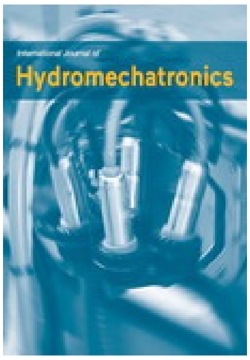
Linking academia, business and industry through research



International Journal of Hydromechatronics

ISSN online: 2515-0472 - ISSN print: 2515-0464
https://www.inderscience.com/ijhm

Position servo with variable speed pump-controlled cylinder: design, modelling and experimental investigation

Haigang Ding, Yalong Li, Quanmin Zhu, Jingyi Su

DOI: 10.1504/IJHM.2024.10063432

Article History:

Received:	24 August 2023
Last revised:	05 October 2023
Accepted:	07 October 2023
Published online:	30 April 2024

Position servo with variable speed pump-controlled cylinder: design, modelling and experimental investigation

Haigang Ding*

School of Mechatronic Engineering,
China University of Mining and Technology,
Xuzhou, Jiangsu, China
and
State Key Laboratory of Intelligent Mining Equipment Technology,
Xuzhou, Jiangsu, China
Email: hgding@cumt.edu.cn
*Corresponding author

Yalong Li

School of Mechatronic Engineering,
China University of Mining and Technology,
Xuzhou, Jiangsu, China
Email: ts22050145p31@cumt.edu.cn

Quanmin Zhu

Department of Engineering Design and Mathematics,
University of the West of England,
Bristol 161QY, UK
Email: Quan.Zhu@uwe.ac.uk

Jingyi Su

School of Mechatronic Engineering,
China University of Mining and Technology,
Xuzhou, Jiangsu, China
Email: 2337251182@qq.com

Abstract: This paper proposes a position servo system with variable-speed pump-controlled cylinder (VSPC), which controls an asymmetric cylinder in four quadrants by adjusting servomotor speed, and elucidates the system working principle. The system mathematical model is established, an anti-integral saturation PID controller is designed, and the resonance frequency and damping ratio are obtained. The system simulation model and experimental bench are established, and step response and sinusoidal tracking experiments are carried out, revealing the change rule of servomotor speed and torque, and analysing the gas volume and pressure change process of accumulators. The results show that the actuator can accurately and quickly track the command signal under positive and negative loads by regulating the pump speed; when the cylinder position is unchanged, the motor operates at low speed with a fixed steering to compensate for system leakage and maintain pressure to keep the desired position.

Keywords: variable speed pump control; variable-speed pump-controlled cylinder; VSPC; position servo; resonance frequency; four quadrants; servomotor.

Reference to this paper should be made as follows: Ding, H., Li, Y., Zhu, Q. and Su, J. (2024) 'Position servo with variable speed pump-controlled cylinder: design, modelling and experimental investigation', *Int. J. Hydromechatronics*, Vol. 7, No. 2, pp.155–175.

Biographical notes: Haigang Ding has been an Associate Professor at the China University of Mining and Technology (CUMT), and the header of Electro-Hydraulic Control Lab of CUMT. His research interests include electro-hydraulic transmission and control, and novel hydraulic components.

Yalong Li has been a graduate student in Mechanical Engineering at CUMT. His research interests are in electrohydraulic servo control.

Quanmin Zhu has been a Professor with the Department of Engineering Design and Mathematics, University of the West of England, Bristol, UK. His main research interests include nonlinear system modelling, identification, and control.

Jingyi Su has been a graduate student in Mechanical Engineering at CUMT. His research interests are in electrohydraulic servo control.

1 Introduction

Traditional pump control systems use variable displacement control, i.e., a fixed speed motor drive that regulates the system flow by changing the pump displacement. This wastes energy because the motor keeps rotating at high speed and requires an expensive and complex variable structure. With the popularity of servo motors, variable speed pump control becomes a new technology of hydraulic transmission and control, compared with variable pump control, it has the following advantages: high energy efficiency, easy to control, simple structure, high reliability (Caliskan et al., 2015; Rahmfeld and Ivantysynova, 2000). A typical application of variable speed pump control technology is the direct control of hydraulic cylinders, i.e., a variable speed pump-controlled (VSPC) cylinder. This pump-controlled direct drive actuator is also commonly referred to as an electro-hydraulic actuator (EHA). The VSPC system mainly consists of a servo motor, a variable speed dosing pump, an asymmetric cylinder and a measurement and control system, which controls the direction of movement, position, speed and output force of the hydraulic cylinder by adjusting the motor speed. In contrast to conventional hydraulic systems, in VSPC systems the output power is controlled by a variable speed pump, changing the speed of the pump means changing the flow or pressure of the fluid. Variable speed drive combines the advantages of hydraulics with the advantages of servo drives and consume 80% less energy than conventional hydraulic systems (Sarkar et al., 2023). Some well-known hydraulic companies have introduced VSPC products, such as

Rexroth's servo hydraulic actuator (SHA), Moog's electro-hydrostatic actuator (EAS), Voith's DrivAx CLDP and RQ4. These variable speed pumping systems are mainly used in industrial hydraulics, such as aircraft, presses and bending machines (Wang et al., 2020; Waheed et al., 2016). The main advantage of variable speed pump control is its high efficiency, and some researchers at home and abroad have tried to use this technology in construction machinery to improve its energy efficiency (Hippalgaonkar and Ivantysynova, 2013; Chen et al., 2019; Lin et al., 2020, 2021). Williamson et al. (2008) pointed out that pump controlled linear actuator with differential cylinder is a way to save primary energy in mobile machines. Danaee et al. (2018) proposed utilisation of direct position control in an electro-hydraulic system with a new hydraulic zonal system architecture implemented with direct driven hydraulics.

The VSPC system can effectively reduce the energy consumption of heavy-duty robotic arms by recovering and utilising the gravitational potential energy of the robotic arm (Chen and Zhao, 2017). Quan et al. (2014) reviewed energy efficient direct pump controlled cylinder electro-hydraulic technology, proposed two principles of potential energy recovery and utilisation with asymmetric pump-controlled asymmetric cylinder (Wang et al., 2020) and VSPC three-chamber cylinder (Hao et al., 2019), and applied to loaders and excavators. The proposed principle can balance the asymmetric flow caused by the single-outlet hydraulic cylinder and achieve the recovery of gravitational potential energy of the moving arm, which can reduce the system energy consumption by up to 48% compared to the original valve-controlled system. To improve the efficiency of construction machinery driven by pure electric power, Fu et al. (2020) proposed a variable-pressure differential control strategy based on variable-speed control for a quantitative pump load-sensing system. Zhang et al. (2019, 2020) investigated the performance of two VSPC systems for asymmetric cylinders under a four-quadrant operating condition, Quan et al. (2004) studied the principle of closed loop control differential cylinder with double speed variable pumps. Wang and Leaney (2020) analysed energy efficiency of a hybrid pump-controlled asymmetric cylinder drive system. to achieve precision motion control, Kilic et al. (2014) proposed a pressure prediction method using structured recurrent neural networks for a variable-speed pump controlled hydraulic system; Helian et al. (2020, 2021) proposed an adaptive robust control with a backstepping design for the cylinder actuator is direct-driven by a servomotor pump.

