Simulation Study on the Power of the Load Sensitive Proportional Variable Pump

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Abstract. Regarding the study of load sensitive proportional variable pump power, this article constructs a mathematical model of the variable pump and its control mechanism to obtain its key parameters. Using the Amesim platform to control the key parameters obtained from the mathematical model for simulation analysis, and finally comparing the output power simulation curves of the proportional variable pump system and the load sensitive variable pump system, it is found that the load sensitive variable pump has lower power when completing the same task.

Keywords: variable pump; load sensitive; simulation study

1 Introduction

The study of power characteristics of load sensitive proportional variable pumps is a hot topic in the field of hydraulic pump technology. It not only helps to deepen the understanding of the performance of pumps under different operating conditions but also provides a solid theoretical basis for the optimization design and performance improvement of load sensitive proportional variable pumps. Through this study, the working mechanism of the pump can be revealed. Its behavioral characteristics under different loads and flow demands can be grasped, providing strong support for the efficient operation of the pump in practical applications.

In order to further study the characteristics of the variable displacement pump, this article decomposes it into three modules based on its functional structure: the inclined disc axial piston pump, the inclined disc angle adjustment cylinder, and the inclined disc angle adjustment proportional valve. These three modules each undertake different functions and together determine the overall performance of the pump. We have established mathematical models for these three modules to accurately describe their working processes and key parameters.

On the basis of constructing a mathematical model, this article successfully built a simulation model of a load sensitive proportional variable pump using the Amesim simulation platform. Through this simulation model, it is possible to simulate the operation of the pump under different working conditions and visually observe its core performance indicators, such as power output and flow rate changes. This simulation research method not only has the characteristics of high efficiency and accuracy but also can predict and verify the performance of the pump before practical application, thereby greatly reducing development costs and risks [1, 2].

2 Mathematical Model of Variable Pump

The variable displacement pump is roughly composed of three parts [3], namely the inclined disc axial piston pump 1, the inclined disc angle adjustment cylinder 2, and the inclined disc angle adjustment proportional valve 3, as shown in Figure 1.

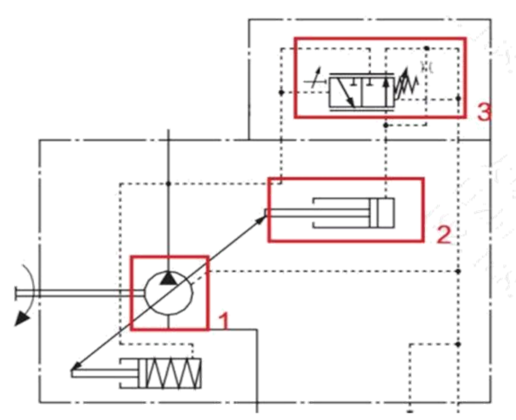

Fig. 1. Control principle of variable pump.

2.1 The Mathematical Model of the Proportional Valve for Adjusting the Angle of the Inclined Plate

A proportional electromagnet converts the current signal input from a proportional control amplifier into force. The force acting on the valve core is proportional to the input current signal. The larger the input current is, the greater the force acts on the valve core. The magnitude of the force is expressed as:

$$\mathrm{F}_1 = \mathrm{K}_\mathrm{i}\mathrm{I} \tag{1}$$

In the formula:

F_1—Force exerted on the valve core (N).

K_i—Thrust gain of a proportional electromagnet (N/A).

I—Input current (A).

According to the force balance of the proportional valve, it can be expressed as:

$$F_2 = m_v \frac{d^2 x_v}{dt^2} + B_v \frac{dx_v}{dt} + K_v x_v \tag{2}$$

In the formula:

m_v—Mass of the valve core of the proportional valve (kg).

x_v—Valve core opening degree (m).

B_v—Damping coefficient generated by transient liquid flow (N · m/s).

K_v—Elastic coefficient of liquid flow force (N/m).

