Pulsation Simulation and Energy Consumption Analysis of Series Pump Valve Cooperative Control Hydraulic System

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Abstract

In order to reduce the pulsation and the energy consumption of the hydraulic system, the series pump and valve cooperative control hydraulic system is designed, and the pulsation simulation and energy consumption analysis of it is carried out. Firstly, the working principle of series pump valve co control system is studied. Secondly, the mathematical model of series pump valve cooperation control system is established. And then the Controller of series pump valve cooperation control system is designed. Finally, the simulation analysis of the proposed hydraulic system is carried out, and results show that the proposed system has high stability and low energy consumption.

Keywords: Series pump and valve cooperative control, hydraulic system, pulsation simulation, energy consumption.

1 Introduction

The electrical hydraulic servo valve control system has some advantages, such as strong load capacity, big load stiffness, large power density, fast

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response and high controlling precision, therefore it has irreplaceable position in movement controlling field, it has been applied in many fields, such as aerospace, weapon, machine tool, and ship. The servo direct drive pump control system has been widely applied for its high efficiency, wide range of regulating speed and good stability. The servo valve controlling system has many advantages mentioned above, but it has also many disadvantages, such as low controlling precision and low response. In order to combine the advantages of valve controlling system and pump controlling system, the valve controlling system and direct drive system can be combined to form a novel cooperative control hydraulic system.

In the normal series pump vale cooperative controlling system, the pump controlling circuit and valve controlling circuit are combine in series. The basic theory of it is to match the output flow rate of pump station system and load flow through changing rotational speed of motor or displacement of pump, and then the overflow loss of the relief valve is reduced, the energy utilization efficiency can be improved. Based on the series flow control valve can regulate the load flow precisely and the system bandwidth can be increase.

Zhicheng Xu et al. proposed a hydraulic system combing valve-pump combined controlled hydraulic system with multiple accumulators, and the energy efficiency of the proposed system is improved [1]. Gong Guofang et al. proposed valve and pump compounded pressure control system for the hydro viscous clutch to enlarge the working pressure range, and the pressure control range can be increased [2]. Bing Xu et al. Designed a three level controller to improve the energy efficiency, through coordinate control of the pump and valves, the meter in valve opens maximally and the pressure losses across the valves is minimized [3]. Litong Lyu et al. proposed a parallel connected pump valve coordinated system, and carriedout further improvements to control design of it, experiment results showed that the proposed sysetm has better tracking performance and can improve the energy efficiency [4]. Prabhakar Kushwaha et al. compared the dynamic performances of the valve control system, the pump control system and prime mover control system for a wide range of operating conditions. Analysis results showed that valve controlling system is more sensitive than other systems, and the pump control system has less overshoot [5]. Du Jia et al. proposed a pump-valve coordinated composite control hydraulic system. Experiment results showed that the proposed system can reduce the power consumption greatly [6]. Liu Hua et al. proposed an energy saving system to coordinate control the pump speed and independent metering, experiment results showed that the proposed system could improved pressure control accuracy. References [7]. Wang yunpeng et al. designed a pump and valve coordinated control electro-hydrostatic actuator, and the pump control and valve control models are constructed. The advantages of the proposed system are verified through experiment analysis [8]. Ying yizhi et al. proposed the intelligent prestressed tensioning equipment based on cooperative control technology of servo pump valve. The proposed system can control oil pump stably [9]. Zhao Shuli proposed an industrial intelligent pump valve coordinate control platform based on internet of things technology, the proposed system could realize collaborative operation between the initial inusrial level intelligent pump and valve control systems that independently deal with specific functions, and effectively integrates the existing related application business and new application business into one [10].

The dead time and nonlinear friction characteristics of hydraulic components make the electro-hydraulic servo control system have significant nonlinear characteristics and uncertain parameters. It is necessary to improve its control performance through control technology. The main controlling algorithms conclude PID controlling algorithm, self adaption controlling algorithm, sliding mode variable structure control algorithm and robustness controlling algorithm. PID control has the advantages of simple principle, easy to understand, reliable operation and so on. Moreover, it does not need an accurate system model, and the parameter adjustment is simple. It is widely used in various engineering applications. However, the traditional PID controller is a simple linear superposition of proportional term, integral term and differential term, and the parameters remain unchanged in the control process, so it is difficult to solve the contradiction between the rapid response and stability of the system, and the control performance of PID can not be guaranteed in the system with variable parameters. The parameters of the hydraulic system are variable and have significant nonlinearity, so the traditional PID control can not achieve good control effect.