VSPC is a current research hotspot, and the current research mainly focuses on the dynamic characteristics of hydraulic cylinders, but there is less research on the dynamic characteristics of motors in the control process, and there is a lack of systematic research. This paper designs a system for VSPC, elucidates the system structure and working principle, establishes a simulation model, and builds an experimental bench for experimental verification, focusing on the dynamic characteristics and motion laws of asymmetric cylinders and servomotors during pump-controlled cylinders. This work will provide a technical reference for the design and control of VSPC cylinders.

2 System design of position servo with VSPC

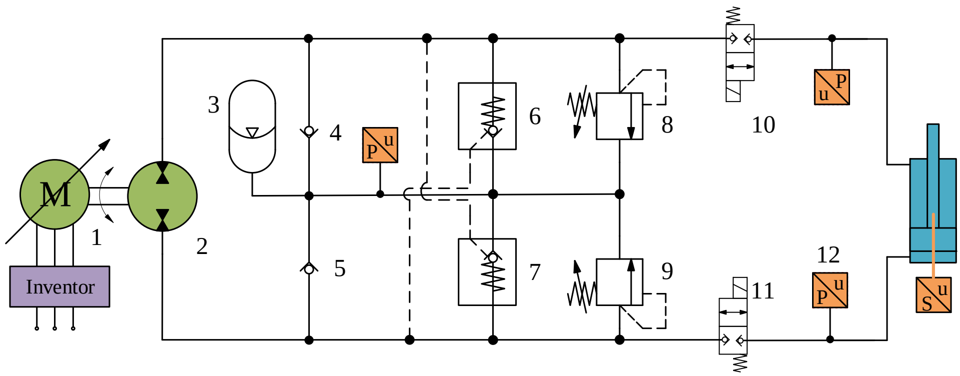
The position servo system with VSPC consists of a servomotor, a fixed displacement pump, an asymmetric cylinder, an accumulator and an auxiliary valve unit, as shown in Figure 1. The auxiliary valve unit is connected in parallel to the closed circuit and consists of a check valve assembly, a pilot operated check valve assembly, a high pressure relief, valve assembly and a shut-off valve assembly. The positive displacement pump is driven by a servomotor. By changing the speed of the motor, the flow rate and direction of fluid flow is adjusted, thereby controlling the stroke and direction of movement of the cylinder. The pilot operated check valve group is used to compensate for the flow differential of the asymmetric cylinder, allowing the system to operate in four quadrants. High pressure relief valves protect the system from overflow and shut-off valves are used to lock the cylinder.

As shown in Figure 2, this system is a closed system with an asymmetric cylinder as an actuator, which can work in four quadrants, where *F* represents the external load and V represents the running direction of the hydraulic cylinder. In the first quadrant, load extending working condition, and the third quadrant, load retracting working condition, the servomotor does the work and drives the quantitative pump to rotate, and the pump outputs the high-pressure fluid, thus driving the hydraulic cylinder to overcome the external load. In the second quadrant, beyond the retracted condition, and the fourth quadrant, beyond the extended condition, the external load does work, and the quantitative pump is in the motor state to drive the servo motor to rotate, and the servo motor is in the power generation state, which can recover the potential energy or kinetic energy of the load. During the four quadrants operation, the accumulator is replenished to the low-pressure side of the system through the check valve group and the liquid-controlled check valve group, and the excess low-pressure oil is charged to the accumulator through the pilot operated check valve group to balance the flow of the system. The accumulator is a low-pressure accumulator with a volume of 3 L and a filling pressure of 2.3 bar. In the process of extending and retracting the cylinder, the pressure of the accumulator changes by 1bar, so that the pressure on the low-pressure side of the system remains basically unchanged. The accumulator is a low pressure accumulator with a volume of 3.5 L and a pre-charge pressure of 2.3 bar. As the ram extends and retracts, the accumulator pressure changes by 1 bar, so that the pressure on the low pressure side remains essentially constant.

In a pump-controlled cylinder position system, the requirements for the pump are as follows: the ability of the pump to operate in at least two quadrants, can rotate in bi-directions to change the operation direction of the cylinder by changing the direction of rotation; low-speed stability to ensure positional accuracy when the load is held; and high volumetric and mechanical efficiencies to ensure that the system has a high level of efficiency. Currently, there are relatively few quantitative pumps that can work in four quadrants. Only when there is beyond the load, only the pump is required to be able to work in four quadrants, when the pump is in the hydraulic motor status, and the motor is in the power generation status.

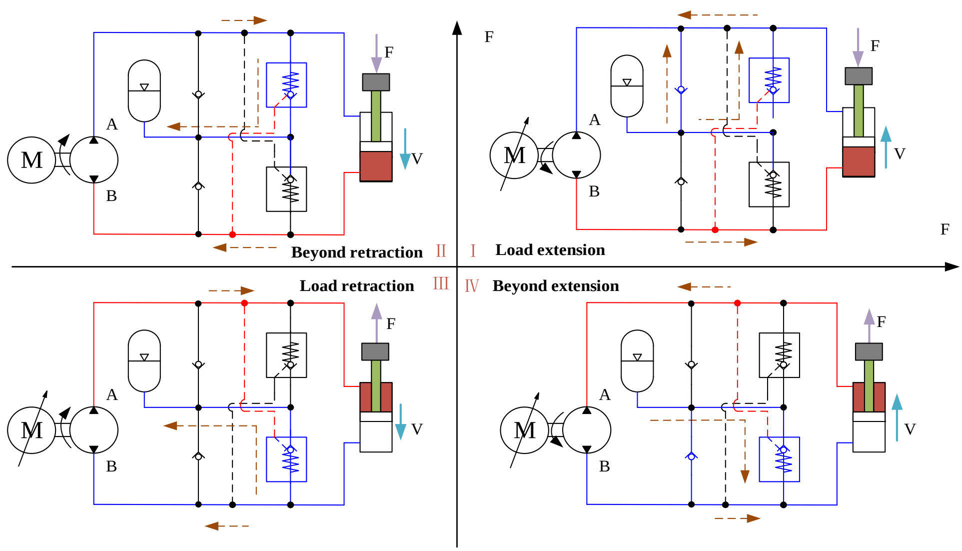
The valve-controlled system has two control chambers, whereas the pump-controlled system has only one control chamber. Therefore, for the same load and cylinder, the natural frequency of the pump-controlled system is 0.707 times that of the valve-controlled system, and because the response of the servomotor or variable pump is slower than that of the control valve, the pump-controlled system will generally respond more slowly than the valve-controlled system.

Figure 1 Hydraulic schematic of position servo with VSPC (see online version for colours)



Notes: 1 – servo motor, 2 – fixed displacement pump, 3 – accumulator, 4, 5 – check valve, 6, 7 – pilot operated check valve, 8, 9 – safety valve, 10, 11 – solenoidal ball valve, 12 – pressure sensor, 13 – asymmetric cylinder, and 14 – stroke sensor.