2.2 The Mathematical Model of the Inclined Plate Angle Adjustment Oil Cylinder

As shown in Figure 2, the inclined plate angle adjustment cylinder is connected to the inclined plate angle adjustment proportional valve at both ends. Its force state is shown in the figure:

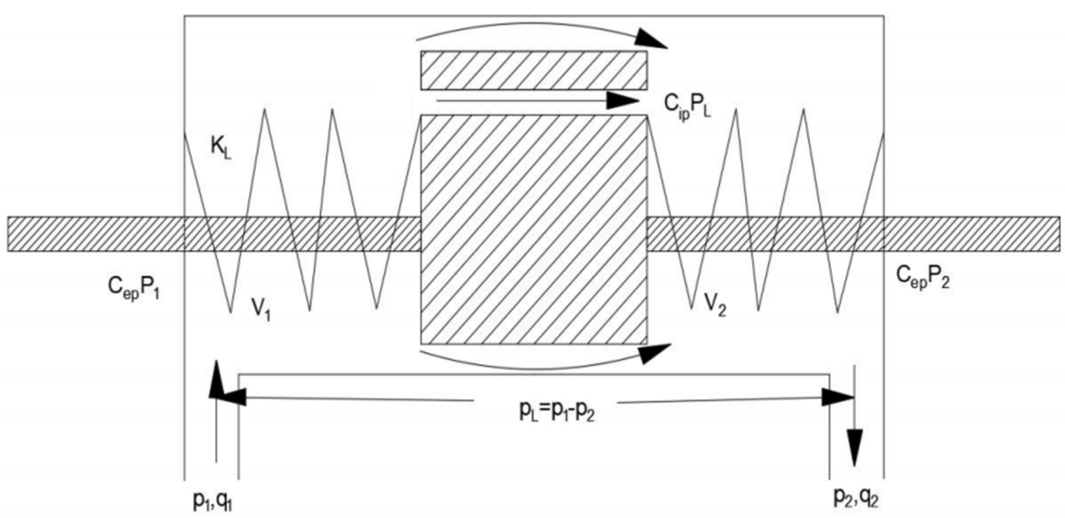

Fig. 2. Tilt plate angle adjustment cylinder.

The expression for the flow rate at its inlet is:

$$q_1 = C_{ep}P_1 + C_{ip}P_L + q_t + \frac{dV_1}{dt} + \frac{V_1}{\beta t}\frac{dP_1}{dt} \tag{3}$$

The expression for the flow rate of its return oil chamber is:

$$q_2 = C_{ip}P_L - C_{ep}P_L + q_t - \frac{dV_2}{dt} - \frac{V_2}{\beta t}\frac{dP_2}{dt} \tag{4}$$

In the formula:
q_1—Flow rate at the inlet of the oil cylinder (L/min).
q_2—Flow rate of oil cylinder outlet (L/min).
C_{ep}—Leakage coefficient (m^3 · s-1/MPa).
C_{ip}—Internal leakage coefficient (m^3 · s-1/MPa).
q_t—Flow rate through the orifice (L/min).
P_1—Pressure at the inlet of the oil cylinder (MPa).
P_2—Pressure at the outlet of the oil cylinder (MPa).
t—Effective volume modulus of hydraulic oil.
V_1—Left volume of oil cylinder (m^3).
V_2—Right side volume of the oil cylinder (m^3).

By combining the flow equation and the force balance equation of the oil cylinder, the transfer function of the inclined plate angle adjustment oil cylinder can be obtained:

$$x_c = \frac{4\mu\beta_\tau A_c K_q K_i I - [4\beta_\tau(C_p - \mu K_c) + \mu V_c s](m_v s^2 + B_v s + K_v)F_c}{[4\beta_\tau(C_p + \mu K_c) + \mu V_c s][(m_a s^2 + B_p S + K_p) + 4\mu\beta_\tau A_c K_q s](m_v s^2 + B_v s + K_v)} \tag{5}$$

2.3 Mathematical Model of Variable Pump

The relationship between the piston displacement of the oil cylinder and the inclination angle of the pump swash plate can be written as:

$$x_c = L_c \sin\alpha \tag{6}$$

In the formula:

x_c—Displacement of the oil cylinder piston (m).

L_c—The distance (m) from the point of force exerted by the oil cylinder on the inclined plate to the center of the inclined plate when the angle of the variable pump inclined plate is zero.