The adaptive control algorithm can change the structure or parameters of the controller by identifying the model parameters of the time-varying system, so that the system can have the characteristics of high control accuracy and strong anti-interference at the same time, so it is widely used in the electro-hydraulic servo system with significant time-varying characteristics. In practical application, adaptive control has some shortcomings, such as the identification process of system time-varying parameters is complex, the amount of calculation is large, and it is difficult to meet the requirements of real-time control for the system with fast response speed.

Because the design of sliding mode is independent of disturbance and plant parameters, the variable structure control has the advantages of insensitive to disturbance and parameter changes, fast response, no need to identify system parameters online, and simple physical implementation. The disadvantage of sliding mode variable structure control is that when the state trajectory reaches the sliding surface, it will travel back and forth on both sides of the sliding surface, resulting in chattering, and can't strictly slide along the sliding surface towards the equilibrium point.

Robust control is a kind of control algorithm which can adapt to system model uncertainties and system micro perturbations. It can still ensure the good performance of the system when dealing with system perturbations. The robust controller is designed under the perturbation boundary conditions of the system. The designed robust controller can ensure the normal operation of all cases within the perturbation boundary of the system, and ensure good control performance.

Working Principle of Series Pump Valve Co Control System

According to the function, the series pump valve co control system can be divided into two parts: pump control system and valve control system. The function of the pump control part is to adjust the output flow of the oil source according to certain rules on the premise of meeting the demand of the system load flow, so as to reduce the overflow loss of the system and improve the efficiency of the system.

Under the ideal condition, when the output flow of oil source is equal to the load flow, the overflow flow of the system is zero, and the flow efficiency of the system is the maximum. However, in the actual hydraulic system, there is a certain amount of leakage in the hydraulic pump and cylinder. In order to maintain the pressure stability of the system, there must be a certain overflow flow at the relief valve, so the oil source flow is usually greater than the load flow. The valve control part of the series pump valve co control system is actually a typical valve control system. The only difference is the oil source. There are inevitably a lot of overflow loss and throttling loss in valve control system. In the series pump valve coordinated control system, the output flow of the pump control part is adjustable, which is equivalent to providing a load sensitive constant pressure oil source for the valve control part. Therefore, the series pump valve coordinated control system can also have the excellent dynamic performance and control precision of the traditional valve control system.

According to the different ways of adjusting the output flow of oil source in the pump control part of the system, the series pump valve coordinated control system can be divided into the following three structural forms:

(1) AC servo motor + one-way quantitative pump: the scheme adjusts the output flow of the oil source by changing the speed of the hydraulic pump. Its advantages are simple structure, small space occupation, high reliability and low cost; The disadvantages are large moment of inertia of motor and hydraulic pump, low frequency response and poor dynamic characteristics of the system.
(2) Constant speed motor + servo variable pump: the scheme adjusts the output flow of oil source by changing the displacement of variable pump. At present, servo variable displacement pump is usually composed of swash plate piston pump and servo variable displacement mechanism. Servo variable mechanism is actually a valve controlled cylinder system. The advantages of this scheme are mature technology and good dynamic characteristics of the system. The disadvantages are that the swash plate of the servo variable displacement pump needs to be driven by the valve controlled cylinder. The system needs a set of auxiliary oil source device. There is a waste of energy in the operation of the valve controlled cylinder variable mechanism. Moreover, the cost of the servo variable displacement pump is expensive and the system structure is more complex.
(3) AC servo motor + servo variable pump: this scheme can change the speed and displacement of the hydraulic pump at the same time, so that the dynamic characteristics of the system can be greatly improved, and the reliability of the system is significantly improved due to the dual redundancy control. But on the other hand, this method also increases the complexity of the system structure and the difficulty of the control strategy design. After decades of development, AC servo motor and frequency control technology have greatly improved in dynamic performance and control accuracy.