Figure 2 Hydraulic circuits of VSPC system in four quadrants (see online version for colours)



3 System modelling and parametric analysis

In order to capture the main features of the VSPC system and ignore some minor factors, the following assumptions are made: in the analysis of the hydraulic cylinder continuity equation, it is assumed that the piping connecting the pump and the hydraulic cylinder is symmetrical, and all connecting piping is short and thick, the pressure loss in the piping, the influence of fluid quality and pipeline dynamics can be ignored: the pressure in each working chamber of the hydraulic cylinder is equal everywhere, the oil temperature and bulk modulus of elasticity is constant, the leakage inside and outside the hydraulic cylinder is laminar flow.

This paper establishes mathematical models for the load extension and load retraction conditions, and the overrun extension and overrun retraction conditions are similar and will not be repeated.

3.1 Pump flow equation

Taking into account the internal and external leakage of the pump, the pump actual flow is

$$q_p = nD_p - C_{ip}\left(p_1 - p_2\right) - C_{ep}\,p_1 \tag{1}$$

where n is the servomotor speed, D_p is the pump displacement, p_1 and p_2 are the outlet pressure and inlet pressure of the pump, respectively, C_{ip} and C_{ep} are the internal and external leakage coefficients of the pump, respectively.

The Laplace-transformed flow equation of pump is written as

$$Q_p = ND_p - C_p P_1 \tag{2}$$

where $C_p = C_{ip} + C_{ep}$, is the total leakage coefficient of the pump. Consider a first order transfer function for the servomotor, the servomotor speed is given by

$$N = \frac{fK_u}{1 + s/\omega_m} \tag{3}$$

where f, K_u and ω_m are the input frequency, frequency-speed gain and resonance frequency of servomotors, respectively.

3.2 Flow continuity equation

The flow from the pump to the hydraulic cylinder is not only required to drive the piston movement, but also to compensate for fluid compression and pipeline expansion, as well as to compensate for internal and external leakage of the hydraulic cylinder, that is

$$q_p = A\frac{dx}{dt} + C_{ig}\left(p_1 - p_2\right) + C_{eg}\,p_1 + \frac{V_1}{\beta_e}\frac{dP_1}{dt} \tag{4}$$

where x is the cylinder stroke, C_{ig} and C_{eg} are the internal and external leakage coefficients of the cylinder, respectively, V_1 is the control chamber volume (including the connection pipe and working chamber volume), A is the effective area of cylinder control chamber, β_e is the oil elastic moduli.

The Laplace-transformed flow continuity equation is written as

$$Q_p = AsX + C_g P_1 + \frac{V_1}{\beta_e} sP_1 \tag{5}$$

where $C_g = C_{ig} + C_{eg}$, is the total leakage coefficient of the cylinder.

Combining equations (2) and (5) and with speed input to the servomotor gives

$$ND_p = AsX + \left(C_{pg} + \frac{V_1}{\beta_e} s\right)P_1 \tag{6}$$

where C_{pg} is the total leakage coefficient of VSPC systems.

3.3 *Force balance equation*

Ignoring piston and cylinder friction and other nonlinear loads, as well as the influence of fluid quality, the output force of the hydraulic cylinder is equal to the load force. The load force includes inertia force of moving parts, viscous friction force of moving parts, and other load forces. The force balance equation of the hydraulic cylinder and the load is as follows

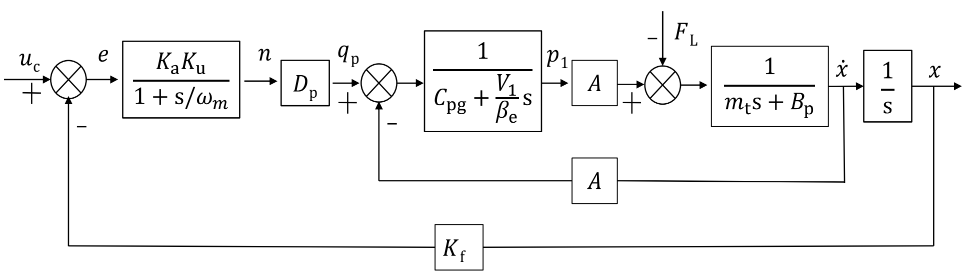
$$p_1 A - p_2 A_r = m_t \frac{d^2 x}{dt^2} + B_p \frac{dx}{dt} + F_L \tag{7}$$

where A_r is the effective area of low-pressure chamber, m_t is the total converted mass, B_p is the viscous damping coefficient, F_L is the load force.

The linearised and Laplace-transformed force equation is written as

$$P_1 A_1 = m_t s^2 X + B_P sX + F_L \tag{8}$$

Figure 3 Block diagram of the position servo with VSPC



3.4 *Control block diagram*

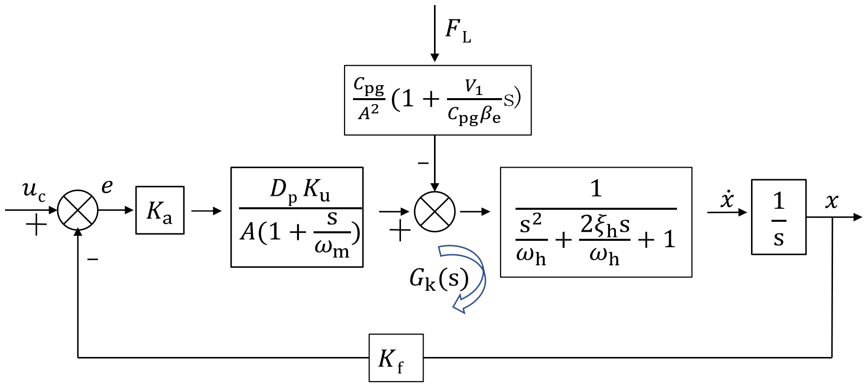
Introducing the position feedback gain K_f, the inventor gain K_a, command signal u_c, the servomotor speed n becomes

$$N\left(U_c - K_f X\right)\frac{K_a K_u}{1+s/\omega_m} \tag{9}$$

By using the equations (6), (8) and (9) a block diagram in closed loop control of the VSPC system is as shown in Figure 3.

The block-diagram in Figure 3 can now be reduced to the following form, shown in Figure 4.

