【Short Paper】

Driving pulsation analysis of electro-hydraulic robot

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E-mail:cgzhu@126.com**Received: 8 October 2019; Revised: 29 December 2019; Accepted: 28 January 2020****Abstract**

The hydraulic driving source plays an important role in the performance of the electro-hydraulic robot. In this paper, the Lumped parameter method is used to establish the pressure-flow characteristic model of the constant pressure variable pump of a 6-DOF electro-hydraulic robot, and attain the influencing factors of the flow pulsation. The simulation model of the electro-hydraulic robot is established, which is used to study the oil property and plunger leakage characteristics of the constant pressure variable pump on the affection of the flow pulsation. The experimental results display that the elastic pulsation is the main form of the output flow pulsation of the constant pressure variable pump, and the pulsation rate will increase with the load quantity of the robot.

Keywords : Electro-hydraulic robots, Constant pressure variable pump, Pressure-flow characteristic, Flow pulsation, Simulation

1. Introduction

The motor-driven robot has been widely used in recent years, but it has some shortcomings, such as low power density, small output torque, etc. The electro-hydraulic robot with low cost, large speed range and strong load-carrying capacity has attracted more and more attention. The constant pressure variable pump is the main driving source of the electro-hydraulic robot. It has the advantages of energy saving and high output because it can realize full displacement output under small load and self-adaptive output performance under large load flow. Since there exists the pulsation characteristics in the constant pressure variable pump, which are the key factors affecting the motion stability of electro-hydraulic robots. Therefore, it is of great significance to study the pulsation characteristics of the constant pressure variable pumps in the electro-hydraulic robot.

In 2012, (Bergada et al., 2012a, 2012b) focused on the relationship between leakage flow and output flow pulsation of plunger variable displacement pump, and verified the relationship between flow pulsation output pressure and inclined plate of plunger variable displacement pump through experiments. In order to explore the method of improving the pressure-flow pulsation of plunger variable displacement pump. The effects of piston diameter, maximum piston strokes (inclined plate angle and pitch diameter) on the pulsation characteristics of the plunger variable displacement pump are determined. Then the internal mechanical structure of the plunger variable displacement pump is further optimized (Bae et al. and Sa et al., 2014a, 2015b). (Wei J, et al., 2015) designed a nonlinear controller based on the control-oriented mathematical model, and the stability of the whole system is proved using Lyapunov theory. In addition, a reducing the output flow pulsation rate method by optimizing the valve plate structure of the plunger variable displacement pump was proposed in 2017 (Wu et al., 2017), and the optimal simulation results were obtained by using AMESIM software: when the output pressure of the constant pressure variable pump was 200 bar, 300 bar and 400 bar, the output flow pulsation rate decreased by 37.05%, 38.54% and 41.04%, respectively.

Based on the complexity of the mathematical modelling of the electro-hydraulic robot and the flow pulsation characteristics of the constant pressure variable pump, the sensitive factors analysis method and AMESIM simulation modelling method are used to study the influence of output flow pulsation rate of the constant pressure variable pump in this paper. The pressure-flow of the end actuator of the electro-hydraulic robot and the constant pressure variable pump are analyzed through experiments. The experiment results can provide the guarantee for optimizing the control

performance of the electro-hydraulic robot and the constant pressure variable pump.

Our contributions can be summarized as follows:

- In view of the shortcomings of the electro-hydraulic robot system that it is difficult to accurately establish a mathematical model, a detailed and accurate simulation model is obtained by using the physical characteristics and self-defining characteristics of AMESIM.

- The pressure flow characteristic equations are established according to the formation mechanism of the pulsation characteristics of the constant pressure variable pump. Combined with the physical simulation model and experimental data, the influence factors to the pulsation characteristics of the constant-pressure driving source is analyzed, and then the optimization direction is determined, which lays a foundation for improving the control accuracy of the end actuator.

2. System principle of electro-hydraulic robot system

A 6-DOF electro-hydraulic robot system is shown in Fig. 1.