In this research, the structural diagram of series pump valve coordination system is shown in Figure 1. From the perspective of functional division, the independent control subsystem of import and export realizes the compound control of the position and pressure of the actuator by controlling the spool displacement of the two servo valves, so as to reduce the energy consumption and improve the stability of the system; The pump control subsystem provides flow and pressure for the system by controlling the speed of the pump.

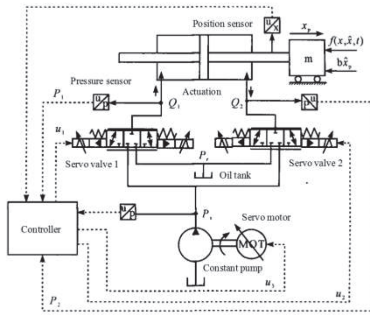

Figure 1 Diagram of series pump valve coordinated controlling system.

Mathematical Model of Series Pump Valve Co Control System

The balance equation of actuator output force and load force is expressed by

$$m\ddot{x}_p = \alpha_1 p_1 - \alpha_2 p_2 - \lambda \dot{x}_p - f(x, \dot{x}, t) \tag{1}$$

where m represents the load mass, $\dot{x}_p$ represents the load speed, x_p represents the load location, p_1 and p_2 are the pressure of two chambers of actuator, α_1 and α_2 are the areas of two chambers of actuator, λ represents the viscous damping coefficient of piston and load, $f(x, \dot{x}, t)$ is function of unmodeled force and external interference force.

The servo valve dynamic is simplified to proportional link, which is expressed by

$$d_{fi} = \kappa_{fi} I_i,\ i = 1, 2 \tag{2}$$

where d_{fi} denotes the spool displacement of servo valve, I_i denotes the input signal of servo valve, κ_{fi} denotes the gain of servo valve.

The flow equations of two chambers of servo valve are expressed by

$$q_1 = \eta_1 \kappa_{f1} I_1 [Q(u_1)\sqrt{p_g - p_h} + Q(-u_1)\sqrt{p_1 - p_h}] \tag{3}$$

$$q_2 = \eta_2 \kappa_{f2} I_2 [Q(u_2)\sqrt{p_2 - p_h} + Q(-u_2)\sqrt{p_g - p_2}] \tag{4}$$

where $Q(x) = \begin{cases} 1, x \geq 0 \\ 0, x < 0 \end{cases}$, $\eta_i = \mu_i \phi_i \sqrt{2/\rho}$, μ_i represents the discharge coefficient, ϕ_i represents the area gradient o servo valve, ρ denotes the density of hydraulic oil, q_i represents the flow of two chambers of hydraulic tank, p_h represents the supplying oil pressure, p_g represents the returning oil pressure.

The dynamic equations of two chambers of actuator are expressed by

$$p_1 = \frac{\delta_e}{W_1}(q_1 - \alpha_1 \dot{x}_p) \tag{5}$$

$$p_2 = \frac{\delta_e}{W_2}(\alpha_2 \dot{x}_p - q_2) \tag{6}$$

where W_1 and W_2 are the volume of two chambers of hydraulic tank, δ_e represents the volume elastic module of oil.

Suppose that the frequency response of motor is higher than working frequency of actuator, the dynamics of motor can be simplified as portion part, which is expressed by

$$\omega_m = \kappa_m I_3 \tag{7}$$

where κ_m represents the gain of motor rotational speed, I_3 denotes the input signal of servo motor.

The flow of servo valve is expressed by

$$q_{in} = Q(x_d - x_p)q_1 + Q(x_p - x_d)q_2 \tag{8}$$

Suppose the frequency response of pump control subsystem is higher than working frequency of actuator, the mechanical dynamics in pump is ignored, and the outlet pressure dynamics of pump is expressed by

$$p_o = \frac{\delta_e}{W_b}[d_b \kappa_b I_3 - q_{in} - q_1] \tag{9}$$

where W_b represents the volume of hydraulic pump, d_b represents the discharge of constant pump.