Figure 4 Simplified block diagram of servo with VSPC (see online version for colours)



3.5 Parameter analysis

If the term $\frac{B_p C_{pg}}{A^2} << 1$, the hydraulic resonance frequency ω_h and the damping ratio ξ_h shown in Figure 5 will be expressed as

$$\omega_h = \sqrt{\frac{\beta_e A^2}{V_1 m_t}}, \xi_h = \frac{c_{pg}}{2A}\sqrt{\frac{\beta_e m_t}{V_0}} \tag{10}$$

In order to study the stability of the servo system the open loop gain $G_k(s)$ must be analysed.

$$G_k(s) = \frac{K_a K_u D_p / A}{s\left(1+\frac{s}{\omega_m}\right)\left(\frac{s^2}{\omega_h}+\frac{2\xi_h s}{\omega_h}+1\right)} = \frac{K_v}{s\left(1+\frac{s}{\omega_m}\right)\left(\frac{s^2}{\omega_h}+\frac{2\xi_h s}{\omega_h}+1\right)} \tag{11}$$

where K_v expresses the steady state loop gain and the value of this parameter must be set to a certain level to make sure that the control system will be stable. The control system will be stable if the amplitude margin is positive, which gives the stability criteria as

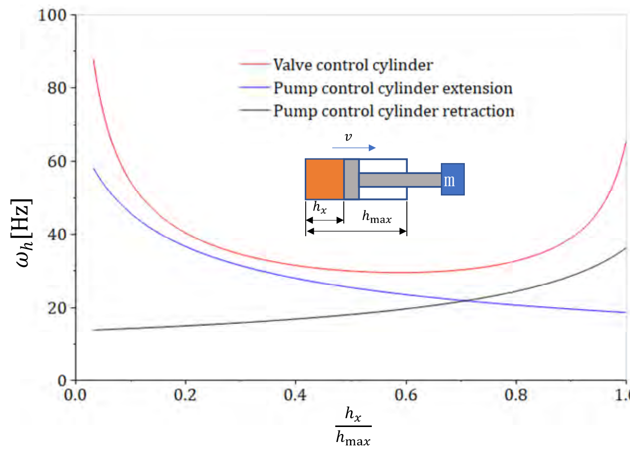
$$K_v \leq 2\omega_h \tag{12}$$

The hydraulic resonance frequency and the damping ratio are the two key factors for a hydraulic control system. Under the same conditions, the resonance frequency and damping ratio of pump-controlled and valve-controlled systems are significantly different as list:

1 The resonance frequency of a VSPC system is $1/\sqrt{2}$ that of a valve-controlled system. Since it has only one control chamber while the valve-controlled system has two control chambers, the hydraulic spring stiffness of the VSPC system is half that of the valve-controlled system, resulting in a lower hydraulic natural frequency of the VSPC system than that of the valve-controlled system.

2 The resonance frequency of the VSPC system is L-shaped, decreasing as the stroke increases, with the lowest natural frequency at the end of the stroke, as shown in Figure 5. The natural frequency of the valve-controlled cylinder system is U-shaped, with the lowest natural frequency at the mid-stroke. For pump controlled cylinder systems, the resonance frequencies of the cylinder extension and retraction phases are different, and the resonance frequency of the extension phase is high due to the fact that the rodless chamber is the control chamber when the cylinder is extended and its effective area is greater than when it is retracted (note that the resonance frequencies in Figure 6 are calculated using the parameters in Table 1 and the cylinders and mass involved are the same for valve and pump control).

Figure 5 Resonance frequency of the position servo system (see online version for colours)



3 The response of VSPC systems is lower than that of valve-controlled systems. This is partly because of the lower resonance frequency of the former, and partly because the response of the servo motor is lower than that of the servo or proportional valve.

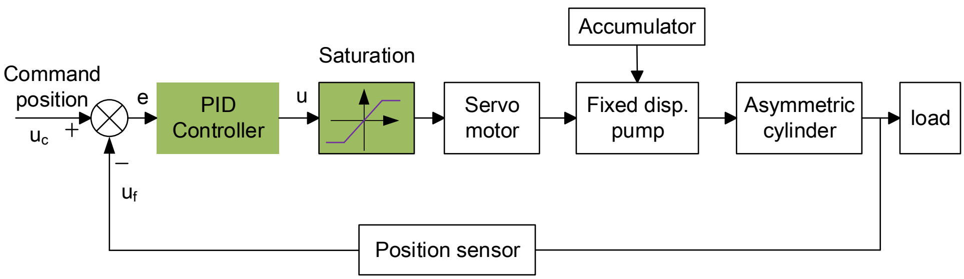
4 The damping ratio of the VSPC system is lower than that of the valve-controlled system, but more stable. Since the total leakage coefficient C_{pg} of VSPC systems is smaller than the total pressure-flow coefficient of valve-controlled systems and is essentially constant, VSPC systems are almost always under-damped, with the under-damping typically being 0.2~0.4. Bypass leakage channels or internal pressure feedback loops are usually used to increase the damping ratio to improve control performance.

5 The gain K_v and damping ratios ξ_h of VSPC systems are more stable than those of valve-controlled systems, so the dynamic characteristics of VSPC systems are less affected by changes in operating point and are easier to predict and control (Lin et al., 2020).

4 Anti-saturation control strategy

Figure 5 shows the control flowchart of the experimental system, which uses a traditional PID controller to control the position of the cylinder in a closed loop. In PID controllers, the function of the integrator is to eliminate the system steady state error and ensure the steady state accuracy of the system, but the integral term can easily fall into saturation, resulting in system overshoot and sluggishness (Chen et al., 2003; Zhu et al., 2023). For controllers containing integral links, it is usually necessary to introduce an anti-integral saturation algorithm.

Figure 6 Pump control cylinder simulation model (see online version for colours)



The output v of the PID controller is of the form

$$v(t) = K_p e(t) + K_i \int_0^t e(\tau)d\tau + K_d \frac{de(t)}{dt} \tag{13}$$

where K_p is the proportional gain, K_i is the integral gain, K_d is the differential gain, τ is the sampling period and e is the error.

The discrete form of equation (13) is

$$v(k) = u_p(k) + u_i(k) + u_d(k) \tag{14}$$

where $u_p(k)$ is the proportional link output, $u_p(k) = K_p e(k)$, $u_i(k)$ is the integral link output, $u_i(k) = u_i(k-1) + \tau K_i e(k)$, and $u_d(k)$ is the differential, $u_d(k) = K_d(e(k) - e(k-1))$.

Integral anti-saturation control method is shown in Figure 8: the integrator part receives a zero signal when the PID output goes outside the saturation limits and the input error e has the same sign as the output v. The intergration is resumed when the PID output lies inside the saturation limits or when the input has opposite sign to the PID output v.

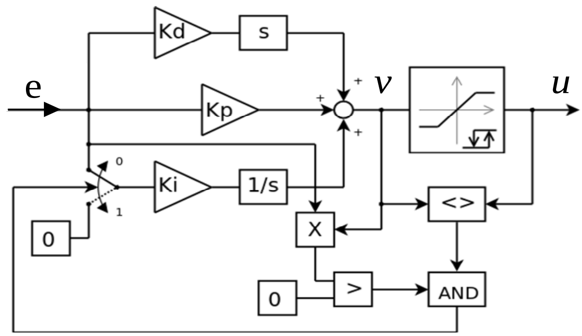
By introducing anti-saturation factor $f(k)$, the integral link output is

$$u_i(k) = u_i(k-1) + \tau K_i e(k) f(k) \tag{15}$$

$$f(k) = \begin{cases} 0, & (v(k-1) \le u_{\min} \vee v(k-1) \ge u_{\max}) \wedge v(k-1)e(k) \ge 0 \\ 1, & \text{else} \end{cases}$$

where $u_{\max}$ and $u_{\min}$ are the maximum and minimum values of the limit, respectively, and $u_{\max} = 10$ and $u_{\min} = -10$.