Fig. 1 A 6-DOF electro-hydraulic robot system

1) constant pressure variable pump; 2) tank; 3) load-sensitive proportional multiplex valve; 4) pressure gauge; 5) end actuator; 6) wrist; 7) lower arm; 8) median arm; 9) upper arm; 10) waist; 11) base

As shown in Fig. 1, A 6-DOF electro-hydraulic robot system consists of a 6-DOF electro-hydraulic robot and a hydraulic station, and the 6-DOF electro-hydraulic robot includes seven parts: end actuator, wrist, a lower arm, a median arm, an upper arm, a waist, a base, and forming six joints, while the hydraulic station mainly includes a constant pressure variable pump, a tank, and a load-sensitive proportional multiplex valve. Each joint of the robot is directly driven by a hydraulic motor to do rotary motion. The working principle of the robot system is shown in Fig. 2.

Fig. 2 Workflow diagram of 6-DOF electro-hydraulic robot system

In Fig.2, when the hydraulic motor starts, the constant pressure variable pump sucks the low pressure oil from the tank, and outputs the high pressure oil to the load-sensitive proportional multiplex valve. The load-sensitive proportional multiplex valve controls the input oil pressure and flow of the hydraulic motor of the robot joints by the console, thus the

constant pressure variable pump can drive the motions of the joints of the robot.

The main regulating mechanism of the constant pressure variable pump is shown in Fig. 3.

Fig. 3 Main regulating mechanism of the constant pressure variable pump
1) inclined plate; 2) variable piston; 3) constant pressure valve.

The constant pressure variable pump is composed of a plunger variable displacement pump and a constant pressure valve. A_v is the control area of the constant pressure valve (mm^2), F_0 is the pre-tightening force of the spring of the constant pressure valve (N), A_1 is the cross section area of the upper cavity of piston (mm^2), A_2 is the cross section area of the lower cavity of piston (mm^2), δ_p is the flow leakage coefficient ($\text{m}^3/\text{s}/\text{Pa}$), p_s is the output pressure of the constant pressure variable pump (MPa), x_v is the core displacement of the constant pressure valve (mm).

In Fig.3, the inclined plate is in the position of maximum angle inclination before the load quantity of the robot doesn't reach the setting pressure of the constant pressure valve. The displacement of the constant pressure variable pump is maximum and constant. When p_s increases to a setting pressure of the constant pressure valve, the core displacement of the constant pressure valve starts to move with x_v , which push the variable piston and change the inclined plate angle.

3. Output pulsation characteristics of constant variable pump

In electro-hydraulic robot system, the working characteristic of the constant pressure variable pump is the main factor of causing the flow pulsation. In order to improve the output accuracy of the end actuator of the robot and reduce the noise of the constant pressure variable pump, it is necessary to study the influence factors of causing the flow pulsation of the constant pressure variable pump.

3.1 Pressure-flow characteristics of constant pressure variable pump

Pressure-flow characteristics of the constant pressure variable pump is studied using Lumped parameter method, the constant pressure valve and the pipeline after the output of the plunger variable displacement pump are modeled as a whole "pipeline" (Xu et al., 2012). Pressure-flow principle of the constant pressure variable pump is shown in Fig. 4.

Fig. 4 Pressure-flow principle diagram of the constant pressure variable pump
1) tank; 2) filter; 3) constant pressure variable pump; 4) safety valve; 5) oil pipeline; 6) throttle valve; 7) hydraulic motor

In Fig.4, the constant pressure variable pump outputs the oil to the hydraulic motor through the "pipeline". The output flow of the plunger variable displacement pump, the "pipeline" and the throttle valve are q_d , q_s , q_z , respectively. p_H is the "pipeline" pressure; p_T is the return pressure of the constant pressure variable pump.

According to the motion principle of the plunger, the output flow q_d of the plunger variable displacement pump under ideal conditions is obtained.

$$q_d = \frac{\pi d^2}{4} R \omega \tan \beta \frac{\sin [\varphi_1 + (m-1)\pi/z] \sin m\pi/z}{\sin \pi/z} \quad (1)$$

Where z is the number of plungers in the plunger variable displacement pump. d , φ_1 are the diameter and initial angle of the piston, respectively. R is the distribution circle radius of the pistons. ω is the rotation rate of the plunger variable displacement pump. β is the inclined plate angle. m is the number of the oil discharge plungers.

It can be seen from Eq. (1) that q_d has the pulsation characteristic, which is affected by many sensitive factors (Cidras et al., 2013). Usually, the ratio of the pulsation difference and the average value of q_d is used to express the output flow pulsation of the plunger variable displacement pump, which is called the flow pulsation rate σ_q .