The pump valve coordinated control system is a multiple input multiple output system with strong coupling; The specific performance is: there is a coupling relationship between the output pressure of the pump and the flow rate of the flow servo valve, and the pressure of the two cavities of the pump is also coupled with the piston speed.

Controlling Algorithm of Series Pump Valve Co Control System

Suppose that uncertain nonlinearity Δ can be decomposed into slowly time-varying parts Δ_h and fast time-varying parts Δ_k. In addition, the boundary between Δ_h and Δ_k is definite, and the following expresses can be obtained.

$$\Delta = \Delta_h + \Delta_k \tag{10}$$

$$|\Delta_h| \leq \xi_h \tag{11}$$

$$|\Delta_k| \leq \xi_k \tag{12}$$

where ξ_h and ξ_k are positive real number determined.

Next, the system is divided into motion dynamics and pressure dynamics, and the controller is designed in two steps.

(1) Design of controller 1

The expected trajectory is defined by $x_e(t)$, and the tracking error is defined by $E_g = x_1 - x_e$, and the switching surface is defined by $z_q = E_g + \kappa_g E_g$, where κ_g is the positive feedback gain, therefore the z_q convergence is equivalent to E_g convergence, and the semipositive definite Lyapunov function is expressed by

$$L_g = \frac{1}{2}\psi z_q^2 \tag{13}$$

where ψ is the positive weight factor, the derivative of L_g with respect to time is expressed by

$$\dot{L}_g = \psi z_q \left[\frac{1}{m}(\lambda \dot{x}_p - \kappa_{f1} \arctan(10^4 x_p)) - f(x, \dot{x}, t) - \Delta_h - \Delta_k - \ddot{x}_p\right] \tag{14}$$

The output force of driving cylinder is defined by P_q, which is used as virtual controlling input of expression (14), and the controlling functions are expressed by

$$f_e = f_{1m} + f_{1r} \tag{15}$$

where f_e denotes the expected controlling force, which concludes f_{1m} and f_{1r}. f_{1m} compensates the modeling forces such as Coulomb friction force, gravity force and load force in the system, and introduces the compensation of expected inertial force and expected viscous resistance into the forward channel as the model compensation term. f_{1r} is a robust control item, which is expressed by

$$f_{1r} = f_{1r1} + f_{1r2} \tag{16}$$

where f_{1r1} is the nominal stabilization part, which is linear feedback control, including proportional feedback and differential feedback of position error,

f_{1r2} is designed to deal with uncertainty, nonlinearity and model error. It is a smooth function.

(2) Design of controller 2

Let p_{1e} and p_{2e} represent the expected pressures of the two cavities of the driving cylinder respectively. In order to make z_q converge, they must satisfy the constraint condition:

$$\alpha_1 p_{1e} - \alpha_2 p_{2e} = f_e \tag{17}$$

When the leakage is ignored and the hydraulic cylinder works at a steady working point, the flow rate should meet the requirements

$$\frac{q_1}{q_2} = \frac{\alpha_1 x_1}{\alpha_2 x_2} \tag{18}$$

The following expression can be deduced:

$$\begin{bmatrix} p_{1e} \\ p_{2e} \end{bmatrix} = \vec{J}_e^{-1} p_e \tag{19}$$

where

$$\vec{J}_e = \begin{cases} \begin{bmatrix} \alpha_1 & -\alpha_2 \\ 1 & \frac{\alpha_1^2}{\alpha_2^2} \end{bmatrix} & x_2 > 0 \\ \vec{J}_e' & x_2 = 0 \\ \begin{bmatrix} \alpha_1 & -\alpha_2 \\ \frac{\alpha_1^2}{\alpha_2^2} & 1 \end{bmatrix} & x_2 < 0 \end{cases}, \quad p_e = \begin{bmatrix} f_e \\ p_y \end{bmatrix},$$

p_y represents the pressure of source.