Figure 7 Integral link anti-saturation control block diagram



After the saturator, the final output $u(k)$ of the PID is expressed as follows

$$u(k) = \begin{cases} u_{\max}, & v(k) \geq u_{\max} \\ u_{\min}, & v(k) \geq u_{\min} \\ v(k), & u_{\min} < v(k) < u_{\max} \end{cases} \tag{16}$$

The gain K_v and damping ratios ξ_h of VSPC systems are more stable than those of valve-controlled systems, so the dynamic characteristics of VSPC systems are less affected by changes in operating point and are easier to predict and control (Lin et al., 2020).

5 Simulation analysis

5.1 Simulation model

Hydraulic systems have strong time-varying nonlinearities that are difficult to model mathematically and accurately. Simulation based on physical modelling is an important tool for quickly understanding the dynamic characteristics of hydraulic systems, with the advantages of high accuracy, low cost and short cycle times, four steps. We have used AMESim to build a simulation model of the proposed VSPC system, as shown in Figure 8, where the cylinder position is controlled by a PID controller with anti-saturation method, and the loads include inertial and elastic loads. Table 1 shows the main parameters of the system.

5.2 Simulation analysis

Figure 9 shows the sinusoidal position tracking curves, which indicate that the cylinder is able to accurately track the desired sinusoidal curve under pump-controlled direct drive, and the position error also varies sinusoidally with a maximum error of about ±5 mm. The output of the PID controller also varied sinusoidally from –6.6 V to 8.7 V with no saturation.

When the cylinder extends and retracts according to the sinusoidal law, the speed of the servomotor changes according to the cosine, first increasing and then decreasing and changing smoothly as shown in Figure 10. When the cylinder is halfway through its stroke, the motor speed reaches a maximum of 2,560 rpm. When the cylinder is at its maximum displacement, the speed of the cylinder is close to zero, but at this time the system pressure reaches a maximum of 289 bar, and the motor must deliver a certain speed (415 rpm) to compensate for the internal leakage of the pump in order to maintain the position of the cylinder. In the cylinder extension stage, the motor torque direction is opposite to the speed direction and the motor is in the driving state. In the retracting stage, the spring does the work, the motor torque direction and speed direction are the same, and the motor is in the power generation working condition. In the retracting stage of the cylinder, during the movement, the rod chamber is a low pressure chamber and its pressure remains basically unchanged at about 3 bar, while the rodless chamber is a high pressure chamber and it varies sinusoidally between 140 bar-300 bar.

Figure 8 Simulation model of VSPC systems based on AMESim platform (see online version for colours)

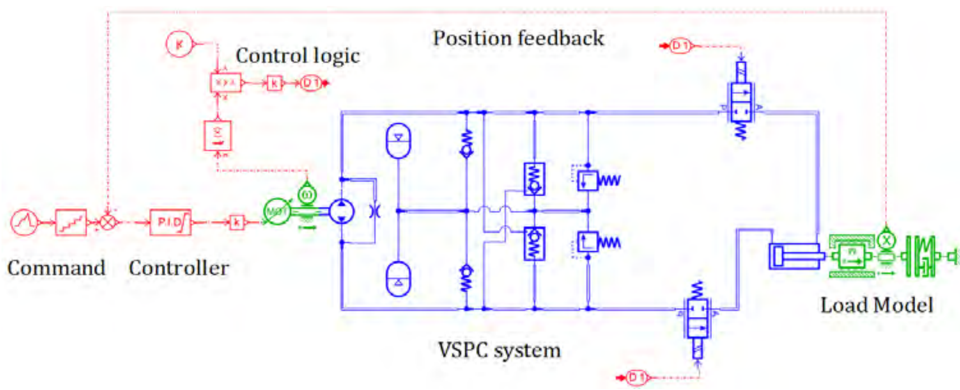


Table 1 Simulation parameters

Parameters	*Values*
Servo motor rated speed	3,000 r/min
Pump displacement	14 mL/r
Hydraulic cylinder bore and rod diameter	63 mm/40 mm
Accumulator volume	3.5 L
Gas pre-charge pressure	2.3 bar
Pilot operated check valve cracking pressure	2 bar
Pilot operated check valve pilot area ratio	4.5
Check valve cracking pressure	0.2 bar
Relief valve cracking pressure	315 bar
Mass weight	74 kg
Spring stiffness	500 N/mm
PID controller proportional gain/integral gain	1/0.2

In the VSPC system, the function of the accumulator is to equalise the volume difference between the two chambers of the cylinder. During cylinder extension and retraction, the flow rate, filling pressure and gas volume of the accumulator change according to a sine or cosine curve, as indicated in Figure 11. During cylinder extension, the accumulator discharges oil to the low pressure side, the gas volume increases and the pressure decreases. When the cylinder reaches the end, the gas volume reaches maximum and the gas pressure reaches minimum. As the cylinder retracts, the low pressure side fills the accumulator with oil, the gas volume decreases and the pressure increases.

Figure 9 Sinusoidal position tracking curves (see online version for colours)

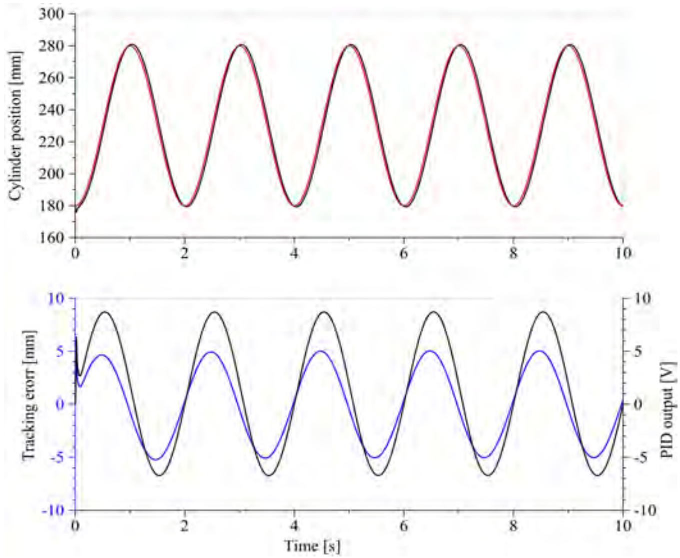


Figure 10 Motor and cylinder response curves (see online version for colours)