α—The inclination angle of the inclined plate of the pump (m).

In the relationship between the valve-controlled cylinder, there is a vibration link. The natural frequency of the vibration link is:

$$\omega_c = \sqrt{\frac{4\beta_\tau A_p^2}{V_P m_a}} \tag{7}$$

In the formula: *Ap*—the area per unit angle of the variable displacement pump plunger (m^2/rad).

When the angle $\frac{\pi}{2} + x$ between the distribution plate and the inclined plate shaft is, the force analysis of the variable displacement pump shows that the relationship between the force exerted by the oil cylinder on the inclined plate can be written as:

$$F_P = \frac{2RA_pP_s \sin x - \pi R^2 m_a \sin\alpha\omega_p^2\alpha}{L_c} \tag{8}$$

In the formula:

R—When the inclination angle of the inclined plate is zero, it is the distance between the variable pump slide shoe and the main shaft on the projection of the inclined plate (m).

p—Speed of variable displacement pump (rad/s).

P_s—Pressure difference of variable displacement pump (kg).

Substituting Equations (6) and (8) into Equation (5), the functional expression for the displacement of the variable pump inclined plate is:

$$\alpha = \frac{4\mu\beta_\tau A_c K_q K_i I - 4\beta_\tau(C_p+\mu K_c)(m_v s^2 + B_v s + K_v)2RA_pP_s \sin x}{4\beta_\tau(m_v s^2 + B_v s + K_v)[\mu A_c^2 L_c^2 s - \pi R^2 m_a \omega_p^2(C_p+\mu K_c)]} \tag{9}$$

3 Simulation Modeling

Using the hydraulic and HCD libraries of AMEsim software, models of various components of the variable displacement pump are established for simulation modeling. Based on component selection and design, simulation parameters are set, and then simulation analysis is conducted. The traditional quantitative pump load sensitive system has flow loss, but the system stability is better. The traditional load sensitive quantitative pump system can automatically adjust the output flow rate, but the system stability is poor. The system uses a proportional variable pump as the power source. When the input signal remains constant, the variable pump is equivalent to a quantitative pump. In order to highlight that the proportional variable pump load sensitive system is more stable and energy-saving than these two traditional load sensitive systems, this section conducts Amesim modeling on the proportional variable pump, and Amesim modeling on the load sensitive variable pump and the quantitative pump. Finally, a comparative analysis is conducted [4, 5].

3.1 Quantitative Pump Model

In order to compare the energy-saving performance with the load sensitive system of the proportional variable pump, an Amesim simulation model of the load sensitive quantitative pump was established, as shown in Figure 3. This model is established through the fixed displacement pump model PU001 sub-model. The displacement of the quantitative pump is 71 mL/r, with a mechanical efficiency of 0.95 and a volumetric efficiency of 0.95.

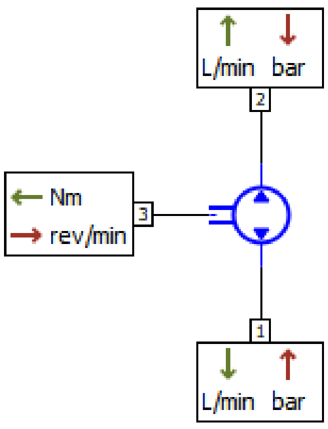

Fig. 3. Quantitative pump model.

3.2 Load Sensitive Variable Pump Model

The load sensitive variable pump model mainly consists of a variable pump, a load sensitive valve, a constant pressure valve, and a variable cylinder. When the variable pump is in the regulating state, the load pressure is fed back to the spring chamber of the load sensitive valve, which is connected to the variable cylinder, thereby changing the displacement of the variable pump by controlling the piston displacement of the variable cylinder. When the system pressure is higher than the set pressure of the constant pressure valve spring chamber, it will push the piston of the constant pressure valve to move and close the connection channel between the load sensitive valve and the variable cylinder, thereby reducing the tilt angle of the variable pump to near zero by controlling the piston displacement of the variable cylinder. The Amesim simulation model of the load sensitive valve is shown in Figure 4.

