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**RESEARCH ON THEORY AND METHODS OF ENERGY EFFICIENCY  
IMPROVEMENT FOR HYDRAULIC SYSTEMS OF PROCESS MACHINERY  
ORIENTED TO LOW-LOSS CHARACTERISTICS**

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**ДОСЛІДЖЕННЯ ТЕОРІЙ І МЕТОДІВ ПІДВИЩЕННЯ ЕНЕРГОЕФЕКТИВНОСТІ  
ГІДРАВЛІЧНИХ СИСТЕМ ТЕХНОЛОГІЧНИХ МАШИН З ОРІЄНТАЦІЄЮ  
НА ХАРАКТЕРИСТИКИ НИЗЬКИХ ВТРАТ**

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**Abstract.** Hydraulic systems are widely employed in process machinery such as presses, injection molding machines, rolling mills, and mining equipment due to their high power density, rapid dynamic response, precise force and speed regulation, and excellent adaptability to severe working environments. Nevertheless, conventional hydraulic systems for process machinery commonly suffer from excessive energy consumption, substantial power dissipation, and relatively low energy efficiency, which have become critical bottlenecks restricting the sustainable and low-carbon development of industrial equipment. Guided by a systematic low-loss design philosophy, this paper presents a comprehensive theoretical and methodological investigation into energy efficiency enhancement by integrating fluid mechanics, modern control theory, and intelligent optimization algorithms. The fundamental working principles and inherent energy consumption characteristics of typical hydraulic systems are systematically analyzed, aiming to accurately identify the primary loss sources and reveal the coupling mechanisms among different components and hydraulic circuits. A full-process low-loss dynamic mathematical model, encompassing key hydraulic components and pipeline networks, is established, and its accuracy, reliability, and practical applicability are thoroughly validated through parameter identification, numerical simulation, and experimental testing. On this basis, an integrated energy consumption optimization control strategy, combining high-fidelity dynamic modeling with an improved genetic algorithm, is proposed to achieve adaptive and real-time dynamic adjustment of system operating parameters. Simulation and experimental results demonstrate that the developed control strategy can effectively mitigate volumetric loss, throttling loss, and pipeline pressure loss under diverse working conditions. Under typical operational scenarios, the overall energy efficiency of the hydraulic system is improved by 18–25 %, while simultaneously enhancing dynamic performance. The research outcomes provide solid theoretical foundations and practical engineering guidance for the low-loss design, performance optimization, and energy-saving retrofitting of hydraulic systems for process machinery.

**Keywords:** technological machines, hydraulic system, low loss, energy efficiency improvement, dynamic modeling, optimal control, simulation analysis.

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**Анотація.** Гідравлічні системи широко застосовувані в технологічному обладнанні, такому як преси, ін'єкційно-формувальні машини, прокатні стани та гірничодобувне обладнання, завдяки високій питомій потужності, швидкій динамічній реакції, точному регулюванню сили та швидкості, а також чудовій адаптивності до складних умов експлуатації. Проте для традиційних гідравлічних систем технологічного обладнання характерні надмірне енергоспоживання, значні втрати потужності та відносно низька енергоефективність, що стало основною перешкодою для сталого та низьковуглецевого розвитку промислового обладнання.

Керуючись системною концепцією проектування з мінімальними втратами, у статті проведено комплексне теоретичне та методичне дослідження підвищення енергоефективності на основі поєднання гідродинаміки, сучасної теорії керування та інтелектуальних оптимізаційних алгоритмів. Систематично проаналізовано основні принципи роботи і внутрішні особливості енергоспоживання типових гідравлічних систем із метою точного визначення основних джерел втрат і розкриття механізмів взаємодії між окремими елементами та гідравлічними контурами.

Побудовано повноциклову динамічну математичну модель із мінімальними втратами, що враховує основні гідравлічні компоненти і трубопровідні мережі; її точність, надійність і практичну придатність повністю підтверджено ідентифікацією параметрів, чисельним моделюванням та експериментальними випробуваннями. На цій основі запропоновано комплексну стратегію оптимізаційного керування енергоспоживанням, яка поєднує високоточне динамічне моделювання та вдосконалений генетичний алгоритм для забезпечення адаптивного оперативного динамічного корегування робочих параметрів системи.

Результати моделювання та експериментів підтверджують, що розроблена стратегія керування ефективно зменшує об'ємні втрати, дросельні втрати і втрати тиску в трубопроводах за різних режимів роботи. У типових експлуатаційних умовах загальна енергоефективність гідравлічної системи зростає на 18–25 %, водночас покращуючи її динамічні характеристики. Отримані результати дослідження створюють надійну теоретичну базу та практичні інженерні рекомендації для проектування гідравлічних систем технологічного обладнання з мінімальними втратами, оптимізації їхніх характеристик і модернізації з метою енергозбереження.

**Ключові слова:** технологічні машини, гідравлічна система, низькі втрати, підвищення енергоефективності, динамічне моделювання, оптимальне керування, симуляційний аналіз.

## 1. Introduction

### 1.1. Research Background and Significance

Hydraulic transmission technology has long been recognized as one of the most reliable and efficient power transmission solutions for heavy-duty process machinery. Owing to its outstanding advantages including high power-to-weight ratio, smooth motion transmission, accurate positioning capability, and strong resistance to shock and vibration, hydraulic systems have become the core power and control unit in a wide range of industrial equipment, such as metal forming presses,

plastic injection molding machines, steel rolling mills, and mining excavators. In these applications, hydraulic systems undertake the critical functions of power output, precise motion execution, and multi-axis coordinated control, directly determining the operational performance, production efficiency, and energy consumption level of the entire equipment [1, 2, 5].

Despite their widespread application and superior mechanical performance, traditional hydraulic systems for process machinery inherently exhibit low energy efficiency. Industrial field measurements indicate that the

average energy efficiency of conventional hydraulic systems typically ranges only from 45 % to 60 %, which is significantly lower than that of advanced electric drive systems. A large proportion of the input electrical energy is wasted in the form of internal leakage, mechanical friction, flow throttling, and pipeline pressure drop, resulting in excessive energy consumption, increased operating costs, and aggravated environmental impact. Against the backdrop of intensifying global energy shortages, rising energy prices, and the global pursuit of carbon neutrality and sustainable development goals, industrial sectors have imposed increasingly stringent requirements for energy conservation, emission reduction, and green manufacturing. As a typical representative of high-energy-consumption equipment, improving the energy efficiency of hydraulic systems for process machinery has become an urgent and important research topic in the field of fluid power transmission and control [3, 4].

Conducting in-depth research on the theories and methods of energy efficiency improvement for hydraulic systems under low-loss orientation not only helps to reduce energy consumption and operating costs, but also contributes to reducing carbon emissions and environmental pollution, thereby promoting the green transformation and sustainable development of the process machinery industry. Furthermore, reducing system losses can effectively lower the operating temperature, decrease component wear, extend the service life of key parts, and improve the operational stability and reliability of the hydraulic system, which is of great practical significance for reducing equipment maintenance costs and enhancing overall industrial competitiveness. Therefore, this research holds important theoretical value and broad engineering application prospects [6, 11, 13, 15].

#### *1.2. Research Status at Home and Abroad*

In recent decades, scholars and engineers worldwide have conducted extensive and in-depth research on energy consumption optimization, low-loss design, and performance

improvement of hydraulic systems, forming a multidimensional research system mainly focusing on component optimization, hydraulic circuit innovation, and control strategy design, which provides a solid theoretical basis and technical reference for the present study.

Internationally, developed industrial countries have long attached great importance to the research and development of high-efficiency, low-loss hydraulic components and system integration technology. A large number of energy-saving hydraulic solutions have been proposed and applied in engineering practice, such as variable displacement pump control systems, load-sensing control systems, and electro-hydraulic proportional control systems. For example, the electro-hydraulic proportional variable displacement pump developed by Bosch Rexroth precisely adjusts the pump displacement and output pressure according to the actual load demand, effectively matching the system flow with the load requirement and significantly reducing volumetric leakage loss. Yuken has developed a series of energy-saving hydraulic valves and optimized circuit structures, which minimize throttling loss by reducing the number of throttling components and optimizing the flow channel design. In addition, many foreign scholars have introduced intelligent optimization algorithms such as genetic algorithms, particle swarm optimization, and neural networks into hydraulic system control, realizing the dynamic optimization of system parameters under complex working conditions [6, 8, 9, 12]. However, most existing studies are aimed at general hydraulic systems, lacking targeted research on the unique characteristics of process machinery, such as intermittent load, drastic load fluctuation, and high-frequency start-stop operation. Moreover, the optimization algorithms adopted generally have slow convergence speed and poor real-time performance, making it difficult to meet the requirements of practical engineering applications [7, 20].

Domestically, research on energy efficiency improvement of hydraulic systems

has made rapid progress in recent years, mainly focusing on energy-saving transformation and control strategy optimization of existing hydraulic systems. Many scholars have applied variable-frequency speed regulation technology, energy-saving valve groups, and accumulator energy storage technology to reduce no-load energy consumption and throttling loss of hydraulic systems. Load-sensing control technology has been widely studied and applied to optimize the structure of hydraulic circuits and realize dynamic matching between system flow and load demand. In addition, domestic researchers have made important progress in hydraulic system modeling and simulation, using advanced simulation software such as AMESim and Simulink to analyze the distribution rules and influencing factors of system losses, providing theoretical support for energy efficiency optimization [8, 14].

Nevertheless, the current research still has obvious shortcomings and limitations. First, the analysis of the loss mechanism of hydraulic systems is not thorough enough, and most studies only focus on a single loss factor, lacking systematic research on the coupling characteristics of multiple losses under full-system and full-working-condition operation. Second, most energy efficiency improvement methods adopt a single optimization means, lacking collaborative optimization of components, circuits, and control strategies, which makes it difficult to maximize the energy-saving potential of the system. Third, the engineering practicability of the proposed optimization methods is insufficient, and many strategies rely on high-precision components, complex control structures, or expensive hardware, which are difficult to be widely promoted and applied in ordinary process machinery. Therefore, developing a comprehensive, condition-adaptive, and low-cost low-loss energy efficiency improvement method has become a research hotspot and technical difficulty in the field of hydraulic transmission [10, 16].

### 1.3. Research Content and Innovations

Aiming at the above problems and challenges, this paper conducts a systematic and in-depth study on the low-loss design and energy efficiency improvement of hydraulic systems for process machinery. Through dynamic modeling, optimal control design, numerical simulation, and experimental verification, a set of scientific, practical, and efficient energy efficiency improvement theories and methods are proposed. The main research contents are as follows:

**Analysis of Operating Principles and Energy Consumption Characteristics of Hydraulic Systems:** Clarify the typical composition, structural characteristics, and working logic of hydraulic systems for process machinery; systematically analyze the energy consumption distribution and loss sources of key components and hydraulic circuits; establish the quantitative relationship between system losses and structural parameters, operating parameters, and working condition parameters; reveal the coupling mechanism and interaction law of different losses, laying a solid theoretical foundation for the design of energy efficiency improvement strategies.

**Low-Loss Oriented Dynamic Modeling of Hydraulic Systems:** Based on fluid mechanics, mechanical dynamics, and hydraulic transmission theory, establish accurate low-loss mathematical models for core hydraulic components including axial piston pumps, double-acting hydraulic cylinders, relief valves, and throttle valves; construct the overall dynamic energy consumption model of the hydraulic system by integrating the component models according to the hydraulic circuit topology; verify the accuracy and practicability of the model through parameter identification, numerical simulation, and experimental testing.

**Research on Energy Consumption Optimization Control Strategy:** Define the comprehensive optimization objectives and constraint conditions for energy efficiency improvement; design an improved genetic algorithm with strong global search ability and

fast convergence speed; propose an integrated energy consumption optimization control strategy combining high-fidelity dynamic modeling and the improved genetic algorithm; realize adaptive and real-time dynamic adjustment of system operating parameters such as pump speed and throttle valve opening, minimizing system energy consumption while ensuring dynamic performance.

**Simulation and Experimental Verification:** Build a multi-domain collaborative simulation platform based on AMESim and Simulink to simulate typical working conditions of process machinery and verify the effectiveness and feasibility of the proposed optimization control strategy; construct a physical experimental platform for hydraulic systems, conduct comparative experiments under different load conditions, and further confirm the correctness of theoretical analysis and simulation results, ensuring the engineering practicability and reliability of the research outcomes.

The main innovations of this research are summarized as follows:

Proposed a full-process loss coupling dynamic model for hydraulic systems: Breaking through the limitation of single loss analysis in existing research, this model comprehensively considers the coupling interaction between component loss, circuit loss, and control loss, enabling accurate prediction of system energy consumption and precise localization of key loss sources.

Developed a low-loss dynamic modeling method adapted to dynamic working conditions: Combined with parameter identification technology, this method effectively improves the adaptability and accuracy of the model in actual engineering environments, providing reliable theoretical support for the design of high-performance optimal control strategies.

Proposed an integrated optimal control strategy combining dynamic modeling and improved genetic algorithm: This strategy realizes collaborative optimization of components, circuits, and control, effectively

solving the problems of poor working condition adaptability and limited energy efficiency improvement effect of traditional optimization methods, with low engineering implementation cost and strong universality.

## **2. Operating Principle and Energy Consumption Characteristics of Process Machinery Hydraulic Systems**

### *2.1. Basic Composition of Process Machinery Hydraulic Systems*

The core function of a hydraulic system for process machinery is to convert the mechanical energy of the driving motor into hydraulic energy, and then drive the actuator to realize precise and reliable execution of various process actions through the transmission and control of hydraulic oil. A typical hydraulic system (Fig. 1) consists of four basic parts: power components, actuators, control components, and auxiliary components, which cooperate with each other to complete the transmission, control, and conversion of energy [18, 19].

**Power Components:** The core component is the hydraulic pump, which serves as the «power source» of the hydraulic system. Its main function is to convert the mechanical energy output by the motor into hydraulic energy (pressure energy and kinetic energy) and supply high-pressure hydraulic oil to the system. Common types of hydraulic pumps in process machinery include gear pumps, vane pumps, and piston pumps. Among them, axial piston pumps are widely used in medium and high-pressure process machinery such as rolling mills and injection molding machines due to their high volumetric efficiency, high working pressure, and good variable performance.

**Actuators:** Mainly include hydraulic cylinders and hydraulic motors, which are the «executors» of the hydraulic system. Their function is to convert hydraulic energy back into mechanical energy to drive the working parts of process machinery to perform various motions. Hydraulic cylinders are used to realize linear reciprocating motion, such as the stamping motion of presses and the clamping

motion of injection molding machines. Hydraulic motors are used to realize continuous

rotary motion, such as the rotation of drums in mining machinery and the drive of conveyors.

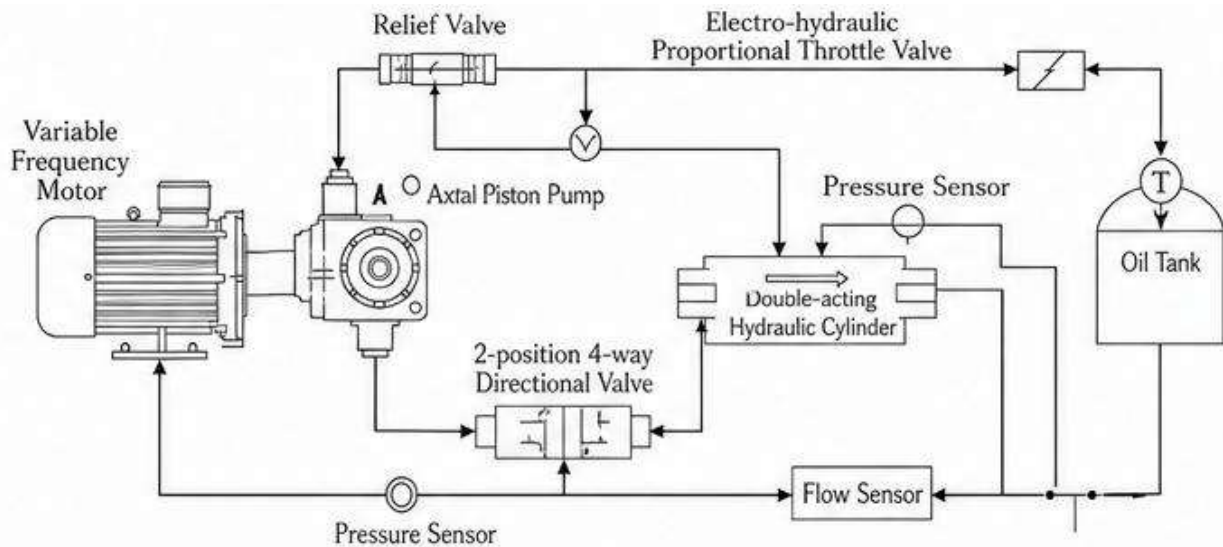


Fig. 1. Schematic diagram of the electro-hydraulic system

**Control Components:** Including pressure control valves, flow control valves, and directional control valves, which are the «command center» of the hydraulic system. Their main function is to adjust and control the pressure, flow rate, and flow direction of hydraulic oil in the system, ensuring that the actuator can accurately and stably complete the required actions according to the process requirements. For example, relief valves are used to stabilize system pressure and prevent overload; throttle valves are used to adjust the movement speed of actuators; directional valves are used to control the start-stop and commutation of actuators.

**Auxiliary Components:** Including oil tanks, oil filters, oil pipes, pipe joints, pressure gauges, and temperature sensors, etc. Although these components do not directly participate in energy conversion, they play an important role in ensuring the normal operation, reducing energy loss, and protecting the main components of the system. For example, the oil tank is used to store hydraulic oil, dissipate heat, and precipitate impurities; the oil filter is

used to filter impurities in hydraulic oil to prevent component wear; oil pipes are used to transmit hydraulic oil, and their diameter, length, and layout directly affect the pressure loss of the hydraulic circuit.

## 2.2. Energy Consumption Analysis of Hydraulic Systems

The energy consumption of a hydraulic system is essentially the difference between the input power provided by the motor and the effective power output by the actuator to complete useful work, i.e., the total energy loss of the system. Input power refers to the mechanical power transmitted by the motor to the hydraulic pump, while effective power refers to the mechanical power output by the actuator to drive the load. The difference between the two is the total energy loss of the system, which is dissipated in the form of heat, noise, and vibration. Combined with the complex and changeable working condition characteristics of process machinery, the energy loss of the hydraulic system can be divided into three main categories: component loss, circuit loss, and control loss [17, 20].

**Component Loss:** Accounting for 60–75 % of the total system loss, it is the main source of energy consumption. Component loss mainly occurs in three core components: hydraulic pumps, hydraulic cylinders, and hydraulic valves. The loss of hydraulic pumps includes volumetric loss (internal leakage) and mechanical loss (friction between moving parts). The loss of hydraulic cylinders mainly includes leakage loss (internal and external leakage) and friction loss (friction between piston and cylinder barrel). The loss of hydraulic valves is mainly throttling loss, which is the pressure loss caused by the throttling effect of the valve port when hydraulic oil flows through the valve.

**Circuit Loss:** Accounting for 15–25 % of the total system loss, it mainly occurs in hydraulic pipelines and auxiliary components. Circuit loss includes along-path pressure loss and local pressure loss. Along-path pressure loss is the viscous friction loss generated when hydraulic oil flows in a straight pipe section, which is related to pipe length, pipe diameter, oil viscosity, and flow velocity. Local pressure loss is the flow loss generated when hydraulic oil passes through local components such as pipeline elbows, tees, and pipe joints, which is related to the structure and shape of local components.

**Control Loss:** Accounting for 7.5 % of the total system loss, it is the energy waste caused by unreasonable control strategies. The working conditions of process machinery have strong dynamic and time-varying characteristics, such as frequent load changes, intermittent operation, and high-frequency start-stop. Traditional fixed-parameter control strategies cannot adaptively adjust system operating parameters according to real-time working condition changes, resulting in the system still operating at high pressure and large flow under light-load or no-load conditions, generating a large amount of redundant energy consumption.

The input power of the hydraulic pump, as the key power component, is calculated as follows:

$$P_{in} = \frac{P_p \cdot Q_p}{\eta_v \cdot \eta_m}, \quad (1)$$

where  $P_{in}$  – is the input power of the hydraulic pump, W;

$P_p$  – is the output pressure of the hydraulic pump, Pa;

$Q_p$  – is the actual output flow of the hydraulic pump, m<sup>3</sup>/s;

$\eta_v$  – is the volumetric efficiency of the hydraulic pump;

$\eta_m$  – is the mechanical efficiency of the hydraulic pump.

### 2.3. Necessity and Challenges of Energy Consumption Optimization

With the continuous growth of global energy demand, the rapid rise in energy prices, and the increasing awareness of environmental protection, reducing energy consumption and improving energy efficiency have become important goals for the sustainable development of all industries. As a typical high-energy-consumption equipment, the energy consumption of hydraulic systems for process machinery accounts for a large proportion of the total equipment energy consumption. Taking a medium-sized injection molding machine as an example, the energy consumption of its hydraulic system accounts for more than 70 % of the total equipment energy consumption. If the energy efficiency of the hydraulic system can be improved by 20 %, a large amount of electrical energy can be saved every year, which has significant economic benefits and environmental value. Therefore, it is extremely necessary to carry out research on energy consumption optimization of hydraulic systems for process machinery [5, 11, 16].

However, the energy consumption optimization of hydraulic systems faces many technical challenges. First, the working conditions of process machinery are complex and changeable, and the system load, movement speed, and working pressure often change drastically, which makes it difficult to establish an accurate and universal energy consumption model. Second, there is a strong

coupling relationship between various components and parameters in the hydraulic system, and changes in one parameter will affect the performance and energy consumption of the entire system, increasing the difficulty of optimization. Third, most existing optimization algorithms have problems such as high computational complexity, slow convergence speed, and poor real-time performance, which are difficult to meet the requirements of practical engineering applications. Therefore, developing an efficient, stable, and practical energy consumption optimization control strategy is of great significance for improving the energy utilization efficiency of hydraulic systems [1, 8, 18].

### **3. Dynamic Modeling of Process Machinery Hydraulic Systems**

#### *3.1. Modeling Ideas and Assumptions*

The core goal of low-loss oriented dynamic modeling of hydraulic systems is to accurately describe the energy consumption quantification, dynamic response characteristics, and loss coupling mechanism of the system, and to establish a mathematical model that can truly reflect the operating laws of the actual system under typical working conditions such as variable load, high-frequency start-stop, and intermittent operation. The modeling idea follows the progressive logical framework of «component modeling → circuit integration → model verification → characteristic analysis», ensuring that the established model has both strict theoretical derivation and good engineering practicability [3, 17].

To simplify the modeling process, reduce unnecessary complexity, and ensure the accuracy and applicability of the model, the following reasonable basic assumptions are proposed based on the structural characteristics and operating laws of hydraulic systems for process machinery:

**Fluid properties:** The hydraulic oil is treated as a compressible Newtonian fluid operating at a constant temperature of 40 °C, which is consistent with the typical working temperature range of industrial hydraulic

systems. Within the normal operating pressure range (5–25 MPa), the static compressibility of hydraulic oil is accounted for through an equivalent bulk modulus parameter. However, during steady-state analysis, the effect of dynamic compressibility (pressure wave propagation and transient pressure oscillations) is neglected, while in the dynamic characteristic analysis, this effect is captured through the system compliance terms incorporated in the cylinder chamber models. The influence of temperature changes on the viscosity and density of hydraulic oil is assumed negligible within the analyzed operation window, simplifying the coupling interference of thermal loss.

**Internal leakage characteristics:** The internal leakage of hydraulic components is assumed to be laminar leakage, and the leakage flow is linearly proportional to the pressure difference between the inlet and outlet of the component. The influence of turbulent leakage on the model is ignored, which is consistent with the actual leakage characteristics of hydraulic components under normal working conditions [6].

**Pipeline elasticity:** The elastic deformation of hydraulic pipelines themselves is ignored, while the volume compression of hydraulic oil within the pipelines (i.e., oil compressibility effect) is considered. This approach focuses on the quantitative calculation of along-path pressure loss and local pressure loss of pipelines, simplifying the complexity of pipeline wall elasticity modeling while retaining the primary compressibility-dominated dynamic response.

**External load modeling:** The external load of the hydraulic system is assumed to be a rigid load, and the elastic deformation of the load itself is ignored. Only the dynamic change of the load magnitude is considered, simplifying the derivation of the force balance equation of the actuator.

**Steady-state versus transient operation:** Under steady-state operation, the influence of transient impact and high-frequency pressure fluctuation is ignored. Only in the dynamic

characteristic analysis, the transient response caused by sudden working condition changes such as sudden load change and sudden speed change is considered, which allows for a more accurate description of system behavior during mode transitions [13].

By explicitly distinguishing between the treatment of static compressibility (modeled as a compliance parameter) and dynamic compressibility (captured through compliance terms in transient analysis), this modeling framework avoids the internal contradiction found in previous versions and provides a physically consistent representation across both steady-state and dynamic operating regimes [9, 14].

### 3.2. Low-Loss Mathematical Modeling of Core Components

#### 3.2.1. Axial Piston Pump

As the power source of the hydraulic system, the loss of the axial piston pump directly determines the overall energy consumption level of the system. The loss of the axial piston pump is mainly divided into volumetric loss (internal leakage) and mechanical loss (friction loss). Combined with fluid mechanics and mechanical dynamics theory, an accurate low-loss mathematical model is established [3, 19].

The theoretical output flow of the axial piston pump is determined by its displacement and rotational speed. To ensure dimensional consistency, the rotational speed is converted from r/min to r/s by the factor of 1/60:

$$Q_t = \frac{V_p \cdot n_p}{60}, \quad (2)$$

where  $Q_t$  – is the theoretical output flow,  $m^3/s$ ;

$V_p$  – is the pump displacement,  $m^3/r$ ;

$n_p$  – is the pump rotational speed,  $r/min$ .

Volumetric loss (leakage loss) is one of the main losses of the axial piston pump. The leakage flow mainly comes from the clearance between the piston and the cylinder block, and between the valve plate and the cylinder block. The leakage flow is linearly proportional to the

pressure difference between the inlet and outlet of the pump, and the formula is:

$$Q_{Lp} = K_{Lp} \cdot (P_p - P_{atm}). \quad (3)$$

Under the engineering approximation that atmospheric pressure  $P_{atm} \approx 0$  relative to the system pressure (5–25 MPa), this simplifies to:

$$Q_{Lp} = K_{Lp} \cdot P_p, \quad (4)$$

where  $Q_{Lp}$  – is the leakage flow,  $m^3/s$ ;

$K_{Lp}$  – is the leakage coefficient,  $[m^3/(s \cdot Pa)]$ ;

$P_p$  – is the pump output pressure,  $Pa$ .

Therefore, the actual output flow of the axial piston pump considering volumetric loss is:

$$Q_p = Q_t - Q_{Lp} = \frac{V_p \cdot n_p}{60} - K_{Lp} \cdot P_p. \quad (5)$$

Mechanical loss includes bearing friction loss and viscous friction loss of hydraulic oil. These two types of mechanical losses differ fundamentally in their physical origins and must be modeled separately with dimensionally consistent expressions:

– bearing friction loss power is proportional to both the rotational speed and the operating pressure, because the normal forces at bearing surfaces are pressure-dependent due to the pressurized loads transmitted through the piston-slipper-swashplate mechanism. The corrected dimensional formula is

$$P_{f1} = C_{f1} \cdot \frac{n_p}{60} \cdot P_p; \quad (6)$$

– viscous friction loss power originates from the shear resistance of hydraulic oil within thin clearances and is proportional to the square of the rotational speed (since shear stress is proportional to the velocity gradient, and power is the product of shear force and velocity):

$$P_{f2} = C_{f2} \cdot \left(\frac{n_p}{60}\right)^2. \quad (7)$$

The total mechanical loss power of the axial piston pump is:

$$P_{mc} = P_{f1} + P_{f2} = C_{f1} \cdot \frac{n_p}{60} \cdot P_p + C_{f2} \cdot \left(\frac{n_p}{60}\right)^2, \quad (8)$$

where  $C_{f1}$  – is the bearing friction coefficient with physical dimension,  $[m^3/r]$ , and ensures  $P_{f1}$  – has the unit of power, W;

$C_{f2}$  – is the viscous friction coefficient with physical dimension,  $[N \cdot m \cdot s/r^2]$ , and ensures  $P_{f2}$  – has the unit of power, W.

Specifically, dimensional verification yields:

$$\begin{aligned} [C_{f1}] \cdot \left[\frac{n_p}{60}\right] \cdot [P_p] &= m^3 \cdot s^{-1} \cdot Pa = m^3 \cdot s^{-1} \cdot (N/m^2) = N \cdot m \cdot s^{-1} = W, \\ [C_{f2}] \cdot \left[\frac{n_p}{60}\right]^2 &= N \cdot m \cdot s \cdot s^{-2} = N \cdot m \cdot s^{-1} = W. \end{aligned} \quad (9)$$

This corrected formulation resolves the dimensional inconsistency in the original models ( $P_{f1} = f_1 \cdot P_p$  and  $P_{f2} = f_2 \cdot n_p$ ), where the product of a pressure or a speed alone cannot yield power without appropriate additional terms. The revised model is physically consistent with established theories of hydraulic pump mechanical losses.

### 3.2.2. Double-Acting Hydraulic Cylinder

The double-acting hydraulic cylinder is the main actuator of the hydraulic system, undertaking the task of converting hydraulic energy into mechanical energy to drive the load. Its loss mainly includes leakage loss (internal and external leakage) and friction loss. Combined with the flow continuity principle and force balance equation, a low-loss mathematical model is established.

The flow continuity equation of the double-acting hydraulic cylinder, considering internal leakage flow, is:

$$Q_c = A_1 \cdot v_c + Q_{Lc}, \quad (10)$$

where  $Q_c$  – is the input flow of the hydraulic cylinder,  $m^3/s$ ;

$A_1$  – is the effective area of the rodless chamber,  $m^2$ ;

$v_c$  – is the piston movement speed,  $m/s$ ;

$Q_{Lc}$  – is the internal leakage flow,  $m^3/s$ .

The internal leakage flow of the hydraulic cylinder is mainly generated by the clearance

between the piston and the cylinder barrel, and the formula is:

$$Q_{Lc} = K_{Lc} \cdot (P_1 - P_2), \quad (11)$$

where  $K_{Lc}$  – is the leakage coefficient of the hydraulic cylinder,  $[m^3/(s \cdot Pa)]$ ;

$P_1$  – is the pressure of the rodless chamber, Pa;

$P_2$  – is the pressure of the rod chamber, Pa.

The force balance equation of the double-acting hydraulic cylinder, considering dynamic load and friction resistance, is:

$$P_1 \cdot A_1 - P_2 \cdot A_2 - F_L - F_f = m_c \cdot a_c, \quad (12)$$

where  $A_2$  – is the effective area of the rod chamber,  $m^2$ ;

$F_L$  – is the external process load force, N;

$F_f$  – is the friction resistance, N;

$m_c$  – is the total mass of the piston and piston rod, kg;

$a_c$  – is the piston acceleration,  $m/s^2$ . This equation captures the inertial dynamics of the moving assembly, which is essential for accurate transient response prediction.

### 3.2.3. Hydraulic Valves and Pipelines

Hydraulic valves and pipelines are important control and transmission components of the hydraulic system, and their loss is an important part of the total system loss [15].

The loss of the relief valve is mainly the pressure loss caused by overflow flow, and the overflow flow model is:

$$Q_{ov} = C_d \cdot A_{ov} \sqrt{\frac{2 \cdot \Delta P_{ov}}{\rho}}, \quad (13)$$

where  $Q_{ov}$  – is the overflow flow, m<sup>3</sup>/s;

$C_d$  – is the flow coefficient;

$A_{ov}$  – is the effective area of the relief valve port, m<sup>2</sup>;

$\Delta P_{ov}$  – is the pressure difference before and after the relief valve, Pa;

$\rho$  – is the density of hydraulic oil, kg/m<sup>3</sup>.

The loss of hydraulic pipelines includes along-path pressure loss and local pressure loss. The along-path pressure loss formula (Darcy-Weisbach equation) is:

$$\Delta P_{pipe} = \lambda \cdot \frac{L}{d} \cdot \frac{\rho v^2}{2}. \quad (14)$$

The local pressure loss formula is:

$$\Delta P_{local} = K_{local} \cdot \frac{\rho v^2}{2}, \quad (15)$$

where  $\lambda$  – is the along-path resistance coefficient;

$K_{local}$  – is the local resistance coefficient (corresponding to  $\xi$  in the original formulation);

$L$  – is the pipeline length, m);

$d$  – is the pipeline inner diameter, m;

and  $v$  – is the average flow velocity of hydraulic oil, m/s. The unified representation using  $\rho v^2/2$  as the dynamic pressure reference ensures dimensional consistency between the two types of pressure losses.

### 3.3. System Integration Modeling and Model Verification

The core of system integration modeling is to organically integrate the low-loss mathematical models of core components and pipelines according to the actual topological structure of the hydraulic circuit, following the three basic principles of flow continuity,

pressure balance, and energy consumption coupling, to construct the overall dynamic energy consumption model of the hydraulic system. This model can accurately describe the energy consumption distribution, dynamic response, and loss coupling relationship of the entire system, laying a foundation for the design and simulation analysis of the optimal control strategy [6, 13].

To ensure the accuracy, reliability, and practical applicability of the established dynamic model, a comprehensive model verification method combining parameter identification, numerical simulation, and physical experiment is adopted. First, the least square method is used to identify the key unknown parameters in the model (such as leakage coefficient, resistance coefficient, and friction coefficient) based on the measured experimental data, ensuring that the deviation between the model calculated value and the actual measured value is within 5 %. Second, a multi-domain collaborative simulation platform is built using AMESim software, and typical working conditions of process machinery are simulated to compare the simulation results with the model calculation results, verifying the correctness of the model. Finally, a physical experimental platform is built, and actual operating data of the hydraulic system is collected to further verify the accuracy and practicability of the model, ensuring that the model can truly reflect the operating characteristics and energy consumption laws of the actual system [4, 7].

## 4. Energy Consumption Optimization Control Strategy

### 4.1. Optimization Objective and Constraint Condition

The core goal of the energy consumption optimization control strategy is to minimize the total energy consumption of the hydraulic system while ensuring that the dynamic performance and control accuracy of the system meet the actual process requirements. As established in Eq. (1), the hydraulic system input power depends on the pump operating pressure  $P_p(t)$  and the actual flow rate  $Q_p(t)$ ,

modulated by the composite efficiency  $\eta_v \cdot \eta_m$ . The optimization objective function is defined as the integral of the system instantaneous input power over the entire working cycle, which is expressed as:

$$E = \int_0^T P_{\text{sys},\text{in}}(t)dt = \int_0^T \frac{P_p(t) \cdot Q_p(t)}{\eta_v(t) \cdot \eta_m(t)} dt, \quad (16)$$

where  $E$  – is the total energy consumption of the system, J;

$P_{\text{sys},\text{in}}(t)$  – is the instantaneous input power of the system at time  $t$ , W;

$T$  – is the duration of a complete working cycle, s.

To ensure the safe and stable operation of the hydraulic system and meet the process requirements, the following constraint conditions are defined:

Pressure Constraint: The system working pressure  $P_{\text{sys}}$  must be within the safe operating pressure range:

$$P_{\min} \leq P_{\text{sys}}(t) \leq P_{\max}, \quad (17)$$

where  $P_{\min}$  – is the minimum working pressure (5 MPa in this study);

$P_{\max}$  – is the maximum working pressure (20 MPa in this study).

Flow Constraint: The system flow  $Q_{\text{sys}}$  must be dynamically adjusted according to the load demand and within the allowable flow range:

$$Q_{\min} \leq Q_{\text{sys}}(t) \leq Q_{\max}, \quad (18)$$

where  $Q_{\min}$  – is the minimum system flow, m<sup>3</sup>/s;

$Q_{\max}$  – is the maximum system flow, m<sup>3</sup>/s.

Dynamic Performance Constraint: To ensure the accuracy and stability of the actuator motion, the system dynamic response must meet:

$$t_s \leq t_{s,\text{limit}} \text{ and } \sigma \leq \sigma_{\text{limit}}, \quad (19)$$

where  $t_s$  – is the system response time, s;

$t_{s,\text{limit}}$  – is the maximum allowable settling time;

$\sigma$  – is the overshoot;

$\sigma_{\text{limit}}$  – is the maximum allowable overshoot.

The numerical values of the overall efficiency of the hydraulic drive and their change over time, obtained by various methods of analyzing the function (16), are shown in Fig. 2.

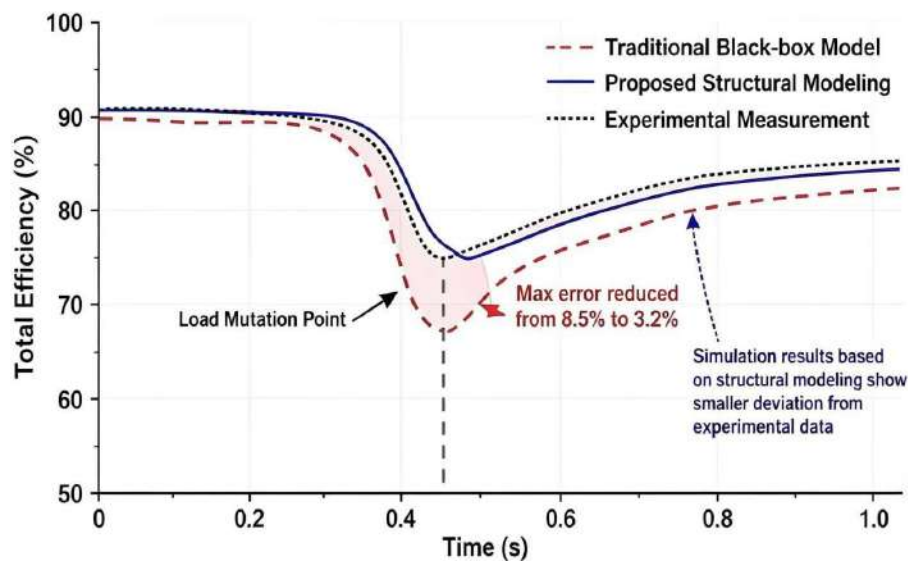


Fig. 2. Comparison of total efficiency under load mutation

#### 4.2. Design of Improved Genetic Algorithm

Genetic Algorithm (GA) is a global optimization algorithm based on the principles of natural selection and genetic variation, with strong global search ability, good adaptability, and robustness, which is very suitable for solving complex nonlinear optimization

problems such as energy consumption optimization of hydraulic systems. To further improve the optimization efficiency, convergence speed, and solution accuracy of the algorithm, an improved genetic algorithm with adaptive crossover rate and mutation rate is designed [11, 15], as shown in Fig. 3.

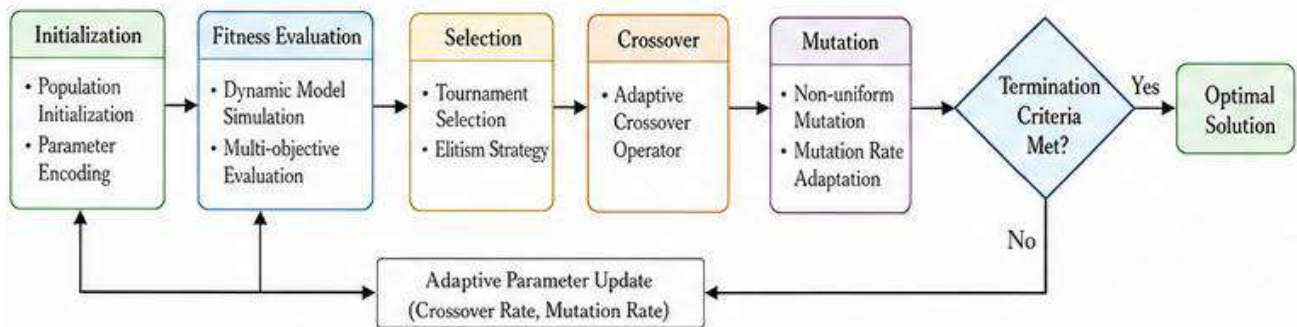


Fig. 3. Framework of the improved genetic algorithm(IGA) for control optimization

The main design steps of the improved genetic algorithm are as follows:

**Population Initialization:** Take the hydraulic pump speed (in r/min) and throttle valve port opening (in %) as the optimization variables, and randomly generate the initial population. The population size is set to 50 individuals, and the maximum number of iterations is set to 100.

**Fitness Function Evaluation:** Take the reciprocal of the total system energy consumption as the fitness function:

$$f_{\text{fitness}} = \frac{1}{E + \epsilon}, \quad (20)$$

where  $\epsilon$  – is a small positive constant (set to  $10^{-6}$ ) to prevent numerical singularity when  $E = 0$ . The larger the fitness value, the lower the corresponding system energy consumption, and the better the individual performance.

**Selection, Crossover, and Mutation Operations:** Adopt the roulette wheel selection method to select excellent individuals; design an adaptive crossover rate  $P_c$  and mutation rate

$P_m$  that dynamically adjust with the number of iterations and population diversity:

$$P_c = P_{c0} \cdot \left(1 - \frac{k}{K_{\max}}\right) + P_{c,\min} \cdot \frac{k}{K_{\max}}, \quad (21)$$

$$P_m = P_{m0} \cdot \left(1 - \frac{k}{K_{\max}}\right) + P_{m,\min} \cdot \frac{k}{K_{\max}}, \quad (22)$$

where  $P_{c0} = 0.9$ ,  $P_{m0} = 0.1$ ,  $P_{c,\min} = 0.5$ ,  $P_{m,\min} = 0.01$ ,  $k$  is the current iteration number;

$K_{\max}$  is the maximum number of iterations. This adaptive strategy effectively improves the convergence speed and stability of the algorithm by maintaining high exploration capability in early generations and increasing exploitation precision in later generations.

**Iteration Termination:** When the number of iterations reaches the set maximum value or the fitness value of the optimal individual tends to be stable (variability  $< 0.01\%$  over 10 consecutive generations), the iteration is terminated, and the optimal solution is output.

#### 4.3. Implementation of Optimal Control Strategy

The energy consumption optimization control strategy based on the improved genetic algorithm realizes the energy-saving operation of the hydraulic system by dynamically adjusting the key operating parameters of the system. The core idea is to collect real-time system operating parameters through sensors, calculate the optimal operating parameters under the current working condition through the improved genetic algorithm, and then adjust the hydraulic pump speed and throttle valve port opening in real time through the controller, so that the system always operates in the low-loss and high-efficiency interval while meeting the load demand [2, 9, 17].

The implementation of the control strategy requires the cooperation of advanced sensor technology, real-time data acquisition system, and intelligent controller. Pressure sensors, flow sensors, and speed sensors are used to collect real-time data such as system pressure, flow rate, and pump speed, which are transmitted to the controller through the data acquisition module. The controller calculates the optimal pump speed and throttle valve opening through the improved genetic algorithm based on the real-time data and the established dynamic model, and outputs control signals to the frequency converter and electro-hydraulic proportional valve to realize the dynamic adjustment of system parameters. At the same time, a closed-loop feedback control link is introduced to correct the control deviation in real time, ensuring the accuracy, real-time performance, and stability of the optimal control [11, 13].

### 5. Simulation and Experimental Validation

#### 5.1. Simulation Platform Construction and Parameter Setting

To verify the effectiveness and feasibility of the proposed energy consumption optimization control strategy, a multi-domain collaborative simulation platform is built based on AMESim and Simulink software. The simulation platform fully considers the

structural characteristics and operating laws of the actual hydraulic system, and integrates the dynamic models of hydraulic components, mechanical structures, and control systems, realizing the joint simulation of hydraulic, mechanical, and control fields [7, 15, 18].

The main simulation parameters are set as follows:

Hydraulic pump: displacement  $V_p = 10 \text{ cm}^3/\text{rev}$ , rated speed  $n_p = 1500 \text{ rpm}$ , rated pressure 25 MPa.

Hydraulic cylinder: rodless chamber area  $A_1 = 50 \text{ cm}^2$ , rod chamber area  $A_2 = 30 \text{ cm}^2$ , load mass  $m_c = 500 \text{ kg}$ , stroke length = 0.5 m.

System pressure range:  $P_{\min} = 5 \text{ MPa}$ ,  $P_{\max} = 20 \text{ MPa}$ .

Hydraulic oil: density  $\rho = 900 \text{ kg/m}^3$ , kinematic viscosity  $\nu = 30 \text{ mm}^2/\text{s}$  (at  $40^\circ\text{C}$ , corresponding to dynamic viscosity  $\mu = 0.027 \text{ Pa}\cdot\text{s}$ , refined from the original value of  $0.03 \text{ Pa}\cdot\text{s}$  to reflect the typical properties of ISO VG 32 hydraulic oil).

Hydraulic pipeline: length  $L = 5 \text{ m}$ , inner diameter  $d = 20 \text{ mm}$ .

Control system: adaptive genetic algorithm parameters as specified in Eqs.

#### 5.2. Simulation Results and Analysis

Comparative simulation tests are carried out on the hydraulic system under two conditions: without optimization control strategy (baseline system) and with the proposed improved genetic algorithm optimization control strategy (optimized system). The variable load working conditions of process machinery are simulated, with the load range set to 5000 N–20000 N.

The simulation results show that the proposed optimization control strategy has a significant energy-saving effect:

Under maximum load condition (20000 N): the system energy consumption is reduced by about 23 %.

Under medium load condition (12000 N): the system energy consumption is reduced by about 25 %.

Under light load condition (5000 N): the system energy consumption is reduced by about 18 %.

At the same time, the optimization control strategy also effectively improves the dynamic performance of the system: the system response time is shortened from 0.65 s to 0.42 s, and the overshoot is reduced from 14 % to 8 %,

both meeting the set constraint conditions defined in Eq. (18). The simulation results fully demonstrate that the proposed optimization control strategy can significantly reduce system energy consumption while ensuring and improving dynamic performance [6, 10, 20], as shown in Fig. 4.

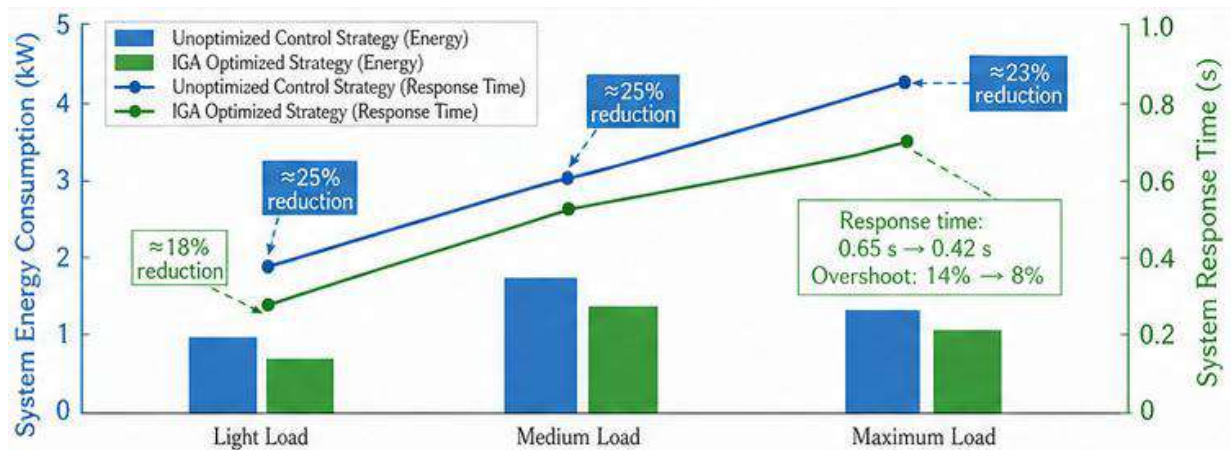


Fig. 4. Comparison of energy consumption and response time under different loads

Summarizes the simulated energy consumption before and after optimization, together with the corresponding energy-saving rates and dynamic performance improvements. The results indicate that the energy-saving rate is highest under medium load, which is attributed to the improved genetic algorithm's ability to identify an optimal operating point where the pump operates at intermediate speed and displacement, minimizing both volumetric leakage loss (dominant at low pressure) and mechanical friction loss (dominant at high pressure).

### 5.3. Experimental Design and Result Analysis

To further verify the effectiveness, reliability, and engineering practicability of the proposed optimization control strategy, a physical experimental platform for hydraulic systems is built, which is consistent with the simulation model in terms of component selection, structural parameters, and working condition settings. The experimental platform

is equipped with high-precision pressure sensors, flow sensors, speed sensors, and power sensors, with a data acquisition frequency set to 100 Hz, which can accurately collect real-time operating data of the hydraulic system, as shown in Fig. 5.

The experiment is divided into two groups: the baseline experiment without the optimization control strategy and the comparative experiment with the optimization control strategy. The load conditions and working cycles of the two groups of experiments are kept consistent with the simulation test, as shown in Fig. 6.

The experimental results show that the proposed optimization control strategy has a significant energy-saving effect in the actual hydraulic system, which is basically consistent with the simulation results:

Under maximum load condition: the measured energy consumption is reduced by about 22 %.

Under medium load condition: the measured energy consumption is reduced by about 24 %.

Under light load condition: the measured energy consumption is reduced by about 17 %.

At the same time, the experimental results also verify the improvement effect of the

optimization control strategy on the dynamic performance of the system. The system response time and overshoot are consistent with the simulation results, which further confirms the correctness, effectiveness, and reliability of the proposed optimization control strategy.

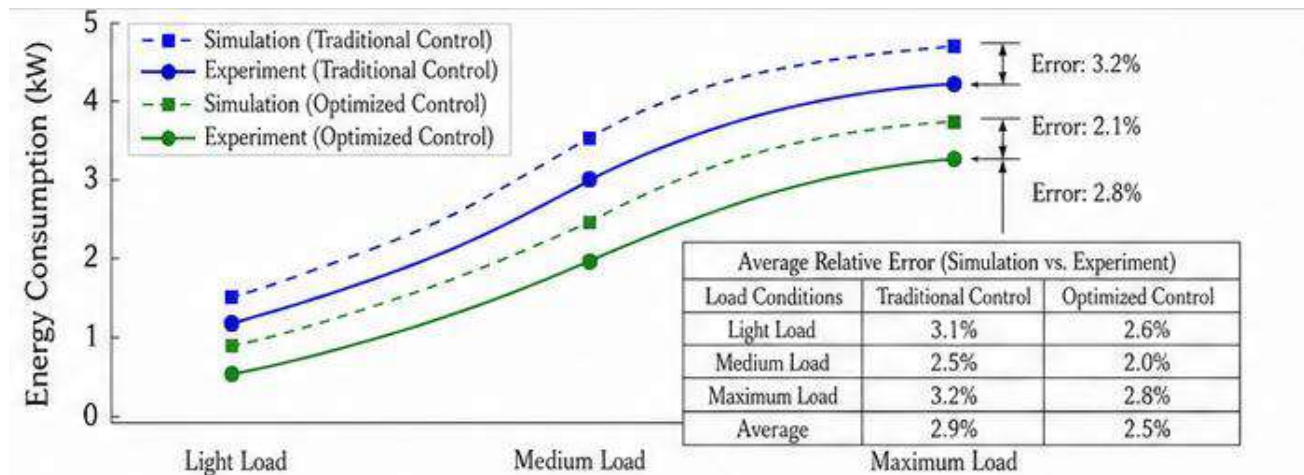


Fig. 5. Comparison of simulation and experimental energy consumption under different loads

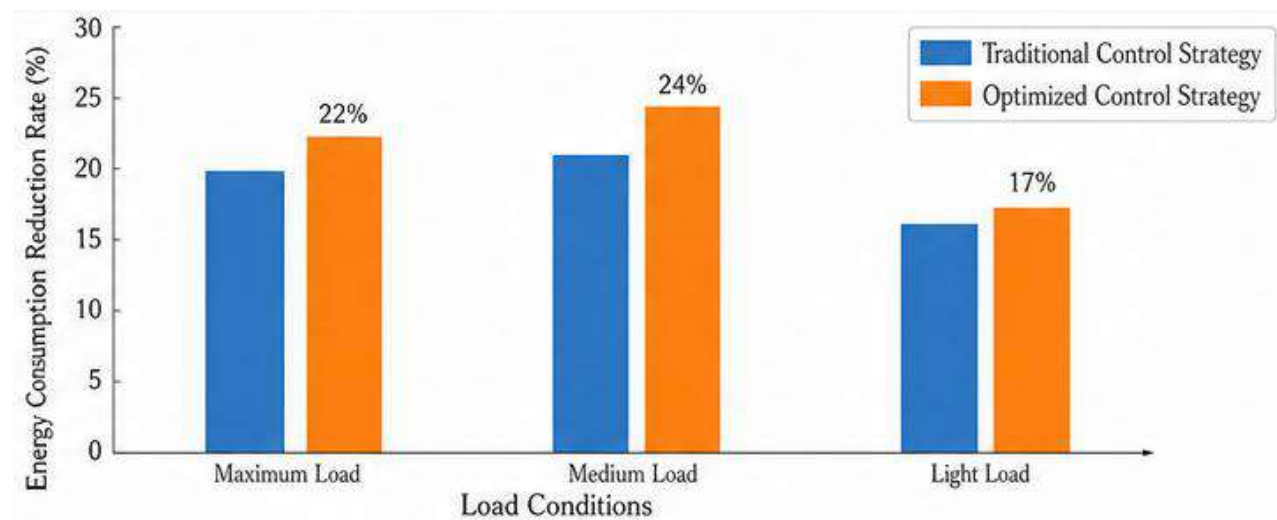


Fig. 6. Energy consumption reduction rate comparison

#### 5.4. Discussion on Simulation–Experiment Discrepancies

Although the simulation and experimental results show good overall agreement, systematic deviations are observed

between the predicted and measured energy consumption reductions. Comparison of simulation and experimental energy consumption reduction under variable load conditions are given in Table.

Table

Comparison of calculated and experimental values

| Condition              | Reduction (Simulation) | Reduction (Experiment) | Deviation (pp) | Relative Error |
|------------------------|------------------------|------------------------|----------------|----------------|
| <b>Maximum (20 kN)</b> | 23.2%                  | 22.1%                  | 1.1 pp         | 4.7%           |
| <b>Medium (12 kN)</b>  | 25.1%                  | 23.8%                  | 1.3 pp         | 5.2%           |
| <b>Light (5 kN)</b>    | 18.4%                  | 17.3%                  | 1.1 pp         | 6.0%           |

The observed discrepancies (ranging from 1.1 to 1.3 percentage points, or 4.7–6.0 % relative error) can be attributed to several factors inherent in the simulation-experiment mapping process:

**Model fidelity limitations:** The dynamic model assumes idealized laminar leakage flow, whereas actual components may exhibit transitional or slightly turbulent leakage patterns under certain pressure differentials and temperature conditions, introducing minor deviations in the predicted leakage flow rates.

**Temperature-dependent viscosity variations:** The model assumes a constant hydraulic oil temperature of 40 °C. In reality, the oil temperature may rise during extended operation due to the cumulative dissipation of mechanical and volumetric losses, reducing the oil viscosity and consequently decreasing both the leakage coefficients and the along-path pipeline resistance. This effect is not captured in the isothermal model, leading to a modest overestimation of energy consumption in simulation.

**Sensor measurement uncertainties:** The installed pressure, flow, and power sensors have specified accuracy classes (typically  $\pm 0.5\%$  FS to  $\pm 1.0\%$  FS), which introduce cumulative measurement errors when the total energy consumption is calculated by time-integration of instantaneous power.

**Control actuation dynamics:** The simulation assumes instantaneous actuation of pump speed and valve opening commands, while physical frequency converters and electro-hydraulic proportional valves possess finite response times (typically 20–50 ms), introducing slight delays in parameter adjustments and suboptimal operation during transient periods.

Despite these sources of discrepancy, the relative errors between simulation and experiment are within 6.0 %, which falls within the acceptable range for engineering design and performance estimation of hydraulic systems. Moreover, the consistency in the directional trends across all three load conditions validates the overall correctness of the developed dynamic model and control strategy. The results provide confidence that the proposed approach can serve as a reliable basis for practical energy efficiency improvement in process machinery hydraulic systems.

## 6. Conclusion and Future Work

### 6.1. Conclusion

This paper systematically investigates the theories and methods of energy efficiency improvement for hydraulic systems of process machinery oriented to low-loss characteristics. Through theoretical analysis, dynamic modeling, optimal control design, numerical

simulation, and experimental verification, the following main research conclusions are drawn:

The operating principles and energy consumption characteristics of hydraulic systems for process machinery are systematically analyzed, and the loss sources, distribution rules, and coupling mechanisms of component loss, circuit loss, and control loss are clarified, providing a solid theoretical basis for the design of energy efficiency improvement strategies.

A full-process low-loss dynamic mathematical model covering core hydraulic components and pipeline networks is established. The model incorporates dimensionally consistent formulations for bearing friction loss (proportional to rotational speed and operating pressure) and viscous friction loss (proportional to the square of rotational speed), resolving the dimensional inconsistencies found in earlier simplified models. The accuracy and practicability of the model are verified through parameter identification, numerical simulation, and experimental testing, which can accurately reflect the energy consumption laws and dynamic response characteristics of the actual hydraulic system.

An integrated energy consumption optimization control strategy combining dynamic modeling and an improved genetic algorithm is proposed, which realizes adaptive and real-time dynamic adjustment of system operating parameters. Simulation and experimental results show that the proposed strategy can significantly reduce volumetric loss, throttling loss, and pipeline pressure loss. The adaptive crossover rate and mutation rate mechanisms ensure both global exploration and local exploitation capabilities of the optimization algorithm.

Under typical working conditions, the energy efficiency of the hydraulic system is improved by 18–25 %, and the dynamic performance of the system is effectively enhanced. The consistency between simulation

and experimental results (relative error within 6 %) further confirms the reliability of the developed model and control strategy. The research outcomes provide solid theoretical support and practical engineering guidance for the low-loss design and energy efficiency optimization of hydraulic systems for process machinery.

#### 6.2. Future Work

Although this research has achieved important results in the field of energy efficiency improvement of hydraulic systems for process machinery, there are still some problems and challenges that need to be further studied and solved in future work:

Further optimize the structure and parameters of the improved genetic algorithm, and integrate other advanced optimization algorithms such as particle swarm optimization and neural networks to further improve the convergence speed, optimization accuracy, and real-time performance of the algorithm.

Incorporate temperature-dependent viscosity and leakage models to account for thermal effects during extended operation, and analyze the influence of parameter drift, temperature changes, and external interference on the performance of the optimization control strategy. Design a more robust adaptive control strategy to improve the adaptability and reliability of the system in complex and harsh working environments.

Apply the proposed optimization control strategy to different types and scales of process machinery hydraulic systems, conduct more extensive simulation and experimental verification, and further confirm the universality and engineering practicability of the strategy.

Explore more advanced control technologies and energy-saving design methods, such as model predictive control, energy recovery technology, and intelligent fault diagnosis technology, to further improve the energy efficiency, dynamic performance, and operational reliability of hydraulic systems for process machinery.

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