The flow at the throttle valve is calculated as the following.

$$q_z = C_r A_c \sqrt{2(p_H - p_T)/\rho} \quad (2)$$

Where q_z is the flow at the throttle valve, C_r is the discharge coefficient, A_c is the overflow area of the throttle valve, p_H is the "pipeline" pressure, p_T is the return pressure of the constant pressure variable pump, and ρ is the density of oil.

The relationship between the flow at the throttle valve and the output flow of the constant pressure variable pump is as the following.

$$q_d - q_z = \frac{V_t}{K_e} \frac{dp}{dt} \quad (3)$$

Where V_t is the pipeline volume between the throttle valve and the constant pressure variable pump, K_e is the bulk modulus of the elasticity of oil.

The following equation can be obtained by Eqs. (2) and (3).

$$\frac{V_t}{K_e} \frac{dp}{dt} + C_r A_c \sqrt{2(p_H - p_T)/\rho} = q_d \quad (4)$$

Eq. (4) builds the relationship between the output flow of the constant pressure variable pump and the pipeline pressure. From Eq. (4), it is obtained that exits flow pulsation characteristics in the output pressure-flow of constant pressure pump.

3.2 Analysis of influencing factors of flow pulsation

The output flow pulsation of the constant pressure variable pump is affected by its mechanical structure, the oil elasticity, the plunger leakage, the inclination angle and the spindle speed of the inclined plate. Due to the oil elasticity and the plunger leakage have a great influence on the flow pulsation, the effects of them will be studied in this section.

Ideally, it is assumed that the oil bulk modulus is constant, while in practice, oil is a kind of compressible liquid, so

the oil bulk modulus will change with various practical factors, which is called the oil effective bulk elastic modulus.

The oil effective bulk elastic modulus can be calculated using Eq.(5) (Kim et al., 2012).

$$K_{es} = \frac{(1-\alpha_x) \left(1 + \frac{i(p-p_0)}{K_{e0} + ip + \sigma_T T} \right)^{\frac{1}{i}} + \alpha_x \left(\frac{p_0}{p} \right)^{\frac{1}{k}}}{\frac{(1-\alpha_x)}{K_{e0} + ip + \sigma_T T} \left[1 + \frac{i(p-p_0)}{K_{e0} + ip + \sigma_T T} \right]^{\frac{i+1}{i}} + \frac{\alpha_x}{kp_0} \left(\frac{p_0}{p} \right)^{\frac{k+1}{k}}} \quad (5)$$

Where K_{es} is the oil effective bulk elastic modulus, α_x is the gas content, p is the absolute pressure, p_0 is the standard atmospheric pressure, K_{e0} is the oil bulk modulus under condition of 0°C and 0 MPa, i is the parameter of the oil bulk modulus, σ_T is the temperature-related constant, T is the oil temperature under the experimental conditions, k is the air variable constant.

From Eq. (5), it can be seen that when the stiffness of the hydraulic pipeline or the container is large enough, Eq. (5) can accurately define the relationship between the oil effective bulk elastic modulus and the gas content, the oil pressure and the oil temperature. Among them, the change of the oil pressure will significantly affect the value of the oil effective bulk elastic modulus.

In a piston pump, under the reciprocating movement of the pistons, the internal oil leakage of the pump will produce the pressure-flow ripple, which leads to the flow and pressure pulsation of the pump (Li et al., 2013). Leakage principle diagram of the plunger variable displacement pump is shown in Fig. 5.

Fig. 5 Leakage principle diagram of plunger variable displacement pump
1) inclined plate; 2) slipper; 3) plunger; 4) cylinder block; 5) port plate

As shown in Fig. 5, the plunger variable displacement pump mainly consists of an inclined plate, a slipper, a plunger, a cylinder block, a port plate. Since there are three kinds of mutual movements between the inclined plate and the slipper, the plunger and the cylinder block, the cylinder block and the port plate, there exists three kinds of leakage flow: the first is the leakage flow between the inclined plate and the slipper, denoted by q_{c1} ; the second is the leakage flow between the plunger and the cylinder block, denoted by q_{c2} ; and the third is the leakage flow between the cylinder block and the port plate, denoted by q_{c3} . The leakage flow of the plunger variable displacement pump q_c can be expressed as the sum of the above three kinds of leakage flows, as shown in Eq.(6) (Ivantysyn et al., 2001).

$$q_c = \frac{\pi d_d^4 \delta_1^3 (p_{\beta} - p_n)}{\mu [6d_d^4 \ln(r_{d2}/r_{d1}) + 128\delta_1^3 l_d]} + \left[\frac{\pi d \delta_2^3}{12\mu l_1} (1 + 1.5\varepsilon^2) (p_{\beta} - p_0) - \frac{\pi d \delta_2 v_p}{2} \right] + \frac{\alpha_e \delta_3^3 (p_{\beta} - p_0)}{12\mu} \left[\frac{1}{\ln(r_{r2}/r_{r1})} + \frac{1}{\ln(r_{r4}/r_{r3})} \right] \quad (6)$$

Where μ is the oil viscosity, δ_1 is the leakage gap between the inclined plate and the slipper, d_d is the diameter of the inner hole of the plunger, r_{d1} is the inner diameter of the slipper sealing belt, r_{d2} is the outer diameter of the slipper sealing belt, l_d is the length of the inner hole of the plunger, p_β is the pressure of the plunger chamber, p_n is the pressure of the inner chamber of the plunger variable displacement pump, δ_2 is the leakage gap between the plunger and the cylinder block, l_1 is the contact length of the plunger in the cylinder block.

From Eq. (6), it can be seen that mechanical structure parameters, the oil viscosity, and the pressure are the main influence factors of the leakage flow of the constant pressure variable pump (Bigliardi et al., 2015).

4. Simulation and analysis

The simulation model of the constant pressure variable pump is shown in Fig.6. Throttle valves with step signals are used to simulate the load on the constant pressure variable pump. The system parameters of the constant pressure variable pump are shown in Table1.

Fig. 6 Simulation model of constant pressure variable pump

- 1) motor unit; 2) plunger; 3) slipper pair; 4) inclined plate; 5) inclined plate control signal; 6) oil-distributing pair; 7) throttle valve; 8) super components; 9) variable piston; 10) constant pressure valve; 11) hydraulic oil properties.

Table 1 System parameter of constant pressure variable pump

Parameters	Meaning	Values
n [r/min]	Speed of the pump	1500
A_v [mm ²]	Control area of the constant pressure valve	78.5
F_0 [N]	Pre-tightening force of the spring of the Constant pressure valve	198
δ_p [m ³ /s/Pa]	Flow leakage coefficient	0.1
A_1 [mm ²]	Cross section area of the upper cavity of piston	176.63
ρ [kg/m ³]	Density of oil	870
d_d [mm]	Diameter of the inner hole of the plunger	8.35
d [mm]	Diameter of the piston	10
K_e [MPa]	Bulk modulus of elasticity of oil	1200
δ_1 [mm]	Leakage gap between inclined plate and Slipper:	0.035
δ_2 [mm]	Leakage gap between plunger and cylinder block	0.05
δ_3 [mm]	Leakage gap between cylinder block and Valve plate	0.0128
α_x	Gas content	1%
μ [cP]	Oil viscosity	46
β [deg.]	Inclined plate angle	12.5

The simulation model in Fig.6 is built using AMESIM software. The power of driving source system is generated by

the motor unit(1), which mainly consists of a constant speed motor, a rotating spring damping and a rotating load. The oil flows into the plunger (2), which is connected with the inclined plate (4) through the slipper pair (3). With the rotation of the motor unit (1), the plunger(2) and the inclined plate (4) absorb and drain oil through the oil-distributing pair(6). The oil flows from the plunger variable displacement pump to the variable piston (9) and then outputs to the robot through the constant pressure valve (10). (5) is the control signal model of the inclined plate, (7) is signal model of simulating the load pressure by the throttle valve with step signal, (11) is the attribute model of the hydraulic oil, because each plunger is the same, so the super model(8) is created here with eight plungers and the oil-distributing pair.

4.1 System dynamic characteristic simulation

According to the regulation principle of the constant pressure variable pump, in order to simulate different loads, a five-stage step signal is loaded on the constant pressure variable pump by the throttle valve in Fig. 6, as shown in Table 2.

Table 2 A five-stage step signal of throttle valve

Time(s)	0-0.3	0.3-0.6	0.6-0.9	0.9-1.2	1.2-1.5
Signal value (null)	0.3	0.13	0.38	0.35	0.16

The output pressure of the constant pressure variable pump p_s is obtained, as shown in Fig.7, and the inclined plate angle of the constant pressure variable pump is shown in Fig.8.