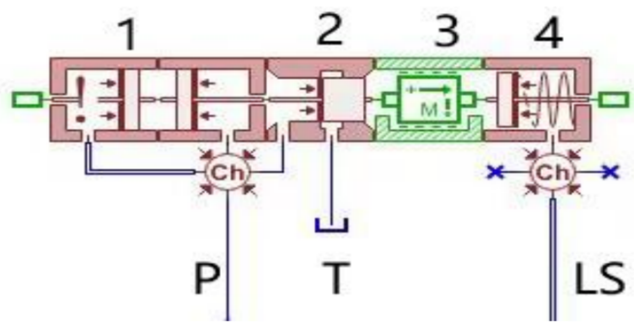

Fig. 4. Load sensitive valve model.

The parameters in the Amesim simulation model of the load sensitive valve are shown in Table 1.

Table 1. Simulation parameter settings for load sensitive valves.

Number	Parameter	Setting value
1	Valve stem outer diameter	10mm
	Valve stem bottom diameter	6mm
2	Valve stem outer diameter	10mm
	Valve stem bottom diameter	6mm
3	Opening amount at zero displacement	-6mm
	Viscous friction coefficient	0.5N/(m/s)
4	The lower limit of displacement	0
	Displacement upper limit	15mm
5	Spring Set Force	125N

The parameters in the Amesim simulation model of the load sensitive valve are shown in Table 2 below.

Table 2. Simulation parameter settings for constant pressure valve.

Number	Parameter	Setting value
1	Valve stem outer diameter	10mm
	Valve stem bottom diameter	6mm
2	Valve stem outer diameter	10mm
	Valve stem bottom diameter	6mm
3	Opening amount at zero displacement	-8mm
	Viscous friction coefficient	0.1N/(m/s)
4	The lower limit of displacement	0
	Displacement upper limit	15mm
5	Spring Set Force	1250N

The Amesim simulation model of the variable cylinder is shown in Figure 5.

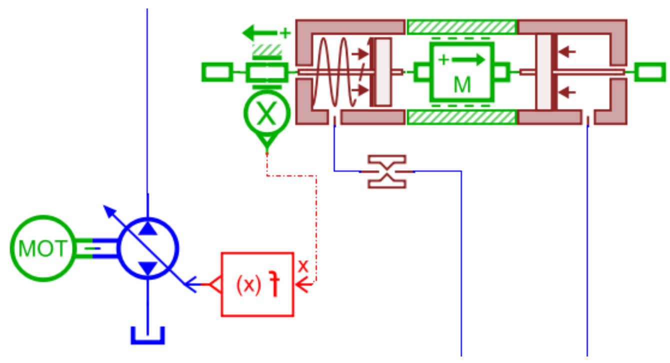

Fig. 5. Variable cylinder model.

The parameters in the Amesim simulation model of the variable cylinder are shown in Table 3.

Table 3. Parameter settings for variable cylinder simulation.

Number	Parameter	Setting value
1	The outer diameter of the valve stem	10mm
	The bottom diameter of the valve stem	6mm
2	Spring setting force	5 N
	Coefficient of viscous friction	0.1N/(m/s)
	The upper limit of displacement	15mm
3	The outer diameter of the valve stem	10mm
	The bottom diameter of the valve stem	0mm

The simulation model of the load sensitive variable pump is shown in Figure 6, where 1 is the load sensitive valve, 2 is the constant pressure valve, and 3 is the variable cylinder.