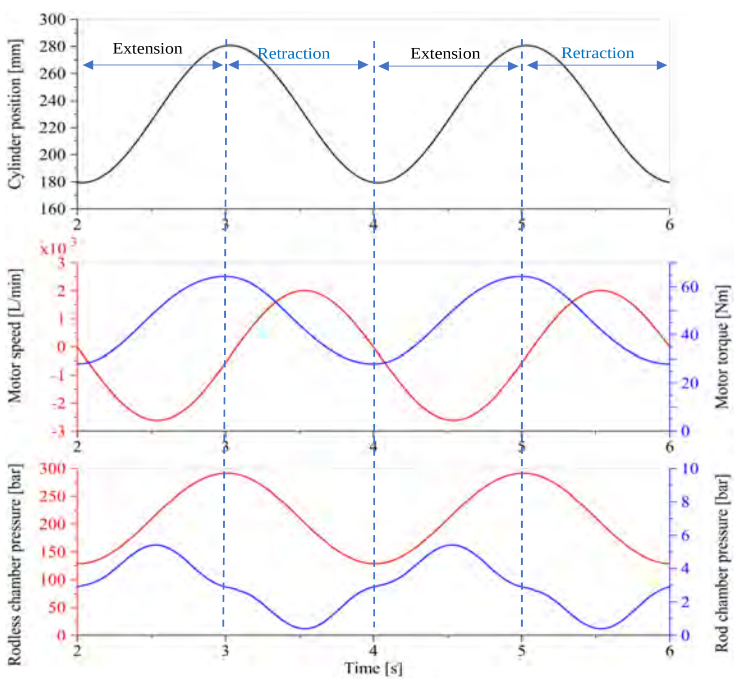
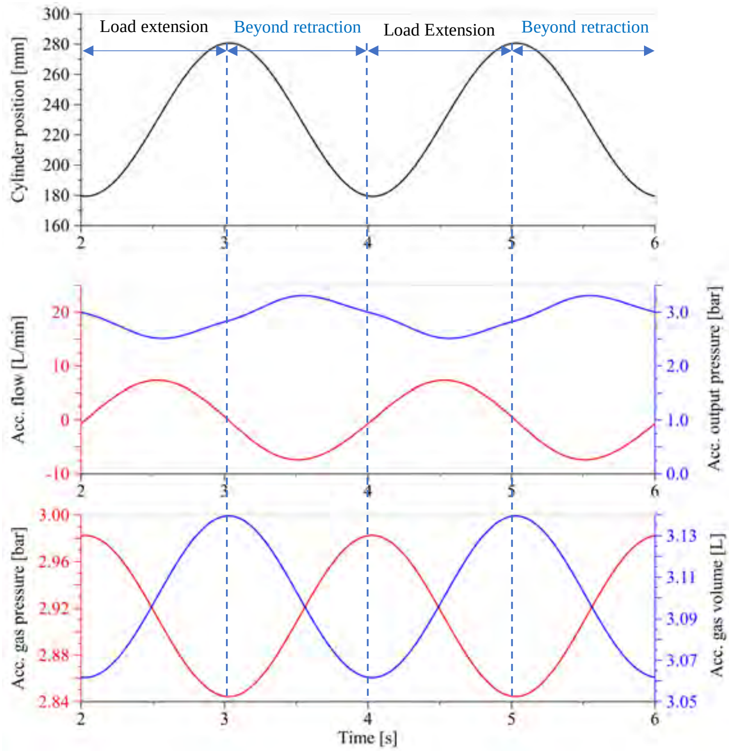


Figure 11 Accumulator response curves (see online version for colours)



6 Experimental investigation

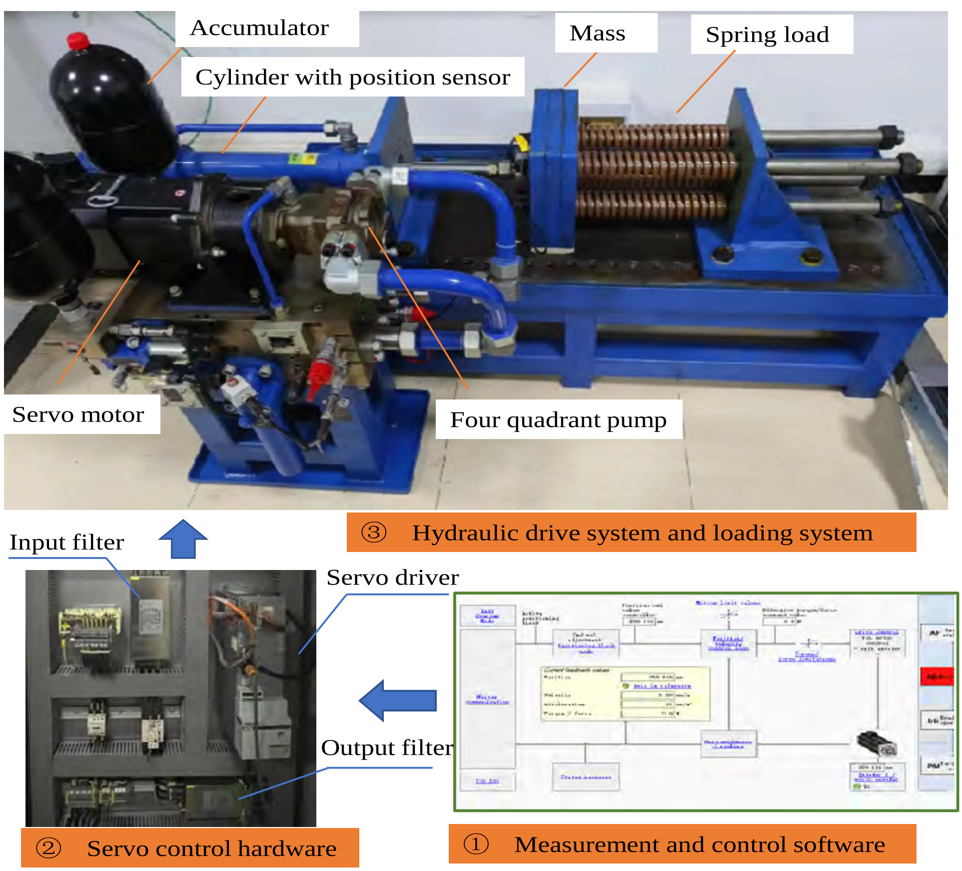
6.1 Experimental setup

The experimental test setup is depicted in Figure 12 according to structure of VSPC system discussed in Section 2, which mainly consists of measurement and control software, servo drive hardware, hydraulic drive system and loading system. In order to prevent power disturbances, input and output filters are installed at the input and output of the servo drive, respectively, and the servo drive integrates an inverter for the servo motor and a closed-loop controller for the cylinder position.

A magnetostrictive sensor integrated inside the cylinder is applied to measurer the cylinder position, which is a non-contact linear position sensor that uses the momentary interaction of two magnetic fields to produce a strain pulse that moves along a waveguide. The advantage to this type of sensor is that it is non-contact and there is no wear or friction. It is also not affected by vibrations so there is no limit on the number of operating cycles. The position sensor stroke is 400 mm, accuracy 1 um, synchronous serial interface (SSI) output. The metering pump is a special piston pump, which is designed for variable speed drives, and could operate in four quadrants, i.e., pump status, motor status and forward and backward rotation. The loading system mainly consists of a mass block and a spring, the weight of the mass block is 74 kg and the spring stiffness is 600 N/mm. The main parameters of the experimental system are shown in Table 2.

The stroke of the cylinder is 400 mm, when the displacement of the cylinder is more than 260 mm, the mass block starts to compress the spring, and spring force changes with the displacement. The load table can be used to simulate the load on the system in three quadrants. As shown in Figure 13, when the cylinder displacement increases from 235 mm to 285 mm, the system works in the first quadrant and is in the load extension state, where the cylinder mainly overcomes the friction and inertia force from 235 mm to 260 mm, and the cylinder compresses the spring and mainly overcomes the elastic force from 260 mm to 285 mm; when the cylinder is retracted from 280 mm to 260 mm, the spring does the work and the system works in the second quadrant and is in the beyond retraction state; when the cylinder displacement is retracted from 260 mm to 235 mm, the system works in the third quadrant and is in the load retraction state. When the cylinder displacement is retracted from 280 mm to 260 mm, the spring works and the system works in the second quadrant, which is the beyond retraction state; when the cylinder displacement is retracted from 260 mm to 235 mm, the system works in the third quadrant, which is the load retraction state, and the cylinder mainly overcomes friction and inertia force.

Figure 12 Experimental setup (see online version for colours)



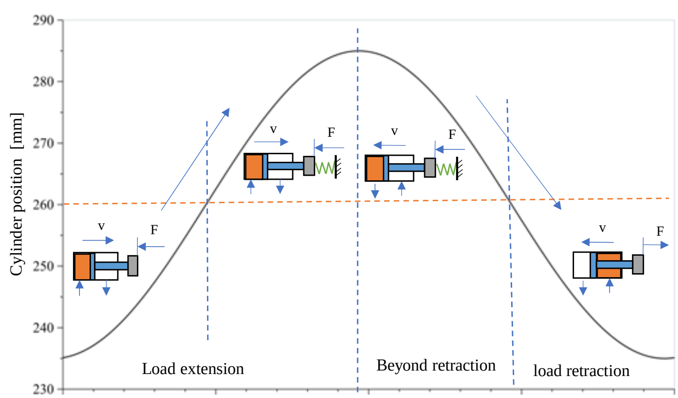
6.2 Step response performance analysis

Step response experiments of position servo are carried out on this test bench. As shown in Figure 14, the displacement command is from 280 mm step to 290 mm, and the load force during this period is from 10.5 kN step to 16.2 kN, and the step time of the cylinder is 0.64 s without overshooting by the variable-speed pumping control, and the tracking accuracy is high. During the step response, the motor speed increases and then decreases, and the motor speed is the highest at the middle displacement, reaching 477.9 r/min. When the cylinder displacement is kept at 290 mm, the motor does not stop or frequently go forward and backward, but fluctuates up and down at 20 rpm.

Table 2 Main component parameters of the experimental system

Components	*Parameters*
Servo motor	Rated power 6.9 kW, rated torque 23 Nm, rated speed 3,000 r/min, 17-bit incremental encoder
Servo driver	Rated power 11 kW, max. current 54A, integrated controller, multi-encoder interface
Fixed pump	Displacement 14 mL/r, max. speed 3,300 rpm, nominal pressure, four quadrants operation
Asymmetric cylinder	Cylinder bore 63 mm, rod bore 40 mm, stroke 300 mm
Position sensor	Magnetostriction, stroke 400 mm, precision 1 um, SSI output
Accumulator	Volume 3.5 L, gas precharge pressure 2.3 bar, polytropic index 1.4
Load	Mass 74 kg, spring stiffness 600 N/mm

Figure 13 Schematic of loads with cylinders operating in three quadrants (see online version for colours)



Typically pumps have a volumetric efficiency of about 95% with internal leakage. when the cylinder position is constant, although the flow required by the load is zero, the motor rotates in one direction and the pump outputs a certain amount of flow to compensate for its own internal leakage. The amount of internal leakage of the pump is proportional to the outlet pressure, the greater the load, the greater the internal leakage, the greater the speed of the motor when the load is maintained. This phenomenon is a new discovery in variable speed pump-cylinder control systems.

6.3 *Sinusoidal tracking performance analysis*

After step response experiments, sinusoidal position tracking experiments of the VSPC system are carried out on this test bench, as indicated in Figure 15. The position command is a sinusoidal signal with a mean level of 260 mm, an amplitude of 25 mm and a frequency of 0.5 Hz. The load on the cylinder is nonlinear. When the displacement is from 235 mm to 285 mm, the cylinder only pushes the inertia block and the load is about 0.65 kN in the push phase and 0.16 kN in the return phase. When the cylinder is displaced from 260 mm to 285 mm, the mass block compresses the spring and the load varies with the displacement and the maximum load is 13.37 kN when the displacement is 285 mm.

Figure 14 Step response of cylinder position (see online version for colours)

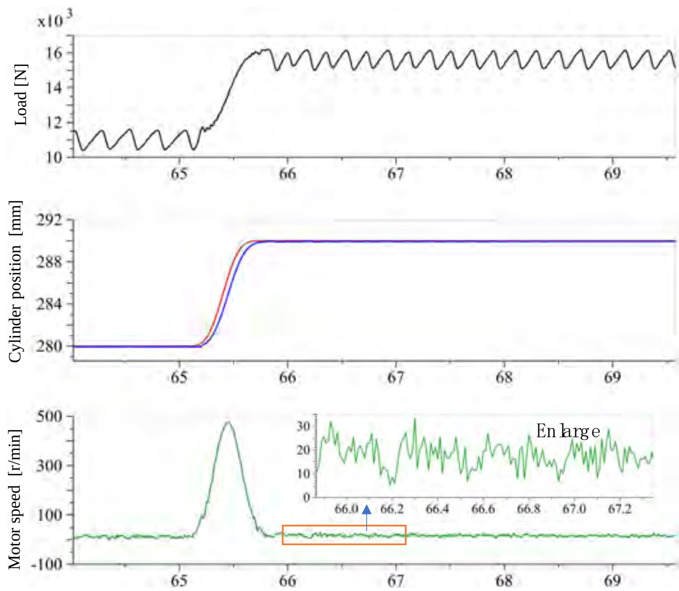
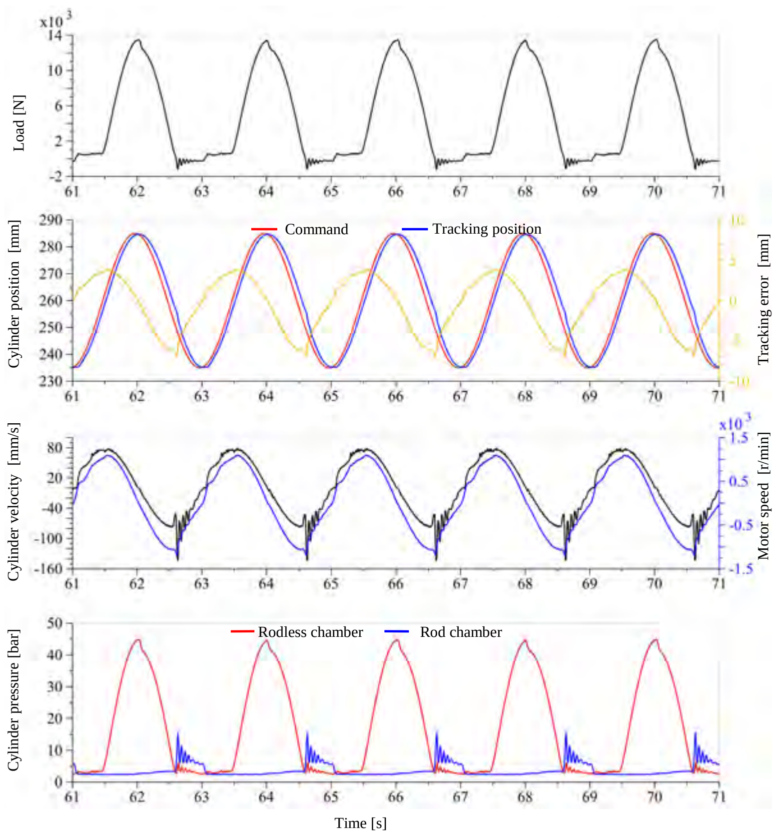


Figure 15 shows that under the control of the variable speed pump, the actuator can track sinusoidal commands quickly, accurately and smoothly, and the tracking accuracy of the push stroke is higher than that of the return stroke. The displacement error varies with cylinder displacement and is basically sinusoidal between –5.3 m and 3.5 mm, with a positive displacement error for the push stroke and a negative displacement error for the return stroke.

In the process of the actuator tracking sinusoidal command, the motor speed also changes according to the cosine, the speed change range is –1,170 r/min to 1,091 r/min, the speed change is smooth and no sudden change, which indicates that the motor variable speed can replace the pump variable to regulate the system flow.

When the cylinder is extended, the motor is in drive and the pressure in the rod chamber remains essentially constant at around 2.5 bar, slightly higher than the pressure in the accumulator. When the mass block is not in contact with the spring, the pressure in the rodless chamber is slightly higher than in the rod chamber to overcome friction and inertia forces. As soon as the mass block compresses the spring, the pressure in the rodless chamber rises rapidly and increases with displacement, and at 286 mm the pressure in the rodless chamber is 44.7 bar. When the cylinder displacement is retracted from 285 mm to 260 mm, the spring is working and the rodless chamber is still a high pressure chamber to form a back pressure, and at this time the pump is operating in a motorised state, with the motor in a power generating state. When the cylinder is retracted from 260 mm back to 235 mm, the rod chamber becomes the drive chamber, the motor is again in the drive state, and the pressure of the rod chamber is slightly higher than that of the rodless chamber to overcome friction and inertia.

Figure 15 Sinusoidal position tracking curves of the VSPC system (see online version for colours)



7 Conclusions

VSPC is a new type of EHA with the advantages of high efficiency, high reliability and easy control, which can replace the traditional form of valve-controlled cylinder. This thesis investigates the VSPC system through design, modelling, simulation and experiment, and the main work of the thesis is as follows:

Designed a VSPC system operating in four quadrants, which uses a servomotor – quantitative pump – differential cylinder structure, the use of volumetric control instead of throttling control, by adjusting the motor speed to change the system flow rate, direct control of hydraulic cylinders, and elucidated the principle of the system in the four-quadrant work.

The mathematical model of the system is established, and the differences between the pump-controlled cylinder and valve-controlled cylinder systems in terms of resonance frequency and damping ratio are compared and analysed, revealing that the resonance frequency of the VSPC system is in the form of L-type, which decreases with the increase of stroke, and the damping ratio is small and stable, and the PID controller with anti-integral saturation is designed, which provides a reference for the design of the closed-loop control algorithm of the system.

Based on the AMESim platform, a simulation model was established, and an experimental bench was built to carry out the sinusoidal position tracking and step response, which reveals the change rule of the servomotor's speed and torque in the process of sinusoidal position tracking, and analyses the change process of the accumulator's gas volume and pressure. The simulation and experimental results show that the hydraulic cylinder can accurately and quickly track the command signal in all quadrants by using the variable speed pump direct drive. The step response experiments show that when the position of the cylinder is unchanged, although the flow rate of the load demand is zero, the servomotor still needs to have a certain rotational speed and the direction of rotation is unchanged in order to compensate for the leakage of the system to generate hydraulic pressure to control the position of the cylinder.

Variable speed pump control is a new technology of hydraulic transmission and control with many advantages, such as high energy efficiency, easy to control, simple structure, high reliability. However, it is difficult to be applied to high-power applications like variable pump control due to the smaller displacement of the quantitative pump. As the response speed of the servomotor is lower than that of the proportional or servo valves, these results in a low response speed of the VSPC system. Future work is to combine valve control and variable speed pump control to improve the response speed of the VSPC system, where the coordinated control of the pump control and valve control is the focus of the research.

Acknowledgements

This work is supported by a Fundamental Research Funds for Central Universities under Grant 2019XKQYMS37, and a Project Funded by the Priority Academic Program Development of Jiangsu Higher Education Institutions.

References

Caliskan, H., Balkan, T. and Platin, E.B. (2015) 'A Complete analysis and a novel solution for instability in pump controlled asymmetric actuators', *J. Dyn. Syst-T ASME*, Vol. 137, No. 9, p.91008.

Chen, B., Lee, T., Peng, K. et al. (2003) 'Composite nonlinear feedback control for linear systems with input saturation: theory and an application', *IEEE Transactions on Automatic Control*, Vol. 48, No. 3, pp.427–439.

Chen, M. and Zhao, D. (2017) 'The gravitational potential energy regeneration system with closed-circuit of boom of hydraulic excavator', *Mechanical Systems and Signal Processing*, Vol. 82, pp.178–192, https://doi.org/10.1016/j.ymssp.2016.05.017.

Chen, Q., Lin, T., Ren, H. and Fu, S. (2019) 'Novel potential energy regeneration systems for hybrid hydraulic excavators', *Mathematics and Computers in Simulation*, Vol. 163, pp.130–145, https://doi.org/10.1016/j.matcom.2019.02.017.

Danaee, S., Nurmi, J., Minav, T. et al. (2018) 'Direct position control of electro-hydraulic excavator', *The BATH/ASME 2018 Symposium on Fluid Power and Motion Control*, Bath, UK, 12–14 September, p.V001T01A049.

Fu, S., Wang, L. and Lin, T. (2020) 'Control of electric drive powertrain based on variable speed control in construction machinery', *Automation in Construction*, Vol. 119, p.103281, https://doi.org/10.1016/j.autcon.2020.103281.

Hao, