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Optimization of mechanical hydraulic system load adaptation based on fuzzy control algorithm

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Abstract Aiming at the problems of insufficient control accuracy and low energy efficiency caused by load fluctuations in mechanical hydraulic systems under complex working conditions, this paper uses fuzzy control algorithms as a research tool to explore the optimization path of load adaptation in hydraulic systems. After completing the modeling of load-sensitive valves, the principle of implied parallelism is borrowed to evaluate the adaptability capability of individual components in the mechanical hydraulic system. At the same time, genetic operators are defined to realize the mathematical theory of genetic algorithms in mechanical hydraulic systems as well as optimization. A new load control algorithm for mechanical hydraulic system is proposed by establishing a fuzzy PID control algorithm and integrating the genetic algorithm. The proposed algorithm is used to design a fuzzy adaptive PID controller for the hydraulic cylinder of the loader rocker arm, and simulation experiments of the loader load hydraulic system are carried out. After the flow and power parameters are stabilized, the flow rate of the multi-way valve port is stable at 73.5539L/min and the power saving is stable at 0.8691kW, which shows the high effectiveness of the proposed algorithm in the optimization of mechanical hydraulic system load.

Index Terms mechanical hydraulic system, fuzzy PID control algorithm, genetic algorithm, load adaptation optimization

I. Introduction

Mechanical hydraulic system refers to the engineering application-based host structure that is composed of three types of components: hydraulic control device, hydraulic pump device, and hydraulic tank [1]. With its unique advantages, this system is widely used in various industries such as engineering machinery, mining machinery, ship aviation, metallurgical machinery, agricultural and forestry machinery, etc. [2], [3]. However, in the actual construction process, although the application range of mechanical hydraulic system is extremely wide, but with the increase of workload, individual component organizations will appear heat, breakage and other phenomena, which will cause the application of energy consumption increased [4]-[6]. The fundamental reason for the inefficient operation of mechanical hydraulic systems lies in the complex process of energy transfer and conversion [7].

The energy transfer and conversion process of mechanical hydraulic system can be briefly described as the mechanical energy input from the prime mover is converted into hydraulic energy by the hydraulic pump, and then filtered, controlled and distributed to the hydraulic actuating element, which is again converted into mechanical energy, and ultimately drives the load to do work [8]-[11]. The energy loss in the transmission process is caused by friction in the mechanical transmission part and mismatch between load variation and hydraulic source output [12], [13]. Therefore, it is of great significance to carry out the research on load fit optimization of mechanical hydraulic system to improve the manufacturing technology level of equipment, promote the green development of the industry, and build a resource-saving society [14]-[16].

Based on the working principle of load-sensitive valve, this paper explains its mathematical modeling process and the calculation method of each module. Secondly, genetic algorithm is introduced, and the working principle of implied parallelism is discussed in detail, which is used as the theoretical framework of genetic algorithm applied to mechanical hydraulic system. At the same time, we design the method of evaluating the adaptability of individual components in the mechanical hydraulic system and the definition formula of the genetic operator, so as to complete the mathematical theoretical research and optimization of the genetic algorithm in the mechanical hydraulic system. Again based on the mechanical hydraulic system to build a fuzzy PID control algorithm, synthesize the genetic algorithm to form the control algorithm of the mechanical hydraulic system load. Finally, the loader rocker hydraulic cylinder is selected as the experimental object, its load characteristics are analyzed, and the algorithm in this paper is used to assist in the design of the fuzzy adaptive PID controller and to carry out the analysis of energy-saving working conditions.

II. Mechanical Hydraulic System Load Control Algorithm

II. A. Load-sensitive valve modeling

In the working principle of the load sensitive valve, the pump outlet pressure P_P and the load feedback pressure P_F to do the difference, the pressure difference ΔP and the load sensitive valve spring preset force P_S to do the comparison, so as to determine the working position of the load sensitive valve. When the load becomes large, the pressure difference ΔP becomes small, the load-sensitive valve works in the right position, the variable cylinder rodless out of the oil through the pressure cut-off valve and the load-sensitive valve into the tank, the variable cylinder rod retracted, the pump displacement becomes larger to adapt to the larger load. When the load becomes small, the pressure difference ΔP becomes large, the load-sensitive valve works in the left position, the oil from the pump outlet enters the rodless chamber of the variable cylinder through the load-sensitive valve and the pressure cut-off valve, the piston rod of the variable cylinder extends, and the displacement of the pump decreases, so as to achieve the purpose of energy saving.

The two-position three-way valve model in Matlab/Simhydraulics is shown in Figure 1.

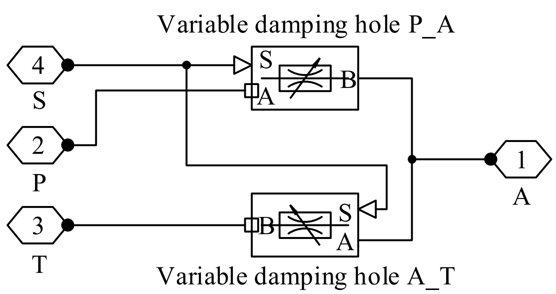


Figure 1: Two position three-way valve model

As can be seen in Figure 1, the two-position three-way valve model has two variable damping holes consisting of four ports, of which the A port is connected to the actuator, the P port is connected to the inlet port, the T port is connected to the return port, and the S port is connected to the valve's control signal, which is used to control the position of the spool.

The opening amount of variable damping orifice P_A is obtained from equation (1):

$$h_{PA} = h_{PA0} + h \tag{1}$$

where h_{PA} - the opening between the variable damping hole P port and A port, mm, h_{PA0} - initial opening between the variable damping hole P port and A port, mm, h --Variable damping hole P_A spool displacement, mm.

The opening amount of variable damping hole A_T is obtained from equation (2):

$$h_{AT} = h_{AT0} - h \tag{2}$$

where h_{AT} - the opening between the variable damping hole A port and T port, mm, h_{AT0} - the initial opening between the variable damping hole A mouth and T mouth, mm.

The variable damping orifice spool displacement in Equation (1) and (2) is determined by the double-acting valve actuator, and the principle of double-acting valve actuator in Matlab/Simhydraulics is shown in Figure 2.v

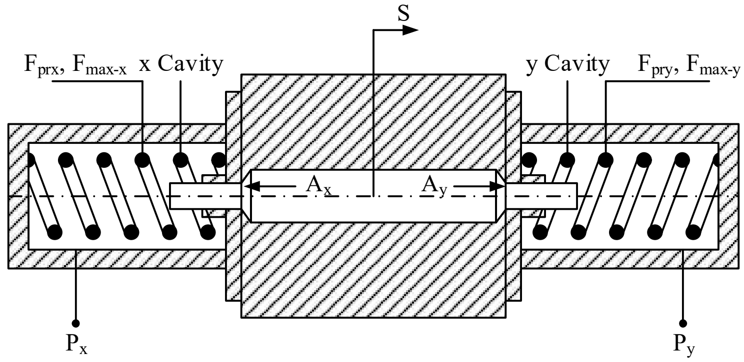


Figure 2: Double acting valve actuator schematic

The amount of double-acting valve actuator spool y cavity movement is equation (3):

$$s_{d-y}=\begin{cases}0 & F_d \le F_{d-pry} \\ L_{d-y}(F_d-F_{d-pry}) & F_{pry}<F_d<F_{d-\max-y} \\ s_{\max-y} & F_d \ge F_{d-\max-y}\end{cases} \tag{3}$$

where s_{d-y} -displacement of double-acting valve actuator spool to y cavity, mm, F_d -Force on the spool of double-acting valve actuator, N, L_{d-y} - spring flexibility of double-acting valve actuator y cavity, mm/N, $s_{\max-y}$ - Maximum spring displacement of double acting valve actuator y cavity, mm, $F_{d-\max-y}$ - Maximum force of double acting valve actuator y cavity spring, N, F_{d-pry} - preset force of spring in double acting valve actuator y cavity, N.