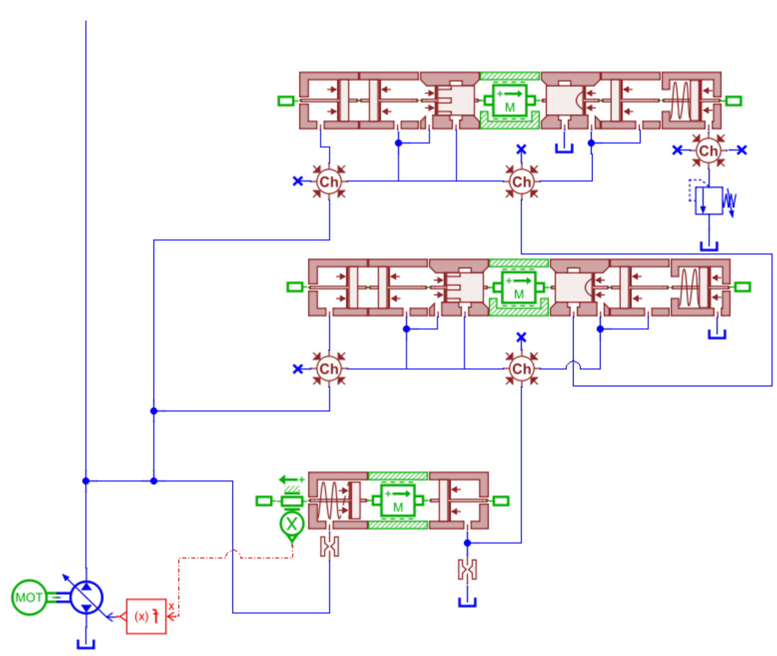

Fig. 6. Load sensitive variable pump model.

4 Simulation Analysis

As shown in Figure 7, the simulation time is 30 seconds, with 0 load for 0-10s, no load for 10-20 seconds, and 6 kW load for 10-20 seconds. There is a comparison of output power simulation curves between the proportional variable pump system and load sensitive variable pump system when no signal is given to the hydraulic cylinder circuit.

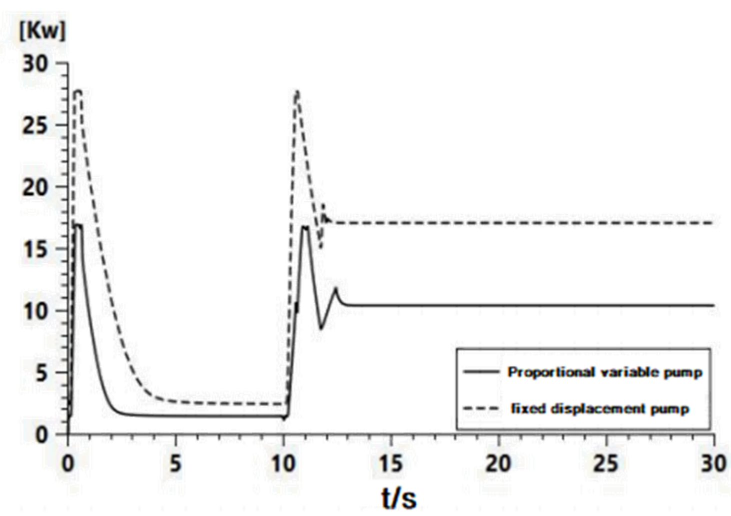

Fig. 7. Comparison of simulation results.

From the figure 8 and figure 9, it can be seen that due to the pressure shock generated by the system at the beginning of the operation, the power curve will fluctuate at the beginning. After the system stabilizes, the output power of the proportional variable pump system under no load is 1.2 kW, and the output power of the quantitative pump load sensitive system is 2.5 kW. Starting from 10 seconds, the power curve of the system fluctuated again due to the sudden load. After the system stabilized, the output power of the proportional variable pump system was 10.4 kW during the sudden load, and the output power of the quantitative pump load sensitive system was 17 kW. Therefore, choosing a proportional variable pump as the power source makes the hydraulic power system more energy-efficient.

To investigate the specific impact of load power changes on pump output pressure, this paper designed a series of simulation experiments. In the experiment, we simulated four different load conditions: under power, 20-second full power, 40-second full power, and 60-second full power. We tested the pressure changes of variable displacement pumps and quantitative pumps under these conditions. The experimental results show that with the increase of load power, the output pressure of both pumps shows an upward trend, but the variable displacement pump exhibits a more flexible and stable pressure regulation ability when dealing with load changes. This discovery further confirms the crucial role of pressure parameters in power variation, providing an important theoretical basis for optimizing the stability and energy efficiency of hydraulic systems in the future.

