controlled circuits in hydraulic systems, reducing throttling losses [6]; Swedish scholars proposed an energy efficiency optimization strategy for hydraulic systems based on model predictive control, realizing real-time energy efficiency optimization under dynamic working conditions [7].

Domestic scholars have also carried out a lot of research in the field of energy efficiency optimization of hydraulic systems. The team from Harbin Institute of Technology established a power loss model of the excavator hydraulic system and improved energy efficiency by optimizing the matching relationship between the main pump displacement and valve opening [8]; researchers from Zhejiang University used particle swarm optimization algorithms to optimize the circuit parameters of hydraulic systems and verified the effectiveness of the algorithm in energy efficiency optimization [9]. However, there are still deficiencies in existing research: first, most power loss models ignore the coupling effects between various losses, resulting in insufficient modeling accuracy; second, the optimization objectives are mostly focused on reducing losses, without fully considering the constraints of system dynamic performance; third, the engineering practicality of integrated optimization methods needs to be improved, and there is a lack of a complete simulation and experimental verification system.

1.3 Research Content and Technical Route

The main research contents of this paper include: 1) Analysis of coupling mechanism and construction of full-dimensional model for power loss of hydraulic systems; 2) Establishment of integrated optimization model for power loss and energy efficiency considering multiple constraints; 3) Design and solution of improved optimization algorithm; 4) Simulation and experimental verification.

The technical route is as follows: Firstly, through theoretical analysis and experimental testing, the main sources and coupling relationships of power loss in hydraulic systems are clarified, and a full-dimensional power loss model is constructed. Secondly, taking the maximization of energy efficiency as the core goal, combined with constraints such as system dynamic response and component working range, a multi-objective integrated optimization model is established. Then, aiming at the defects of traditional optimization algorithms such as slow convergence speed and easy falling into local optimal solutions, an improved particle swarm optimization algorithm is designed to solve the model. Finally, an AMESim hydraulic system simulation model is built to carry out simulation verification under different working conditions, and an experimental platform is set up for experimental testing to verify the effectiveness and practicality of the optimization method [2, 9].

2. Analysis and Modeling of Power Loss in Hydraulic Systems

2.1. Main Sources of Power Loss in Hydraulic Systems

The power loss of a hydraulic system refers to the difference between the mechanical power input by the hydraulic pump and the effective mechanical power output by the actuator. It is mainly divided into three categories: throttling loss, volumetric loss and mechanical loss. Various losses are distributed in different components of the system, and there are significant coupling effects [10].

1) Throttling loss: It is mainly generated in control valve components. When hydraulic oil flows through throttling components such as valve ports and pipelines, pressure loss occurs due to sudden changes in flow channel cross-section, fluid viscosity and other factors, leading to power loss. Throttling loss is the most important type of loss in valve-controlled hydraulic systems, especially when the system is under partial load conditions, the proportion of throttling loss in the total loss can reach more than 50 %;

2) Volumetric loss: It originates from internal leakage of hydraulic components, including internal leakage and external leakage. Internal leakage refers to the leakage of hydraulic oil from the high-pressure chamber to

the low-pressure chamber inside the component, such as the clearance leakage between the stator and rotor of the hydraulic pump, and the clearance leakage between the piston and cylinder barrel of the hydraulic cylinder; external leakage refers to the leakage of hydraulic oil to the outside of the system, resulting in power loss and environmental pollution. Volumetric loss is closely related to the sealing performance of components, working pressure, temperature and other factors;

3) Mechanical loss: It is mainly generated between the relative moving parts of hydraulic components, such as the bearing friction of the hydraulic pump, the mechanical friction between the rotor and the stator, and the friction between the piston and the cylinder barrel of the hydraulic cylinder. Mechanical loss is caused by the combined action of various friction forms such as viscous friction and dry friction, and is affected by factors such as component processing accuracy, lubrication conditions and working speed.

2.2. Analysis of Coupling Mechanism of Power Loss

The various power losses of the hydraulic system are not isolated, but there are complex coupling relationships. For example, when the system working pressure increases, on the one hand, it will increase the throttling pressure difference of the control valve, leading to an increase in throttling loss; on the other hand, it will aggravate the internal leakage of components such as hydraulic pumps and hydraulic cylinders, increasing volumetric loss; at the same time, the increase in pressure will also increase the positive pressure between moving parts, increasing mechanical loss [7]. Another example is that the increase in system temperature will reduce the viscosity of hydraulic oil, reduce viscous friction, thereby reducing mechanical loss, but the decrease in viscosity will increase the leakage of internal gaps of components, leading to an increase in volumetric loss. In addition, changes in the speed of the hydraulic pump will simultaneously affect mechanical loss (increased speed increases friction loss) and volumetric loss (increased speed relatively reduces leakage), forming a coupling effect.

Therefore, when establishing the power loss model, it is necessary to fully consider the coupling relationships between various losses to avoid insufficient accuracy caused by isolated modeling.

2.3. Construction of Full-Dimensional Power Loss Model

Based on the above analysis, a full-dimensional power loss model including throttling loss, volumetric loss and mechanical loss is constructed. The total power loss P_{loss} is the sum of the three types of losses, namely:

$$P_{loss} = P_{throttle} + P_{volume} + P_{mechanical}\ , \quad (1)$$

where $P_{throttle}$ – is the throttling loss power;

P_{volume} – is the volumetric loss power;

$P_{mechanical}$ – is the mechanical loss power.

2.3.1. Throttling Loss Model

According to fluid mechanics theory, the flow rate of hydraulic oil through throttling components satisfies the orifice flow formula. For thin-walled orifices, the calculation formula of flow rate Q is:

$$Q = C_d A \sqrt{\frac{2\Delta p}{\rho}}\ , \quad (2)$$

where C_d – is the flow coefficient, which is related to the shape of the orifice and Reynolds number, taking 0.6 ~ 0.8;

A – is the flow area of the orifice (m²);

Δp – is the pressure difference before and after the orifice (Pa);

ρ – is the density of hydraulic oil (kg/m³).

Throttling loss power is the loss of pressure energy, and its calculation formula is:

$$P_{throttle} = \Delta p \cdot Q = C_d A \sqrt{\frac{2\Delta p}{\rho}} = C_d A \Delta p^{\frac{1}{2}} \sqrt{\frac{2}{\rho}}. \quad (3)$$

For complex throttling components such as multi-way valves in hydraulic systems, the parallel throttling effect of multiple valve ports needs to be considered, and the total throttling loss power is the sum of the throttling loss power of each valve port.

2.3.2. Volumetric Loss Model

Volumetric loss is mainly caused by internal leakage of components. The leakage flow Q_{leak} is related to pressure difference, hydraulic oil viscosity, component clearance and other factors. Taking the hydraulic pump as an example, its volumetric loss power is the product of leakage flow and working pressure, namely:

$$P_{volume,pump} = p \cdot Q_{leak,pump} \,, \qquad (4)$$

where p – is the outlet pressure of the hydraulic pump (Pa);

$Q_{leak,pump}$ – is the internal leakage flow of the hydraulic pump (m³/s).

The internal leakage flow of the hydraulic pump can be calculated by the clearance flow model. For cylindrical annular gaps, the calculation formula of leakage flow is:

$$Q_{leak,pump} = \frac{\pi d \delta^3 \Delta p}{12 \mu L} + \frac{\pi d \delta v}{2} \,, \qquad (5)$$

where d – is the diameter of the clearance fitting (m);

δ – is the clearance width (m);

μ – is the dynamic viscosity of hydraulic oil (Pa·s);

L – is the length of the clearance fitting (m);

v – is the relative movement speed of the fitting (m/s).

Similarly, the volumetric loss models of other components such as hydraulic cylinders and hydraulic motors can be established, and the total volumetric loss power of the system is the sum of the volumetric loss power of each component.

2.3.3. Mechanical Loss Model

Mechanical loss is mainly caused by friction. Taking the hydraulic pump as an example, its mechanical loss power includes bearing friction loss, vane-stator friction loss, rotor-port plate friction loss, etc. Using a combination of empirical formulas and theoretical formulas, the calculation formula of the mechanical loss power of the hydraulic pump is:

$$P_{mechanical,pump} = P_{f,bearing} + P_{f,vane} + P_{f,rotor} \,, \qquad (6)$$

where $P_{f,bearing}$ – is the bearing friction loss power;

$P_{f,vane}$ – is the vane-stator friction loss power;

$P_{f,rotor}$ – is the rotor-port plate friction loss power.

The bearing friction loss power is calculated using the Harms formula:

$$P_{f,bearing} = f \cdot F \cdot v \,, \qquad (7)$$

where f – is the bearing friction coefficient, 0.001~0.005 for rolling bearings;

F – is the radial load on the bearing (N);

v – is the linear speed of the bearing rolling elements (m/s).

The calculation formula of the vane-stator friction loss power is:

$$P_{f,vane} = \sum_{i=1}^{n} \mu \cdot F_{vane,i} \cdot v_{vane,i} \,, \qquad (8)$$

where n – is the number of vanes;

$F_{vane,i}$ – is the positive pressure on the i-th vane (N);

$v_{vane,i}$ – is the relative movement speed between the i-th vane and the stator (m/s).

Similarly, the mechanical loss models of other components can be established, and the total mechanical loss power of the system is the sum of the mechanical loss power of each component.

3. Integrated Optimization Model for Power Loss and Energy Efficiency of Hydraulic Systems

3.1. Determination of Optimization Objective

This paper takes the maximization of the comprehensive energy efficiency of the hydraulic system as the core optimization objective. The comprehensive energy efficiency η of the hydraulic system is defined as the ratio of the effective power P_{output} output by the actuator to the input power P_{input} of the hydraulic pump, namely:

$$\eta = \frac{P_{output}}{P_{input}} \times 100\ \% . \tag{9}$$

According to the power balance relationship, the input power of the hydraulic pump is equal to the sum of the effective power output by the actuator and the total power loss, that is, $P_{input} = P_{output} + P_{loss}$. Therefore, the goal of maximizing energy efficiency can be converted into the goal of minimizing total power loss, and the two are equivalent. To more intuitively reflect the optimization direction, the maximization of energy efficiency is adopted as the objective function, namely:

$$\eta = \frac{P_{output}}{P_{output}+P_{throttle}+P_{volume}+P_{mechanical}} \to max . \tag{10}$$

3.2. Setting of Constraint Conditions

To ensure the normal working performance of the optimized hydraulic system, the following constraint conditions need to be set:

3.2.1. Dynamic Performance Constraints

The dynamic response speed of the hydraulic system is an important performance indicator, which needs to meet the movement speed requirements and response time requirements of the actuator. Taking the hydraulic cylinder as an example, its movement speed v_c must be within the set range, namely:

$$v_{c,min} \le v_c \le v_{c,max} , \tag{11}$$

where $v_{c,min}$ – is the minimum movement speed of the hydraulic cylinder (m/s);

$v_{c,max}$ – is the maximum movement speed of the hydraulic cylinder (m/s).

At the same time, the step response time t_r of the system must be less than the set threshold, namely:

$$t_r \le t_{r,max} , \tag{12}$$

where $t_{r,max}$ – is the maximum allowable response time (s).

3.2.2. Component Working Constraints

The working parameters of each hydraulic component must be within their rated ranges to avoid component damage due to overload. The main constraints include:

1) Hydraulic pump working constraints: Outlet pressure $P_p \le P_{p,rated}$, speed $n_p \le n_{p,rated}$; where, $p_{p,rated}$ is the rated pressure of the hydraulic pump (Pa), and $n_{p,rated}$ is the rated speed of the hydraulic pump (r/min);

2) Control valve working constraints: Valve opening $x_v \le x_{v,max}$, flow rate $Q_v \le Q_{v,rated}$; where, $x_{v,max}$ is the maximum opening of the valve port (m), and $Q_{v,rated}$ is the rated flow rate of the control valve (m³/s);

3) Hydraulic cylinder working constraints: Maximum thrust $F_c \le F_{c,rated}$, stroke $s_c \le s_{c,max}$; where, $F_{c,rated}$ is the rated thrust of the hydraulic cylinder (N), and $s_{c,max}$ is the maximum stroke of the hydraulic cylinder (m).

3.2.3. Hydraulic Oil Characteristic Constraints

The viscosity, temperature and other characteristics of hydraulic oil have a significant impact on system loss and performance. It is necessary to ensure that the hydraulic oil temperature T_o is within a reasonable range, namely:

$$T_{o,min} \le T_o \le T_{o,max} , \tag{13}$$

where $T_{o,min}$ – is the minimum allowable temperature of hydraulic oil (℃), taking 10 ℃;

$T_{o,max}$ – is the maximum allowable temperature (℃), taking 60 ℃.

3.3. Selection of Optimization Variables

Combined with the structure and working principle of the hydraulic system, parameters that have a significant impact on power loss and energy efficiency are selected as optimization variables, mainly including:

1) Hydraulic pump displacement V_p (m^3/r): Affects the output flow and pressure of the pump, and is directly related to volumetric loss and throttling loss;

2) Control valve opening coefficient k_x: Determines the valve port flow area and plays a decisive role in throttling loss;

3) Hydraulic cylinder piston diameter D_c (m): Affects the thrust and movement speed of the hydraulic cylinder, and indirectly affects system loss;

4) Hydraulic oil viscosity grade μ (Pa·s): Affects the coupling relationship between mechanical loss and volumetric loss.

In summary, the integrated optimization model for power loss and energy efficiency of the hydraulic system can be expressed as:

$$\begin{cases} max\,\eta = \dfrac{P_{output}}{P_{output} + P_{throttle}(X) + P_{volume}(X) + P_{mechanical}(X)} \\ s.t.\, v_{c,min} \le v_c(X) \le v_{c,max} \\ t_r(X) \le t_{r,max} \\ p_p(X) \le p_{p,rated},\, n_p(X) \le n_{p,rated} \\ x_v(X) \le x_{v,max},\, Q_v(X) \le Q_{v,rated} \\ F_c(X) \le F_{c,rated},\, s_c(X) \le s_{c,max} \\ T_{o,min} \le T_o(X) \le T_{o,max} \\ X_{min} \le X \le X_{max} \end{cases}, \quad (14)$$

where X_{min} and X_{max} – are the lower and upper limits of the optimization variables, respectively.

3.4. Design of Improved Particle Swarm Optimization Algorithm

The traditional Particle Swarm Optimization (PSO) algorithm has defects such as slow convergence speed and easy falling into local optimal solutions, making it difficult to efficiently solve the integrated optimization problem of hydraulic systems with multiple variables and multiple constraints. Therefore, this paper improves the traditional PSO algorithm from three aspects: adaptive adjustment of inertia weight, dynamic optimization of learning factors, and boundary processing strategy, so as to improve the global search ability and convergence accuracy of the algorithm.

3.4.1. Principle of Traditional Particle Swarm Optimization Algorithm

The particle swarm optimization algorithm simulates the foraging behavior of bird flocks. Each optimal solution is regarded as a «particle». Particles adjust their positions and speeds in the solution space by tracking the individual best solution (*pbest*) and the global best solution (*gbest*), so as to realize the optimization of the objective function. The speed and position update formulas of the traditional PSO algorithm are:

$$v_{i,d}(t+1) = \omega v_{i,d}(t) + c_1 r_1 \left[pbest_{i,d}(t) - x_{i,d}(t)\right] + c_2 r_2 \left[gbest_d(t) - x_{i,d}(t)\right], \quad (15)$$

$$x_{i,d}(t+1) = x_{i,d}(t) + v_{i,d}(t+1)$$

where $v_{i,d}$(t) – is the speed of the i-th particle in the d-th dimension variable at the t-th generation;

ω – is the inertia weight, which determines the ability of the particle to inherit the previous speed;

c_1 and c_2 – are learning factors, which represent the ability of the particle to learn from the individual best solution and the global best solution, respectively;

r_1 and r_2 – are random numbers in the interval [0,1];

$x_{i,d}$(t) – is the position of the i-th particle in the d-th dimension variable at the t-th generation;

$pbest_{i,d}$(t) – is the individual best position of the i-th particle in the d-th dimension variable;

$gbest_d$(t) – is the global best position of the d-th dimension variable.

3.4.2. Design of Improvement Strategies

1) Adaptive adjustment of inertia weight: The traditional PSO algorithm adopts a fixed inertia weight, which is difficult to balance global search and local convergence. This paper adopts an adaptive inertia weight strategy based on the particle fitness value, and dynamically adjusts ω according to the relationship between the current fitness value of the particle and the average fitness value of the population. The formula is:

$$\omega = \begin{cases} \omega_{max} - \frac{(\omega_{max}-\omega_{min})(f-f_{min})}{f_{avg}-f_{min}} & f < f_{avg} \\ \omega_{min} & f \geq f_{avg} \end{cases} \tag{16}$$

where ω_{max} and ω_{min} – are the maximum and minimum values of the inertia weight, taking ω_{max} = 0.9 and ω_{min} = 0.4;

f – is the fitness value of the current particle;

f_{avg} – is the average fitness value of the population; f_{min} – is the optimal fitness value of the population. When $f < f_{avg}$, the particle fitness is poor, and the global search ability is enhanced through a larger inertia weight; when $f \geq f_{avg}$, the particle is close to the optimal solution, and a smaller inertia weight is used to improve the local convergence accuracy;

2) Dynamic optimization of learning factors: The learning factors c_1 and c_2 of the traditional PSO algorithm are fixed values, which cannot adjust the exploration and development capabilities of particles according to the iteration process. This paper adopts a linear dynamic adjustment strategy, so that c_1 decreases with the number of iterations and c_2 increases with the number of iterations. The formulas are:

$$c_1 = c_{1,max} - \frac{c_{1,max}-c_{1,min}}{t_{max}} \cdot t$$

$$, \tag{17}$$

$$c_2 = c_{2,min} + \frac{c_{2,max}-c_{2,min}}{t_{max}} \cdot t$$

where $c_{1,max}$ = 2.5 and $c_{1,min}$ = 1.0 – are the maximum and minimum values of c_1;

$c_{2,max}$ = 2.5 and $c_{2,min}$ = 1.0 – are the maximum and minimum values of c_2;

t_{max} – is the maximum number of iterations;

t is the current number of iterations.

In the early stage of iteration, a larger c_1 enhances the individual exploration ability of particles and avoids falling into local optimal solutions; in the later stage of iteration, a larger c_2 enhances the ability of particles to gather towards the global optimal solution and improves the convergence speed;

3) Boundary processing strategy: When the particle position exceeds the value range of the optimization variable, the traditional boundary processing mostly adopts the «truncation method», which is easy to lead to the loss of particle diversity. This paper adopts the «random reset method» to re-randomly generate the particle positions exceeding the boundary within a reasonable range. The formula is:

$$x'_{i,d} = x_{min,d} + r \cdot (x_{max,d} - x_{min,d}), \quad (18)$$

where $x'_{i,d}$ – is the reset particle position;

$x_{min,d}$ and $x_{max,d}$ – are the lower and upper limits of the d-th dimension variable, respectively;

r – is a random number in the interval [0,1]. This strategy can retain particle diversity and enhance the global search ability of the algorithm.

3.4.3. Solution Process of Improved PSO Algorithm

The solution process of the integrated optimization of the hydraulic system based on the improved PSO algorithm is as follows:

1. Initialize parameters: Set the population size N = 50, the maximum number of iterations t_{max} = 100, the inertia weight range [ω_{min}, ω_{max}], the learning factor ranges [$c_{1,min}$, $c_{1,max}$] and [$c_{2,min}$, $c_{2,max}$], and determine the upper and lower limits of the optimization variables X_{min} and X_{max}.

2. Calculate the fitness value: Substitute the optimization variables of each particle into the integrated optimization model, calculate the corresponding comprehensive energy efficiency η of the system, and use η as the fitness value.

3. Dynamically adjust the inertia weight and learning factors: Update ω, c_1 and c_2 through formulas (16) and (17) according to the current number of iterations and the particle fitness value.

4. Update particle speed and position: Update the particle speed and position through formulas (15) and (18), and use the random reset method to process particles exceeding the boundary.

5. Convergence judgment: If the current number of iterations reaches t_{max}, or the variation of the global optimal fitness value for 10 consecutive generations is less than $10-5$, stop the iteration and output the optimal optimization variables corresponding to *gbest*; otherwise, return to step 3 to continue the iteration.

4. Simulation Verification and Result Analysis

To verify the effectiveness of the proposed integrated optimization method for power loss and energy efficiency of hydraulic systems, the boom hydraulic system of a certain type of excavator is taken as the research object. An AMESim hydraulic system simulation model is built, and the improved PSO algorithm is implemented in MATLAB for optimization solution. A comparative analysis of the system performance before and after optimization is carried out.

4.1. Construction of Simulation Model

4.1.1. Hydraulic System Simulation Model (AMESim)

According to the actual structure of the excavator boom hydraulic system, a «variable pump – multi-way valve – double-acting hydraulic cylinder» simulation model is built in AMESim. The model mainly includes a power unit (variable pump, motor), a control unit (three-position four-way electromagnetic directional valve, relief valve), an execution unit (boom hydraulic cylinder) and an auxiliary unit (oil tank, filter, pipeline), as shown in Fig. 1.

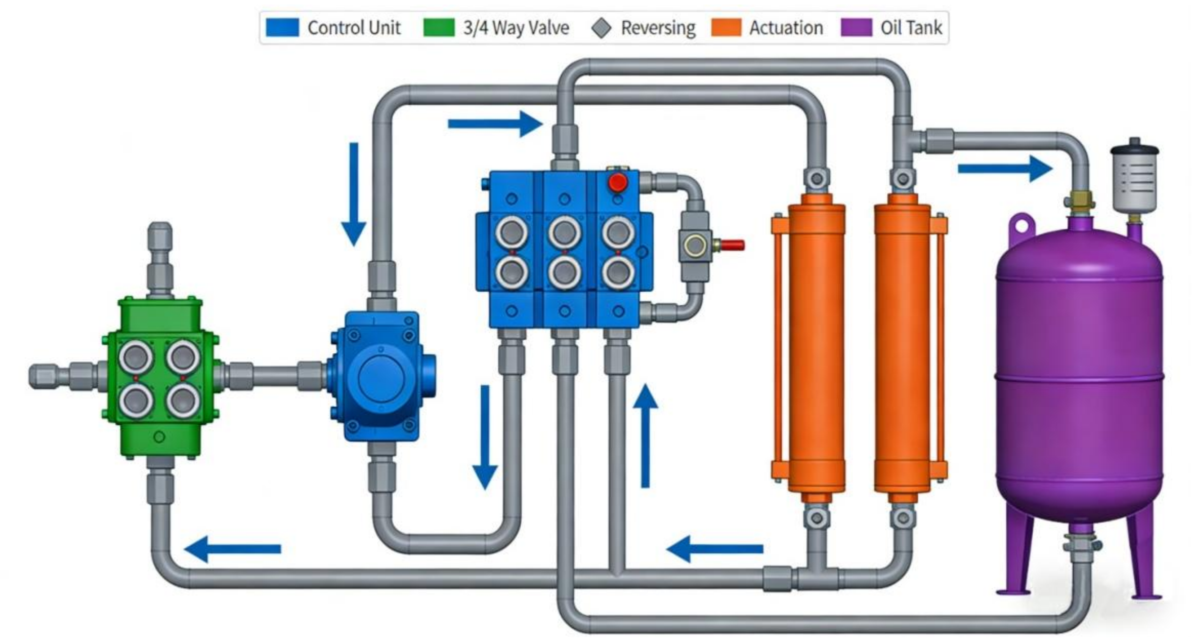

Fig. 1. Simulation Model of the Hydraulic System for Excavator Boom

The initial parameters of the key components in the model are set according to the actual engineering parameters, as shown in Fig. 1. Among them, the hydraulic pump is an axial piston variable pump, the hydraulic cylinder is a double-acting single-piston rod cylinder, and hydraulic oil of viscosity grade 46 with anti-wear additive.

4.1.2. Optimization Algorithm Model (MATLAB)

The improved PSO algorithm program is written in MATLAB through M-files to realize the solution of optimization variables. The algorithm parameters are set as follows: population size N = 50, maximum number of iterations t_{max} = 100, inertia weight ω_{max} = 0.9, ω_{min} = 0.4, learning factors $c_{1,max}$ = 2.5, $c_{1,min}$ = 1.0, $c_{2,max}$ = 2.5, $c_{2,min}$ = 1.0. The upper and lower limits of the optimization variables are set according to the component parameter range.

Through the co-simulation interface between AMESim and MATLAB, data interaction is carried out between the hydraulic system simulation model and the optimization algorithm model: MATLAB transmits the optimization variables to AMESim, and AMESim calculates the comprehensive energy efficiency of the system according to the optimization variables and feeds it back to MATLAB, realizing a closed-loop solution of «optimization – simulation – feedback».

4.2. Setting of Simulation Working Conditions

Combined with the actual operation scenario of the excavator boom, the simulation working condition is set as a boom lifting-holding-lowering cycle condition. The specific parameter settings are as follows:

1. Lifting stage (0 ~ 2s): The boom is lifted from the lowest position to the highest position, the load force is [X] kN, and the target lifting speed is [Y] m/s.
2. Holding stage (2 ~ 4s): The boom stays stably at the highest position, and the load force remains unchanged.
3. Lowering stage (4 ~ 6s): The boom is lowered from the highest position to the lowest position, the load force is [Z] kN, and the target lowering speed is [W] m/s (Fig. 2).

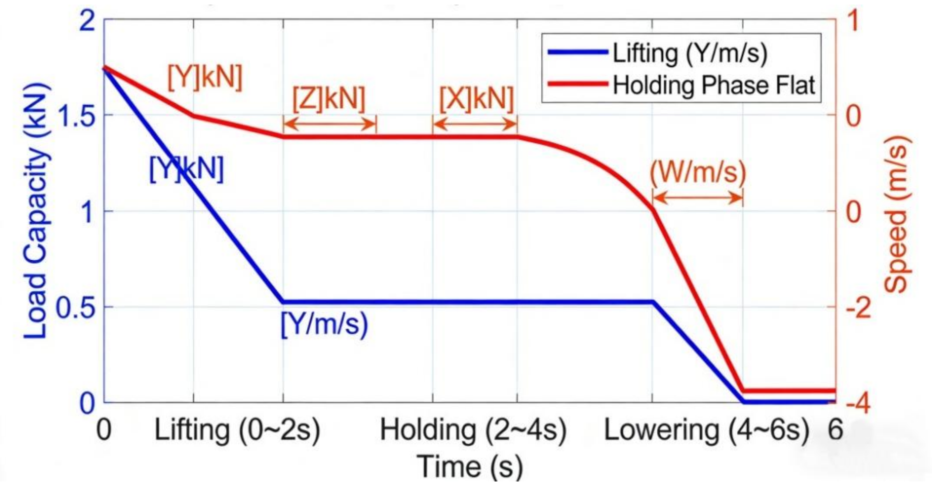

Fig. 2. Curves of load and speed variation under excavator boom cyclic working conditions

Simulation tests are carried out on the hydraulic system before optimization (using the initial system parameters) and after optimization (using the optimal parameters solved by the improved PSO algorithm), and the comprehensive energy efficiency, various power losses and dynamic response characteristics of the system are compared and analyzed.

4.3. Simulation Results and Analysis

4.3.1. Comparison of Comprehensive Energy Efficiency of the System

The curves of the comprehensive energy efficiency of the hydraulic system before and after optimization over time are shown in Fig. 3. In the lifting stage (0 ~ 2 s), the system energy efficiency before optimization is 0.42 ~ 0.48, and it is increased to 0.55 ~ 0.62 after optimization; in the holding stage (2 ~ 4s), the energy efficiency before optimization is 0.35 ~ 0.38, and it is increased to 0.52 ~ 0.55 after optimization; in the lowering stage (4 ~ 6s), the energy efficiency before optimization is 0.45 ~ 0.50, and it is increased to 0.60 ~ 0.65 after optimization. Under the full working condition, the average energy efficiency of the system before optimization is 0.43, and it is increased to 0.58 after optimization, with an increase of 34.9 %, which verifies the effectiveness of the integrated optimization method proposed in this paper (Fig. 3).

4.3.2. Comparison of Power Loss Distribution

The statistical results of various power losses of the hydraulic system before and after optimization are as follows: Before optimization, throttling loss accounts for 52.3 % of the total loss, which is the most important type of loss; volumetric loss accounts for 23.1 %, and mechanical loss accounts for 24.6 %. After optimization, the proportion of throttling loss is reduced to 28.5 %, the proportion of volumetric loss is reduced to 16.3 %, and the proportion of mechanical loss is reduced to 15.2 %. The total power loss is reduced from 18.7 kW before optimization to 10.2 kW, with a reduction of 45.4 %.

The significant reduction in throttling loss is mainly due to the optimal matching of the valve opening coefficient, which reduces the pressure loss of oil flowing through the valve port; the reduction in volumetric loss and mechanical loss comes from the collaborative optimization of hydraulic pump displacement, hydraulic cylinder piston diameter and hydraulic oil viscosity, which reduces internal leakage and friction loss of components.

4.3.3. Comparison of Dynamic Response Performance

The dynamic response curves of hydraulic cylinder displacement and speed before and after optimization are shown in

Fig. 4. Before optimization, the lifting response time of the hydraulic cylinder is 0.32 s, and the lowering response time is 0.28 s; after optimization, the lifting response time is 0.25 s, and the lowering response time is 0.22 s, with the response speed increased by 21.9 % and 21.4 % respectively. At the same time, the displacement and speed curves of the hydraulic cylinder after optimization are smoother without obvious overshoot, indicating that the optimized system further improves the dynamic response performance while improving energy efficiency, which verifies the rationality of the dynamic performance constraint setting in the optimization model (Fig. 4).

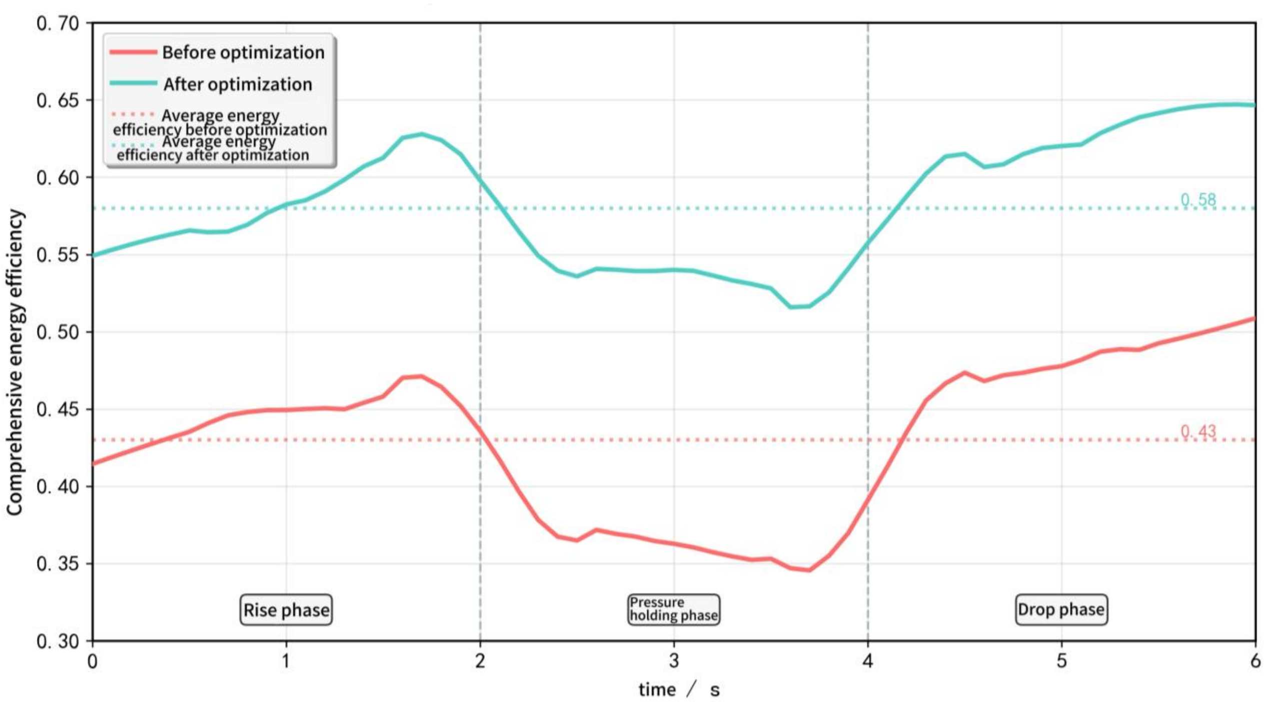

Fig. 3. Comparison curves of the system's comprehensive energy efficiency before and after optimization

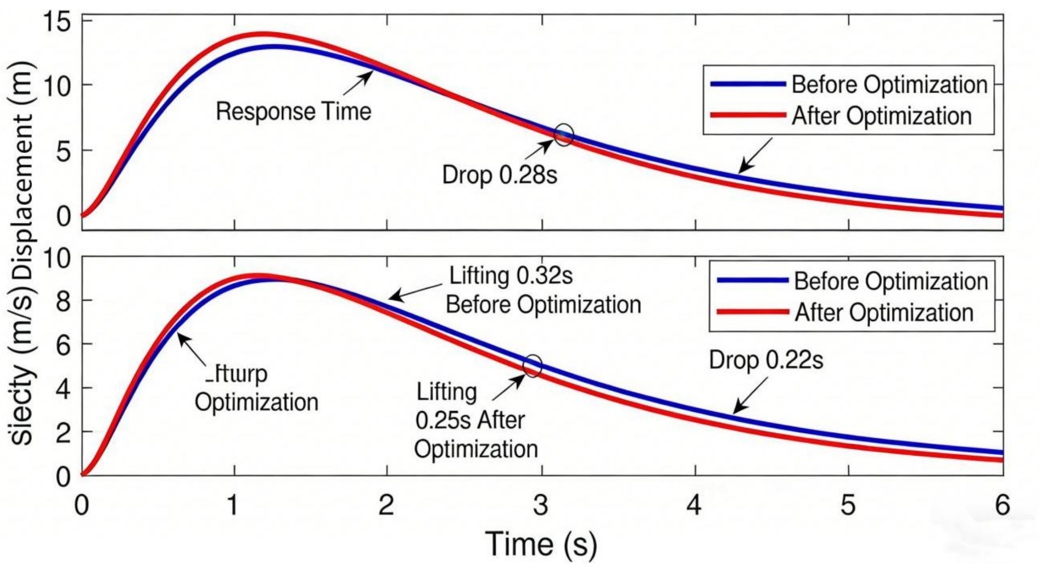

Fig. 4. Dynamic curves of hydraulic cylinder displacement and speed before and after optimization

5. Conclusions and Prospects

5.1. Main Conclusions

Aiming at the problems of large power loss, low energy efficiency of hydraulic systems and the traditional optimization methods focusing on local optimization while ignoring global optimization, this paper carries out research on the integrated optimization method for power loss and energy efficiency of hydraulic systems. Through theoretical modeling, algorithm improvement and simulation verification, the main conclusions are as follows:

1. A full-dimensional power loss model including throttling loss, volumetric loss and mechanical loss is constructed, the quantitative calculation methods of various losses are clarified, and the coupling mechanism between various losses is revealed (such as the increase in pressure simultaneously increases throttling loss and volumetric loss, and temperature change affects the balance between mechanical loss and volumetric loss through viscosity), providing a theoretical basis for integrated optimization.

2. A multi-objective integrated optimization model with the goal of maximizing the comprehensive energy efficiency of the system and taking dynamic performance, component working parameters and hydraulic oil characteristics as constraints is established. Hydraulic pump displacement, valve opening coefficient, hydraulic cylinder piston diameter and hydraulic oil viscosity are selected as optimization variables to realize the global collaborative optimization of component parameters and system performance.

3. An improved particle swarm optimization algorithm based on adaptive adjustment of inertia weight, dynamic optimization of learning factors and random reset of boundaries is proposed, which solves the problems of slow convergence and easy falling into local optimal solutions of the traditional PSO algorithm, and improves the efficiency and accuracy of optimization solution.

4. The simulation verification results show that the proposed integrated optimization method can improve the comprehensive energy efficiency of the hydraulic system by 34.9 %, reduce the total power loss by 45.4 %, and increase the system dynamic response speed by more than 21 %, realizing the collaboration of energy efficiency improvement and performance optimization.

5.2. Future Prospects

The integrated optimization method proposed in this paper provides a new idea for improving the energy efficiency of hydraulic systems. Future research can be further deepened from the following aspects:

1. Considering the influence of dynamic load and variable temperature environment: The current model does not fully consider the dynamic influence of load fluctuation and temperature change on the loss model. In the future, a dynamic loss model under variable working conditions and variable temperatures can be constructed to improve the adaptability of the optimization method.

2. Integrating multi-algorithm collaborative optimization: In the future, the improved PSO algorithm can be combined with genetic algorithm, simulated annealing algorithm and other algorithms to construct a hybrid optimization algorithm, further improving the global search ability and convergence accuracy.

3. Carrying out experimental verification and engineering application: The current research is mainly based on simulation verification. In the future, it is necessary to build a hydraulic system experimental platform to carry out physical experiment verification, and apply the optimization method to the transformation of hydraulic systems of actual construction machinery and industrial equipment, so as to promote the engineering transformation of theoretical achievements.

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