Double-acting valve actuator spool x cavity movement amount for the formula (4):

$$s_{d-x}=\begin{cases}0 & |F_d| \le F_{d-prx} \\ -L_{d-x}\cdot\left(|F_d|-F_{d-prx}\right) & F_{d-prx}<|F_d|<F_{d-\max-x} \\ -s_{\max-x} & |F_d| \ge F_{d-\max-x}\end{cases} \tag{4}$$

where s_{d-x} - displacement of double-acting valve actuator spool to x cavity, mm, L_{d-x} - spring flexibility of double-acting valve actuator x cavity, mm/N, $s_{\max-x}$ - Maximum spring displacement of double acting valve actuator x cavity, mm, $F_{d-\max-x}$ - Maximum force of double acting valve actuator x cavity spring, N, F_{d-prx} - preset force of the spring in the cavity of double-acting valve actuator x, N.

The force on the spool of the double-acting valve actuator in Eqs. (3) and (4) is obtained from Eq. (5):

$$F_d = P_x A_x - P_y A_y \tag{5}$$

where A_x - area of double-acting valve actuator x cavity, m^2, P_x -Pressure of double-acting valve actuator x cavity, Pa, A_y - Area of double-acting valve actuator y cavity, m^2, P_y - Double acting valve actuator y cavity pressure, Pa.

Double-acting valve actuator y cavity spring flexibility for the formula (6):

$$L_{d-y}=\frac{s_{\max-y}}{F_{d-\max-y}-F_{d-pry}} \tag{6}$$

Double acting valve actuator X -cavity spring flexibility is Eq. (7):

$$L_{d-x}=\frac{s_{\max-x}}{F_{d-\max-x}-F_{d-prx}} \tag{7}$$

II. B.Mathematical theory of genetic algorithms in mechanical hydraulic systems

Genetic algorithm optimization in mechanical hydraulic system contains 3 aspects of the principle of implied parallelism, system individual fitness, and genetic operator, and the detailed optimization process is as follows.

II. B. 1) Implicit parallelism principle

In genetic algorithms, each generation of child nodes deals with an equal number of informative individuals, however, since one or more genetic modes are implicit in each individual coding string, the whole application of the algorithm essentially deals with more genetic modes at the same time. If we take the binary coding string of a mechanical hydraulic system as an example, a genetic coding string of length m hides at least 2^m genetic modes, and it is possible for such a data population of size X to contain $2^m - X \cdot 2^m$ different genetic modes at the same time. With the implementation of the evolutionary process of data species, these original genetic patterns with too long defined lengths are directly destroyed, while other patterns with relatively short defined lengths can survive. If the estimation is made according to the exponential growth rule that remains constant in every generation of the data population, the original data is also the value of the number of genetic patterns that can be accurately processed by the genetic algorithm of the mechanical-hydraulic system. Provide that c is a fixed value of genetic substring in the mechanical hydraulic system, whose starting position has a total of $(m - m_c + 1)$, and in the case that the length of c value meets the condition of the definition of the genetic pattern of $(m_c - 1)$, then at least one of the values of the data c is equal to the results of the value of 0 or 1. In this case, the total number of genetic patterns implied in all X individuals of the genetic population is the value of the formula (8):

$$Z = X \cdot (m - m_c + 1) \cdot 2^{m_c - 1} \tag{8}$$

II. B. 2) Evaluation of Individual System Adaptation

The genetic algorithm evaluates the superiority and inferiority class conditions of each participating component through the individual fitness index, so as to determine the size of the genetic probability of matching with it. Since the fitness conditions applied to the parent genetic capital can be directly inherited to the offspring capital, the specific numerical level of individuals with genetic characteristics among the offspring individuals can be calculated by the fitness probability coefficients, i.e., the standardized individual fitness function of the system can consider the genetic characteristics of the parent target as the individual fitness value, and while accurately calculating this probability coefficient, the fitness numerical indexes of the other offspring individuals are all normalized to zero. Define V_1 as a standardized parent genetic node in the mechanical-hydraulic system, and v_1 and v_2 as two non-overlapping offspring genetic nodes in the mechanical-hydraulic system, and the numerical influence relationship between the two offspring nodes shown in Eq. (9) is always established without considering the intergenerational genetic behavior.

$$\begin{cases} v_1 \geq 0 \\ v_2 \geq 0 \\ v_1 > v_2 \end{cases} \tag{9}$$

Let N_Z^1 be the adaptation evaluation coefficient corresponding to node v_1, and N_Z^2 be the adaptation evaluation coefficient corresponding to node v_2, and associating Eq. (9), the evaluation criterion of individual adaptation in the mechanical hydraulic system can be defined as Eq. (10):

$$N = \begin{cases} N_Z^1 = v_1 Z + (1 - \chi_1)\overline{v} \\ N_Z^2 = v_2 Z + (1 - \chi_2)\overline{v} \end{cases} \tag{10}$$

where, χ_1 is the genetic condition corresponding to the characterization of v_1. χ_2 is the genetic condition corresponding to the characterization of v_2. $\overline{v}$ is the average of node v_1 and node v_2.