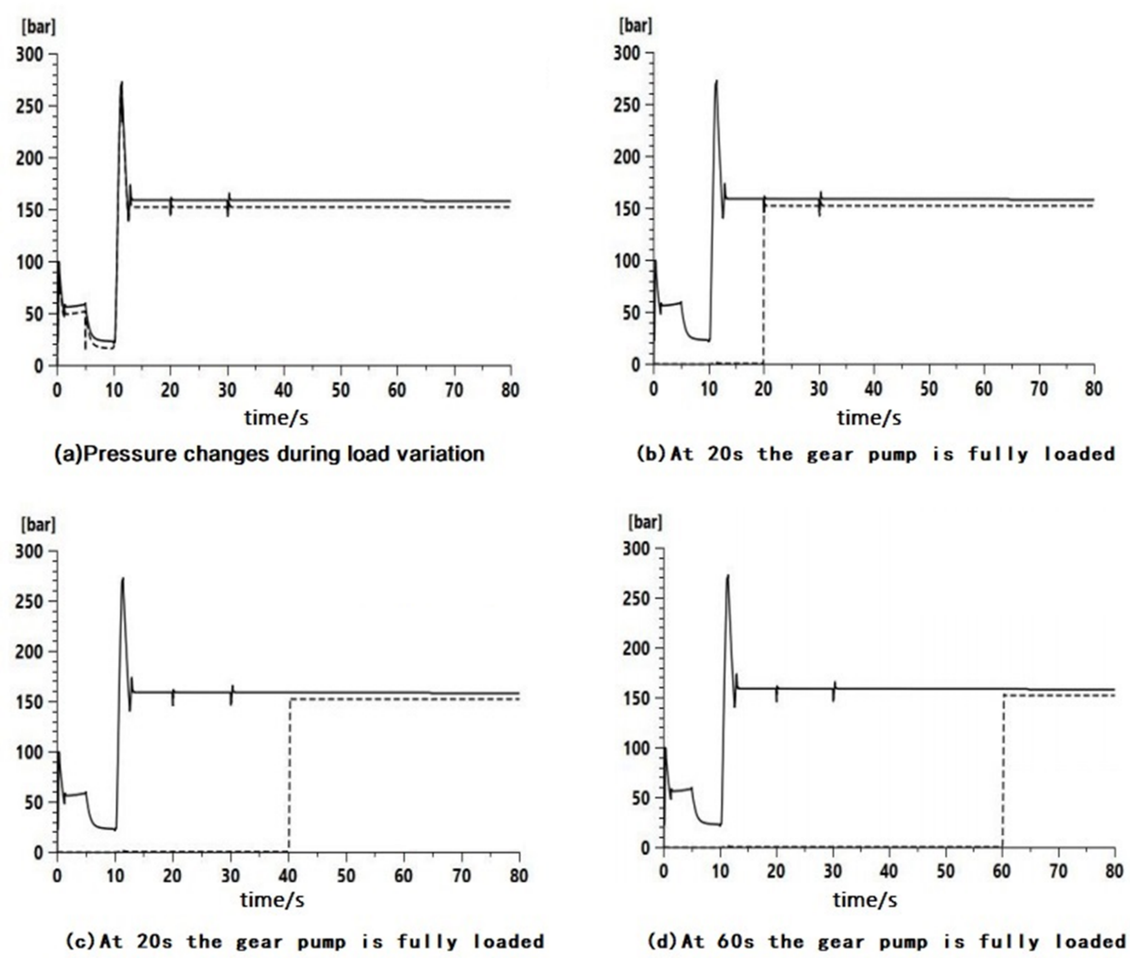

Fig. 8. Simulation of the effect of load on pump pressure.

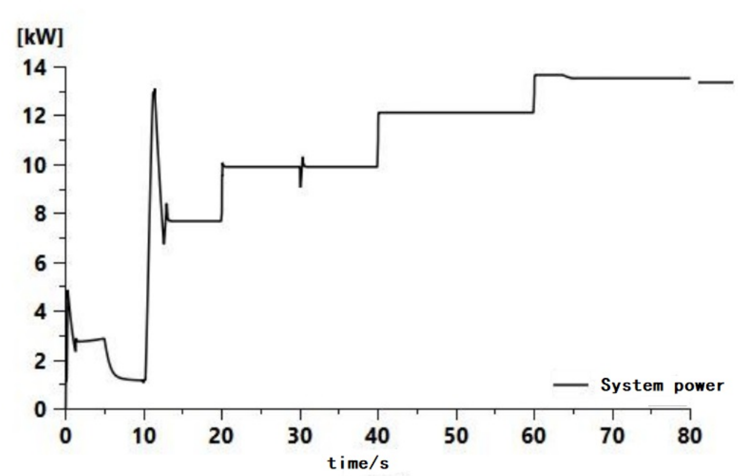

Fig. 9. Simulation of local power characteristics of load sensitive variables.

To verify the power characteristics of the load sensitive variable, this article states that after the system stabilizes, the hydraulic motor of the generator works independently from 0 to 20 seconds. The output power of the proportional variable pump system under no load is 1.2 kW. At 10 seconds, due to the sudden increase in load, the power curve of the system fluctuated again. After stabilization, the output power of the proportional variable pump system was 7.6 kW. Starting from 20s, the oil pump motor also starts working. After stabilization, the output power of the proportional variable pump system is 10.7 kW. At around 40s, the attitude adjustment hydraulic cylinder starts working. After stabilization, the output power of the proportional variable pump system is 12 kW. At around 60s, the attitude adjustment hydraulic cylinder started working. After stabilization, the output power of the proportional variable pump system was 13.3 kW.

5 Conclusion

This article delves into the functional characteristics of load sensitive pumps, innovatively deconstructing their functions into three core parts and establishing an accurate mathematical model based on them. This modeling process not only deepens the understanding of the operating mechanism of load sensitive pumps but also accurately identifies key parameters that affect their performance.

To further validate the practicality of the theoretical model, this paper successfully constructed a simulation model of a load sensitive proportional variable pump based on the Amesim simulation platform. Through simulation, it can be observed intuitively that the pump can flexibly and accurately adjust the output flow rate and pressure according to the actual needs of the system, effectively reducing power loss and significantly improving work efficiency. Its high flexibility ensures stable and reliable performance under different working conditions, demonstrating excellent adaptability.

Of particular note is that the load sensitive proportional variable pump achieves precise control of output, minimizing unnecessary energy consumption and demonstrating astonishing energy efficiency performance. This feature not only conforms to the current trend of green and environmental protection but also brings unprecedented power advantages to hydraulic systems, marking a dual leap in efficiency and environmental performance of hydraulic systems.

This article provides a thorough analysis of the working principle of load-sensitive proportional variable pumps from a theoretical viewpoint. It also demonstrates their significant role in enhancing hydraulic system efficiency, ensuring operational stability, and promoting sustainable development. By establishing and verifying simulation models, the article offers substantial theoretical and practical support for future innovations in hydraulic technology.

Acknowledgments

This article expresses gratitude for the support of the key project 2023-2-TD-ZD010 "Research and Development of Mining Electro Hydraulic Servo Valve and Controller" funded by Tian Di Technology Innovation and Entrepreneurship Fund.

References

1. Zhang Yiliang. Non highway mining dump truck line controlled hydraulic steering system based on electro-hydraulic proportional and load sensitive technology [J]. Hydraulic, Pneumatic and Sealing, 2024, 44 (07): 118-123.
2. Deng Lin, Dong Huairong. Integrated Control Technology of Hydraulic Power Based on Load Sensitivity [J]. Chinese and Foreign Energy, 2023, 28 (S1): 71-75.
3. Qi Zhentao. Design and Simulation of Vehicle mounted Hydraulic Power System Based on Load Sensitivity Principle [D]. Tianjin University of Technology, 2023.
4. Wang Renhao, Wei Yonghe, Yang Xing. Performance Study of Pressure Refueling Control Valve Based on AMESim [J]. Mechanical Engineering and Automation, 2024, (04): 47-49.
5. Fang Min, Hong Wei, Tang Bo. Dynamic Characteristics Analysis of Hydraulic Control Valve for Submersible Pump Based on AMESim [J]. Mechanical and Electrical Engineering Technology, 2024, 53 (06): 166-169+200.