Obviously, if p_1 and p_2 converge to p_{1e} and p_{2e} at the same time, P_q can realize the pairing f_e, so z_q converges. In order to deduce the control law that makes the pressure of the two cavities of the driving cylinder follow the desired pressure, the Lyapunov function is constructed based on the theory of hydraulic passivity.

The semi definite Lyapunov function is constructed by using the pressure error energy storage function of the two cavities of the driving cylinde1r, and the control law is derived by

$$L_g' = \chi(W_1(x_1)D(\hat{p}_1, p_{1e}) + W_q(x_1)D(\hat{p}_2, p_{2e})) \tag{20}$$

where $\hat{p}_1 = p_1 - p_{1e}$, $\hat{p}_2 = p_2 - p_{2e}$, χ represents the positive weight factor.

Case Study

This research uses AMESim and MATLAB co simulation platform for simulation analysis. The hydraulic model is built in AMESim and the control strategy is implemented in MATLAB. The energy efficiency of the system is compared with the energy-saving load sensing system; In the simulation, the hydraulic cylinder drives the load to rise and fall vertically. When the load rises, the gravity hinders the movement of the hydraulic cylinder; When the load drops, the gravity assisted hydraulic cylinder moves beyond the retraction condition.

The position tracking command is $x_d = 0.06 \sin 2\pi t$, the differential pressure command is 15 bar, and the right chamber pressure command is 20 bar. The basic parameters of the pump and valve coordinated control system are listed in Table 1.

The displacement curves are shown in Figure 2. As can be seen from Figure 2, the displacement has a certain lag, but maintains good following characteristics, there is no overshoot in the reduction stage, and the control accuracy is up to 0.5 mm.

Figure 3 shows the pressure curve of pump and valve coordinated control system, in which p_a is the main cylinder pressure, p_h is the return cylinder pressure, and p_o is the pump port pressure. It can be seen from figure 3 that the pressure at the pump port varies with the load pressure of the main cylinder and return cylinder. However, the pressure change rate is slow due to the response speed of the frequency converter motor. The pressure difference

Table 1 Parameters of pump and valve coordinated control system

Parameter	Value
Actuator stroke/m	0.5
Area of left chamber of actuator/m^2	5.932×10^{-4}
Area of right chamber of actuator/m^2	5.932×10^{-4}
Load mass/kg	125
Rated pressure of servo valve/bar	40
Rated flow of servo valve/L/min	42
Constant pump discharge/cc/rec	15
Rated current of servo valve/mA	38
Volume elastic module of oil/bar	7000

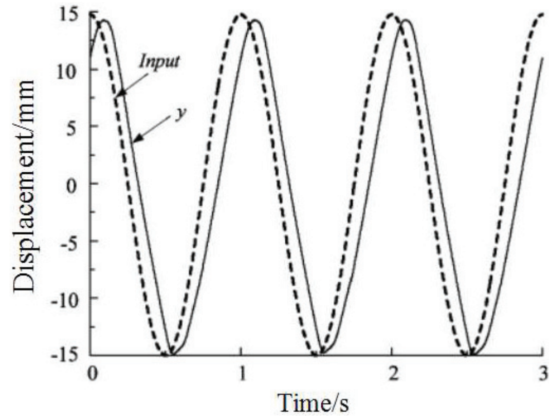

Figure 2 Displacement curve of pump and valve coordinated control system.

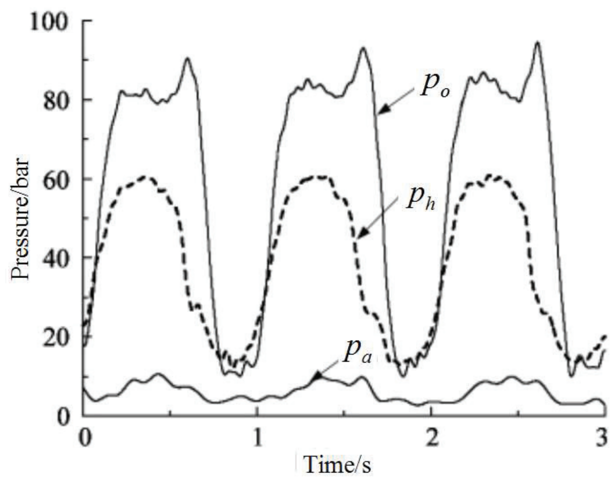

Figure 3 Pressure curve of pump and valve coordinated control system.

between the pump port and return cylinder is basically 20 bar in return stage, and the response time is about 0.15 s. When the return stroke is switched to depression, the pump port pressure shall change with the master cylinder pressure. At the moment of switching, the difference between the pump port and the master cylinder pressure is greater than the set value, and the given signal of the motor is set to zero. However, due to the inertia of the motor, the quantitative pump still inputs flow to the system, resulting in a spike in the pump port pressure. With the decrease of the input flow of the rear pump and

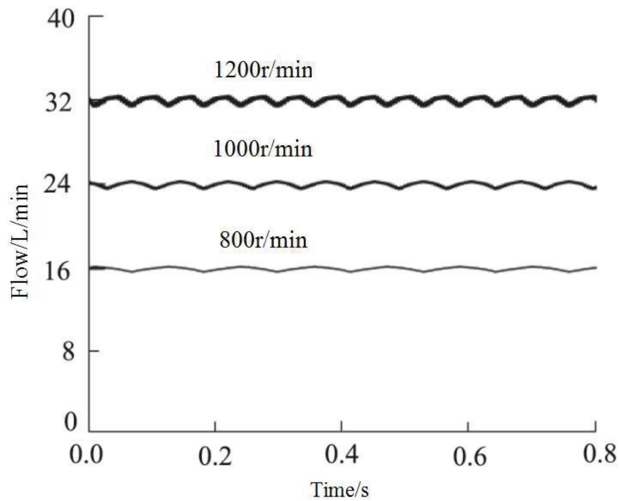

Figure 4 Flow pulsation of series pump and valve coordinated controlled hydraulic system under 25 Bar.

Table 2 Flow pulsation distribution of single pump and series pump controlled hydraulic systems under different pressures

Load/Bar	Minimum Value/L/min	Maximum/L/min	η/%
25	31.3	31.6	2.2
30	33.5	33.7	2.3
35	34.7	34.9	2.5

the opening of the inlet valve of the master cylinder, the pump port pressure decreases.

Figure 4 shows the flow pulsation of series pump and valve coordinated controlled hydraulic system under load of 25 Bar, and Table 2 lists the analysis results of flow pulsation distribution of single pump and series pump and valve coordinated controlled hydraulic system. η indicates the proportion of the output flow fluctuation value to the average flow, which reflects the fluctuation of flow fluctuation to a certain extent. The smaller the value, the better the stability.

The load of the system increases from 25 Bar to 35 Bar, and the difference of the pump output flow of the single pump control hydraulic system also increases, forming a larger flow pulsation range. When the load increases, the flow pulsation range of the series pump control hydraulic system also decreases. This is because increasing the load will increase the burden of the

Table 3 Energy consumption analysis of pump and valve coordinated control hydraulic system

Time/s	Output Energy of New System/kJ	Output Energy of Old System/kJ
5	87	102
10	179	197
15	215	267
20	296	326
25	370	391
30	423	450

hydraulic system, reduce the operation stability, and then show the increase of flow fluctuation. It is noted that the flow pulsation of the series pump controlled hydraulic system does not change significantly with the increase of load, which shows that the design of the system has good stability.

The energy consumption of the pump and valve coordinated control hydraulic system is carried out, and the analysis results are listed in Table 3. As seen from Table 3, in the same time, the energy consumption of the new pump valve cooperative compound control hydraulic system is greatly reduced compared with the traditional hydraulic system.

Conclusions

Aiming at the low energy efficiency of valve control and pump control position servo system, the energy-saving control strategy of pump valve cooperative control servo system is studied, the energy-saving control strategy of pump valve cooperative control hydraulic system is proposed, the mathematical model of pump valve coordinated control system is established, the controller is designed, and the pulsation simulation analysis and energy consumption analysis are carried out. The analysis results show that, the proposed pump valve coordinated control hydraulic system has good stability, and can greatly reduce the energy consumption, so it has better energy saving effect.

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