II. B. 3) Definition of genetic operators

In the rules of inheritance and evolution of organisms in nature, species with higher adaptability to the living environment will be more likely to pass on expressive traits to offspring individuals. Species that are less well adapted to their environment will be less likely to pass on traits to their offspring. Mimicking the above process, the genetic algorithm will control the survival of the offspring in the population by using an “operator”: it will select the traits of the individuals with a high level of fitness and pass them on to the next generation with a high operator coefficient. Individual traits with a low level of fitness are directly suppressed. In the mechanical hydraulic control system, the selection operation of the genetic algorithm is to select the individual objects with the strongest application characteristics from the parent individuals and extract their performance characteristics, which are finally directly inherited to the offspring individuals. If $\mu_{\min}$ represents the smallest amount of individual adaptation characteristics of the parent, $\mu_{\max}$ represents the largest amount of individual adaptation characteristics of the parent, and $\chi_0 \to \chi_1$ represents the actual genetic mapping relationship from the selected parent node to the node of its offspring, and the genetic operator in the mechanical hydraulic control system can be defined in equation (10) as equation (11):

$$B(\chi_0 \to \chi_1) = \frac{\sum_{\mu_{\min}}^{\mu_{\max}} \left[(\chi_1 - \chi_0)^s \right]^{\frac{1}{s}}}{\Delta A \cdot N^2} \tag{11}$$

where, s is the genetic condition of performance characteristics from parent node to child node. ΔA is the amount of genetic change in node characteristics per unit time.

II. C.Fuzzy PID control algorithm construction

In the field of engineering construction, many engineering vehicle machines with special operational requirements, such as pavement paver, asphalt sprinkler, road milling machine, etc., which need to ensure stable driving. However, due to the various uncertainties at the construction site and the random and systematic fluctuations of the load, the traditional control methods have been difficult to meet the demand for control of stable driving of engineering vehicles. In response to this contradiction, previous generations have proposed a variety of solutions, such as the use of PID control, although widely used, but in some cases there are defects that can not be ignored, the control effect by the quality of the mathematical model of the components of the degree of influence of the quality of the pump-controlled motor speed control system in general has a delay, the role of the mechanism is complex and complex and variable conditions in the work process, so it is difficult to use mathematical modeling methods will be Its core workflow abstract extraction, resulting in PID control algorithms can not quickly and accurately achieve the control requirements.

Emerging intelligent control technology provides new ideas for solving this kind of problems, especially the fuzzy control method, which has excellent robustness and stability, and has a better effect on the control of nonlinear complex systems with complex intrinsic laws or nonlinear complex systems that cannot be described by mathematical language. However, it can not eliminate the deviation of the system steady-state output value from the target value, based on this problem, this paper investigates the integration of PID control algorithm and fuzzy control method to optimize the driving stability of engineering vehicles, and constructs the fuzzy PID control algorithm.In the PID control, the difference between the desired value and the feedback value is set to be $e(t)$, and the expression of the control quantity after discretization to adapt to the computer computation $u(t)$ is as shown in Eq. (12) is shown.

$$u(k) = K_P e(k) + K_I \sum_{j=0}^{j=k} e(j) + K_D [e(k) - e(k-1)] \tag{12}$$

where k is the sampling sequence number. K_P is the scale factor. K_I is the integration coefficient, calculated as equation (13):

$$K_I = \frac{T}{T_I} \tag{13}$$

K_D is the differentiation factor, calculated as equation (14):

$$K_D = \frac{T_D}{T} \tag{14}$$

T is the sampling period, T_I and T_D are the integral time constant and differential time constant respectively.

Eq. (12) is also known as positional PID because the output of the controller corresponds to the actual position of the controlled object, and this feature leads to the fact that once the controller makes a mistake, it is easy to cause mis-operation and even produce engineering accidents. To avoid this drawback, the algorithm needs to be improved. To further simplify the calculation process and improve the efficiency of sequence calculation, Equation (12) can also be rewritten as Equation (15):

$$u(k-1)=K_P e(k-1)+K_I\sum_{j=0}^{j=k-1}e(j)+K_D[e(k-1)-e(k-2)] \tag{15}$$

Equation (15) is then subtracted from equation (12) to obtain the incremental PID as shown in equation (16):

$$\begin{aligned}\Delta u(k)=K_P[e(k)-e(k-1)]+K_I e(k)\\ +K_D[e(k)-2e(k-1)+e(k-2)]\end{aligned} \tag{16}$$

Thus the output expression (17) for the incremental PID controller can also be derived:

$$u(k)=u(k-1)+\Delta u(k) \tag{17}$$

Comparison of incremental and positional PID, it can be found that there is a big difference between the two in the form of the former, there is no integral operation, and thus does not cause the integral saturation, and there is no need to limit the integral, and the output of the calculated limit can be. Step back and analyze, even if the calculator runs wrong, because the controller output corresponds to the amount of change in the position of the object, the impact caused by misoperation will be smaller than the positional PID. In order to eliminate the steady state error generated by the control system operation, most cases also need to introduce the integral operation, because the controller output will also be affected by the PWM duty cycle polarity, so the need for PID controller output to do the limiting process, the upper and lower limit values of the controller output were set to $u_{\max}$ and $u_{\min}$, then add the limiting operation after the controller output value as equation (18):

$$u(k)=\begin{cases}u_{\max} & u(k)\geq u_{\max}\\ u_{\min} & u(k)\leq u_{\min}\\ u(k) & u_{\min}<u(k)<u_{\max}\end{cases} \tag{18}$$

Fuzzy control method is a method to control the object by summarizing and converting the experience of experts into a computer program, i.e., simulating the way of human thinking. Its effect on the control of nonlinear complex systems with complex intrinsic laws or nonlinear complex systems that cannot be described by mathematical language is better, and the working principle is shown in Fig. 3.

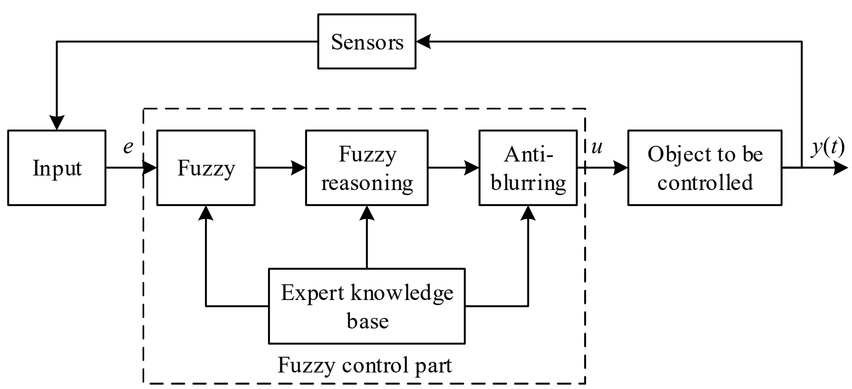


Figure 3: The working principle of fuzzy control method

As can be seen from Fig. 3, the core idea of the fuzzy control method is to formulate fuzzy control commands, which is based on the mature experience of relevant experts in the industry, firstly, the fuzzy relationship R is solved through the corresponding calculation rules, and at the same time, the measured value of the current calculation index is obtained through the sensors, and the deviation value e is calculated according to the two. Then use R to convert e to fuzzy quantity E, combine the operation rules to reason about E, so as to find out the control quantity as equation (19):

$$U=E\circ R \tag{19}$$

Finally, U is transformed into a precise control quantity u using the inverse fuzzy rule, which is sent to the controlled object to perform the corresponding control operation. Fuzzy control also has the disadvantages of low control accuracy and limited adaptive ability, so the incremental PID algorithm is fused with the fuzzy control method, and the fused algorithm flow is shown in Fig. 4.

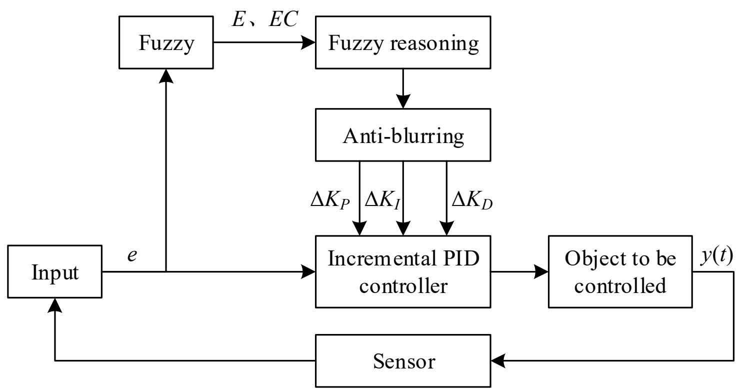


Figure 4: The calculation process of the fuzzy PID algorithm

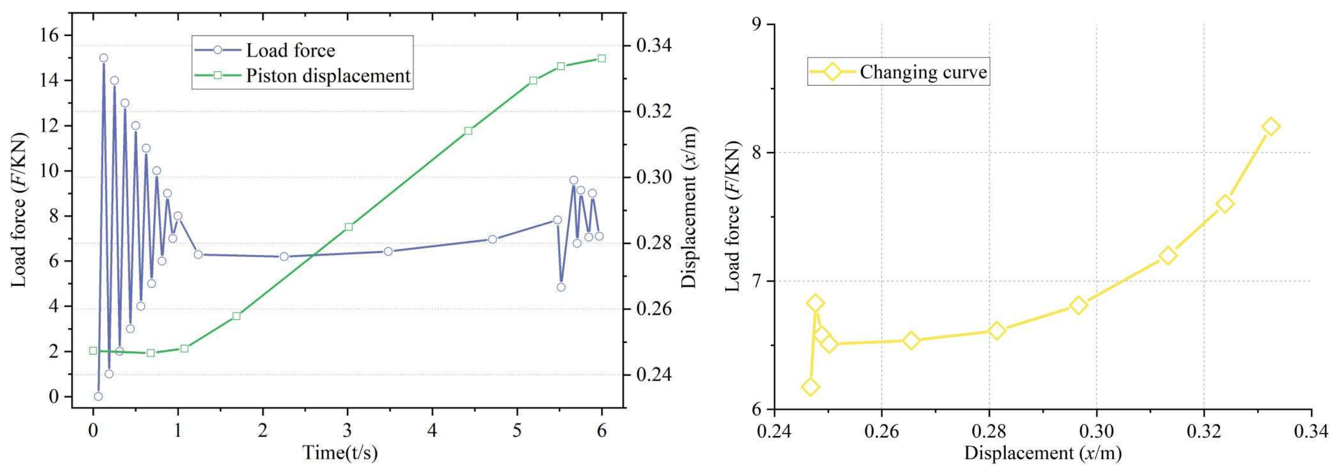
As can be seen from Fig. 4, the fuzzy PID controller consists of two parts: the PID controller and the fuzzy controller, and its core idea is to first convert the expert's PID parameters into fuzzy rules to be deposited into the MCU part, and then the MCU updates the PID parameters through the real-time error and its rate of change, to achieve the purpose of making the movement of the controlled object more stable and smooth.

III. Application and analysis of load control algorithms

III. A. Design of fuzzy adaptive PID controller

III. A. 1) Loader rocker hydraulic cylinder load characteristics analysis

Rocker arm hydraulic cylinder extends the load displacement curve in Figure 5, Figure 5(a) shows with the piston extends the rocker arm hydraulic cylinder load force with displacement curve, the piston displacement from the initial displacement of 0.250m gradually extends to 0.340m, the piston force from 6.71KN to 8.32KN. Figure 5(b) for the displacement curve and the force curve coordinates of the displacement curve replaced with the curve obtained by the change in the load force with the displacement of the curve, become the basis of the load force input signal in AMEsim. The basis of the load force input signal in AMEsim later.



(a) Load displacement curve (b) Load vs. displacement curve

Figure 5: The extended pressure load curve of the rocker hydraulic cylinde

III. A. 2) Fuzzy adaptive PID controller design

The fuzzy PID control algorithm proposed above is applied to adjust the three control parameters k_p、k_i、k_d of the control spool displacements of the hydraulic cylinder of the loader, and set the physical theories of the fuzzy correction coefficients k_p、k_i、k_d respectively. Where k_p is the scale factor, which affects the accuracy and response factor of the control system. k_i is the preset integral coefficient, which affects the control accuracy of the system. k_d is the preset differential coefficient, which affects the regulation time of the control system.

The fuzzy set of signals is obtained by fuzzy segmentation of the input signals in the fuzzy controller, which maps the cleaned inputs into fuzzy quantities using seven fuzzy language values: {negative large, negative medium, negative small, zero, positive large, positive medium, positive large} = {NB, NM, NS, ZO, PS, PM, PB}.

From the perspective of improving the positioning accuracy of the valve cylinder piston, combined with the fuzzy loading rules of the existing mechanical hydraulic system, the fuzzy control rules for the control parameters k_p、k_i、k_d are obtained as shown in Table 1-Table 3.

Table 1: Δkp fuzzy control rule

	NB	NM	NS	Z	PS	PM	PB
NB	PS	PB	PB	PM	NB	PS	ZO
NM	PB	NS	ZO	PS	NB	Z	Z
NS	PM	PM	PM	PS	Z	NS	NM
Z	NS	PS	PS	Z	NM	PM	PB
PS	PS	PS	Z	NS	NS	NM	NM
PM	Z	Z	NS	PM	NS	NS	NB
PB	Z	PM	ZO	NM	NM	NS	NB

Table 2: Δki fuzzy control rule

	NB	NM	NS	Z	PS	PM	PB
NB	PS	NB	NB	PM	NM	Z	Z
NM	PM	PS	PB	PS	NM	Z	ZO
NS	PB	NM	PM	NS	Z	PS	NM
Z	NM	NS	NS	Z	NM	PS	PM
PS	NS	PS	Z	PS	PS	PS	PM
PM	Z	Z	PS	PM	PM	PB	PS
PB	Z	Z	PM	PM	PB	PB	PM

Table 3: Δkd fuzzy control rule

	NB	NM	NS	Z	PS	PM	PB
NB	PS	NM	Z	NB	Z	PB	PB
NM	NS	NS	NS	NS	Z	PM	PM
NS	NB	PM	NM	NS	NM	PS	PM
Z	NB	NM	NM	NM	Z	NB	NB
PS	PM	NM	NS	NS	NS	PS	PS
PM	NM	NS	NS	NS	NS	PM	PM
PB	PS	Z	ZO	NS	NS	PB	NB

III. B. Simulation and analysis study of energy-saving working conditions

Figure 6 shows the flow rate and power loss curve of the main valve port of the pressure compensation valve 11. From the simulation results, it can be found that the flow rate and power parameters reach stability after 0.5s of simulation, and the simulation data is taken when the simulation time is 3s, the flow rate is stabilized at 71.2242L/min, and the power loss is stabilized at 1. 0989kW.

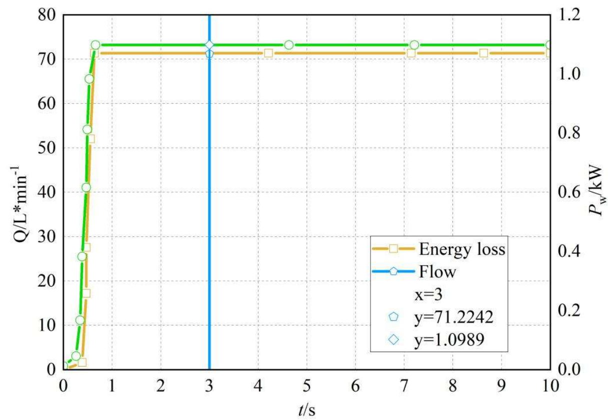


Figure 6: Curve of flow and power loss through the main port of multi-way valve

Set the branch 1 working load constant 25560N, branch 2 working load constant 15670N, multi-way valve throttle port size is the same, the flow control valve spring preload for 30N to remain unchanged, the energy-saving controller spring preload 112N, taking into account the energy-saving controller of energy-saving role of the energy-saving controller, the energy-saving controller spool diameter d = 5.2mm into the pressure calculation formula as formula (20):

$$p = 4F / \pi d^2 \tag{20}$$

The pressure drop generated on the energy-saving controller is obtained to be about 0.65 MPa, and the simulation time is set to be 10 s. The simulation is carried out, and the results of the hydraulic pump outlet, the pressure at the right end of the flow regulating valve, and the change curve of the spring force are shown in Fig. 7 after the energy-saving effect of the energy-saving controller is taken into account.

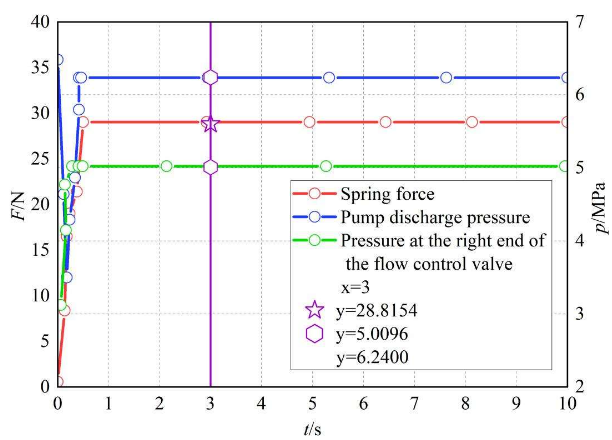


Figure 7: Curve of flow and power loss through the main port of multi-way valve

From the simulation results, it can be found that at the beginning of the simulation, there is a slight oscillation in the pressure difference between the left and right of the valve spool, and it tends to stabilize after 0.4s. Taking the simulation data when the simulation time is 3s, the pressure difference between the left and right chambers of the flow regulating valve is 1.2304MPa when the preload force of the spring of the energy-saving controller is 12N, and the outlet pressure of the pump is lowered by 0.65MPa due to the energy-saving effect of the energy-saving controller. AMESim post-processing function to obtain after considering the energy-saving controller energy-saving effect of the multi-way valve main valve port flow and multi-way valve port saving power curve is shown in Figure 8, from the simulation results can be found, simulation 0.4s after the flow and power parameters to reach a stable, take the simulation time of 3s simulation data, the flow rate stabilized in the 73.5539L/min, saving power stabilized at 0.8691kW.

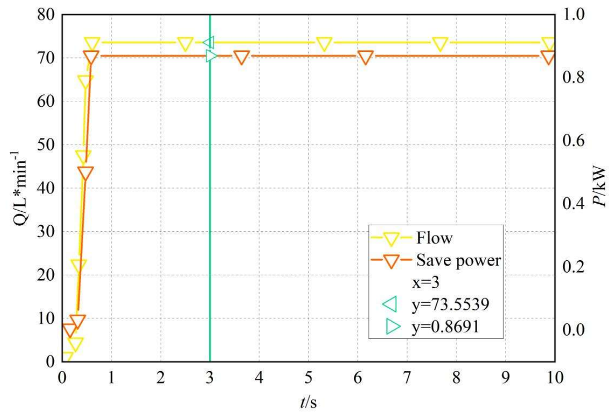


Figure 8: Main valve port flow and power saving curve of multi-way valve port

IV. Conclusion

In this paper, an optimization scheme for load adaptation of mechanical hydraulic system is proposed by modeling the load-sensitive valves of mechanical hydraulic system and designing a fuzzy controller by integrating the optimization genetic algorithm and fuzzy PIND technique.

In the application experiment of loader rocker hydraulic cylinder, after establishing the fuzzy set of affiliation conditions based on the principle of implied parallelism, the control rules of k_p、k_i、k_d three control parameters are obtained and the fuzzy adaptive PID controller is constructed. In the simulation experiment of the mechanical hydraulic system load energy-saving conditions supported by this controller, the main valve port flow, power, hydraulic pump outlet, right end pressure of the flow control valve, spring force, main valve port flow of the multidirectional valve and the multidirectional valve port saving power are all stabilized at 3 s. And the main valve port flow is stabilized at 71%, and the multidirectional valve port saving power is stabilized at 71%. And the main valve port flow rate is stabilized at 71.2242L/min, power loss is stabilized at 1. 0989kW, the spring force reaches 12N when the pump outlet pressure is reduced by about 0.65MPa, and the main valve port flow rate of the multi-way valve is stabilized at 73.5539L/min, and the power saving is stabilized at 0.8691kW.

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