Fig. 7 Output pressure of constant pressure variable pump

Fig. 8 Inclined plate angle of constant pressure variable pump

From Fig.7, it can be seen that the displacement of the constant pressure variable pump is the largest when the load pressure is less than 15 MPa. The output pressure will not continue to rise when the load pressure is more than 15 MPa. Therefore, the setting pressure of constant pressure variable displacement pump is 15 MPa. According to the regulation principle of the constant pressure variable pump, when the load pressure is greater than the setting pressure 15MPa, the inclined plate angle decreases, and the inclined plate angle decreases in direct proportion to the load pressure. In Fig.8, When the load pressure of the constant pressure variable pump reaches the setting pressure 15MPa at 0.3-0.6s stage and 1.2-1.5s stage, respectively, accordingly the inclined plate angle of the constant pressure variable pump starts to decrease from 12.5deg., "-" indicates the opposite direction.

In order to investigate the output pressure characteristics of the end actuator of the robot, the AMESIM model of the load-sensitive proportional multiplex valve is established, as shown in Fig.9. In Fig.9, the load-sensitive proportional multiplex valve control the rotary motion of the joints of the robot through controlling the amount of oil entering the hydraulic motor of the joints of the robot. As the last output of the robot, the performance of the robot's end actuator directly reflects the working performance of the robot. Therefore, we will discuss the characteristics of the robot's end actuator under different loads. Load of the end actuator is shown in Table 3.

Table 3 Load of end actuator

Time(s)	0-0.3	0.3-0.6	0.6-0.9	0.9-1.2	1.2-1.5
Load torque (N. m)	70	200	30	50	180

Fig. 9 AMESIM model of load-sensitive proportional multiplex valve

Combining AMESIM model of the constant pressure variable pump and the load-sensitive proportional multiplex valve, the output pressure of the end actuator (p_x) of the robot is obtained, as shown in Fig. 10.

Fig. 10 Output pressure of end actuator

Compared with Fig. 7 and Fig. 10, it can be seen that the change law of p_x is the same with p_s , and p_x is smaller than p_s because the pressure drop of the throttle valve and the pressure loss of pipeline. Because of p_x is lower and the cushioning characteristic of load-sensitive proportional multiplex valve, p_x has less pressure pulsation^[12]. It can also be seen that p_x is attained by p_s , and the output pressure of the constant pressure variable pump directly affects the output pressure of the end actuator.

4.2 Analysis of influence factors of pulsation characteristics

4.2.1 Influence of oil effective bulk elastic modulus and diameter of plunger leak hole

According to Fig. 5, the leakage flow of the plunger variable displacement pump is mainly composed of the leakage flow formed by the three leakage holes. The leakage flow of the plunger variable displacement pump is proportional to the diameter of plunger leak hole. In Fig. 6, the leakage flow is simulated by the setting different diameter of plunger leak hole. The step load at 0-0.75s stage and 0.75-1.5s stage are 350N.m and 230N.m respectively, which are loaded on end actuator. The oil effective bulk elastic modulus and the diameter of plunger leak hole are increased by 50% and decreased by 50% based on the original parameters respectively.

The output flow chart of the constant pressure variable pump is obtained, as shown in Fig. 11. In Fig. 11, the output flow of the constant pressure driving source in the previous stage is smaller than that in the later stage, because the load quantity of the robot is larger in the 0-0.75s stage and smaller in the 0.75-1.5s stage.

(a) Oil effective bulk elastic modulus

(b) Diameter of plunger leak hole

Fig. 11 Influence change of oil effective bulk elastic modulus and diameter of plunger leak hole

According to the calculation method of the flow pulsation rate, the output flow pulsation rate of the constant pressure variable pump under different oil effective bulk elastic modulus and diameter of plunger leak hole is obtained, as shown in Table 4.

Table 4 Pulsation rate of q_s

Parameter	Time	-50%	1200MPa	+50%
Oil effective bulk elastic modulus	0-0.75s	7.35%	9.28%	8.31%
	0.75-1.5s	5.96%	7.67%	7.13%
Parameter	Time	-50%	0.16mm	+50%
Diameter of plunger leak hole	0-0.75s	9.31%	9.27%	9.47%
	0.75-1.5s	7.93%	7.69%	7.54%

In Table 4, compared with the different loading stages in the three changing modes, the pulsation rates of the oil effective bulk elastic modulus and the diameter of plunger leak hole are higher in 0-0.75s stage than in 0.75-1.5s stage. Because the load quantity in 0-0.75s stage is larger than in 0.75-1.5s stage, the pulsation rate of the output flow increased by the load pressure. The simulation result is obtained by comparing the oil effective bulk elastic modulus and the diameter of plunger leak hole: Under the same load condition, when the oil effective bulk elastic modulus decreases by 50% and increases by 50%, the amplitude of pulsation rate increases much more than the diameter of plunger leak hole. Therefore, when the geometric pulsation caused by the structure of the valve plate is not considered, the elastic pulsation is the main form of the output flow pulsation of the constant pressure variable pump. In order to explore the relationship between q_s and q_x , under the changing mode of sensitive factors in Fig. 11, the simulation diagram of q_x is obtained, as shown in Fig. 12.

(a) Oil effective bulk elastic modulus

(b) Diameter of plunger leak hole

Fig. 12 Simulation diagram of q_x

Compared with Fig. 12 and Fig. 11, the steady-state mean value of q_x is slightly higher than q_s due to the throttling

pressure drop characteristic which caused by the oil passing through the main valve of the load-sensitive proportional multiplex valve (Zhang et al., 2017). According to the principle of fluid pipeline effect, the pipeline damping can improve the system flow pulsation characteristics. Since the load-sensitive proportional multiplexing valves have damping groups and complex internal channels, the flow pulsation of the end actuator should be reduced. However, the machine fluid regulation system inside the valve is affected by the hydraulic power and the spring adjustment pressure pulsation, which makes the compensation special accuracy, is not high. So the simulation results show that q_s does not decrease significantly compared with q_s .

4.2.2 Gas content and oil viscosity

The gas in hydraulic oil is composed of dissolved gas and free gas. It is a two-phase fluid that takes into account both oil and gas properties (Zhou et al., 2016). Under actual working conditions, the components of dissolved and free gases in oil are dynamically transformed with the change of working pressure and ambient temperature. The change of gas content is an important factor of causing the dynamic change of the oil effective bulk elastic modulus. According to Eq. (6), it can be seen that the oil viscosity is another important factor that affects the leakage flow, and then it changes the flow pulsation rate of the constant pressure variable pump.

The oil effective bulk elastic modulus is defined as Eq. (5) and the plunger leakage is defined as Eq. (6) using AMESET tool in AMESIM software. The output flow pulsation rate of the constant pressure variable pump was analyzed under the three mode, which contains reducing 50% and increasing 50% of gas content in 1% and oil viscosity in 46cP.

The simulation diagram is shown in Fig. 13. Based on Fig. 13, the pulsation rate of q_s is obtained, as shown in Table 5.

Fig. 13 Influence of gas content and oil viscosity on q_s

Table 5 Pulsation rate of q_s

Parameter	Time	-50%	1%	+50%
Gas content	0-0.75s	6.55%	8.22%	8.66%
	0.75-1.5s	5.36%	6.90%	7.44%
Parameter	Time	-50%	46cP	+50%
Oil viscosity	0-0.75s	8.32%	7.96%	7.43%
	0.75-1.5s	7.01%	6.69%	6.27%

From Fig. 13(a) and Table 5, it can be seen that when the gas content is reduced by 50%, the pulsation rate of q_s decreases by 1.67%; and when the gas content is increased by 50%, the pulsation rate of q_s increases by 0.44%.

From Fig. 13(b) and Table 5, it can be also seen that when the oil viscosity is reduced by 50%, the pulsation rate of q_s increases by 0.36%, and when the oil viscosity is increased by 50%, the pulsation rate of q_s decreases by 0.53%. Comparing the influence of gas content and oil viscosity on the pulsation rate of q_s under the same load and dynamic change mode, the weight coefficient of the former to the pulsation rate of q_s is larger than that of the latter.

The simulation results furtherly prove that the main form of output flow pulsation of the constant pressure variable pump is elastic pulsation.

5. Analysis of experimental results

The electro-hydraulic robot system experimental platform is built, as shown in Fig. 14.

Fig. 14 Experimental platform

1) console; 2) constant pressure variable pump; 3) pressure transducer; 4) flow sensor; 5) electro-hydraulic robot

Experimental platform illustrates some measure and control tools. The pressure transducer collects the out pressure data of the end actuator; the flow sensor collects the out flow data of the end actuator; the console receive data of the pressure transducer and the flow sensor, and control the constant pressure variable pump and the load-sensitive proportional multiplex valve. The experimental scheme is as follows: the dynamic characteristics of the robot are verified by measuring the output pressure-flow of the end actuator; the relationship between output pressure-flow of the robot and the constant pressure variable pump is verified; the output pressure-flow of the constant pressure variable pump under different working conditions are compared and analyzed.

The load quantity of the robot is realized by installing different weights on the end actuator. The core displacement of the load-sensitive proportional multiplex valve is controlled by the current signal. According to the experimental scheme, three loading modes are designed, as follows in Table 6.

Table 6 Load of end actuator

		Load	current	Load	current	Load	current
Change mode 1	Time	0-10s		10-15s		15-17.5s	
	Parameter	3kg	100mA	1.5kg	200mA	3kg	100mA
Change mode 2	Time	0-5.5s		5.5-8.5s		8.5-10s	
	Parameter	5kg	150mA	12kg	150mA	7kg	150mA
Change mode 3	Time	0-5.5s		5.5-8.5s		8.5-10s	
	Parameter	5kg	150mA	17kg	150mA	7kg	150mA

(1) When the loading mode is changed as the mode 1 in Table 6, the pulsation characteristics of the end actuator are obtained, as shown in Fig. 15.

Fig. 15 Pulsation characteristics of end actuator

From the comparative analysis of Fig. 15 and Fig. 10, it can be seen that the pulsation quantity of p_x and q_x is greater, and it is due to the local loss caused by the complex flow passage and the non-smooth transition of the phase flow passage of the hydraulic oil through the load-sensitive proportional multiplex valve. In Fig. 15, the flow pulsation rate of the end actuator is 15.23% smaller than the pressure pulsation rate of 20.13%. In 10-15s stage, the load pressure is relatively small and the pulsation rate of p_x decreases from 15.23% to 13.16%.

(2) When the loading modes are changed as the mode 2 and 3 in Table 6, the pulsation characteristics of the constant pressure variable pump under two different load modes are compared and analyzed, as shown in Fig. 16. When the load quantity increases, the flow fluctuation rate increases from 14.32% to 15.16%, and the pressure fluctuation rate increases from 15.62% to 16.81%.

Fig. 16 Pulsation characteristic chart of constant pressure variable pump

As it can be seen from Fig. 16, in 5.5-8.5s stage, the flow and pressure pulsations rate in Fig.16(a) are smaller than those in Fig.16(b), because the load quantity in Fig. 16(a) is smaller than that in Fig. 16(b). In Fig. 16(b), the output flow of the constant pressure variable pump tends to decrease in 5.5-8.5s stage when the load quantity of the robot increases. Because the constant pressure variable displacement pump has not reached the constant pressure condition, but it is close to the setting pressure (Kemmetmüller et al., 2010). Therefore, the inclination angle of the inclined plate begins to change and q_s decreases.

(3) When the loading mode is changed as the mode 3 in Table 6, the output pressure-flow of the constant pressure variable pump are obtained when the pre-tightening force of the constant pressure valve is reduced by adjusting the adjustment nut, as shown in Fig. 17.

Fig. 17 Pulsation characteristics of constant pressure variable pump with reduced pre-tightening force of constant pressure valve

Comparing the results of Fig. 17 and Fig. 16(b), it can be concluded that when the load quantity of the robot changes in a certain way, the decrease of the spring preload of the constant pressure valve leads to decrease of the setting pressure of the constant pressure variable pump.

6. Conclusions

In this paper, the pulsation characteristics of electro-hydraulic robots is studied. The following conclusions are obtained.

(1) By comparing the simulation results with the experimental results, it can be concluded that the higher the load pressure of the robot, the greater the fluctuation rate of the output pressure and flow rate of the constant pressure variable pump.

(2) By analyzing the simulation results and experimental results of the flow pulsation at the constant pressure variable pump and the end actuator, it can be concluded that although the load-sensitive proportional multiplex valve has damping groups and pipelines, the pulsation rate of q_x is not much better than the pulsation rate of q_s due to the relatively complex internal structure of the load-sensitive proportional multiplex valve.

(3) When the geometric pulsation caused by the structure of the valve plate is not considered, the elastic pulsation has a larger weight than the leakage pulsation, and it is the main pulsation form of the output flow of the constant pressure variable pump.

(4) When the ambient temperature and load quantity change, the weight of gas content on pulsation rate of q_s is greater than that of oil viscosity.

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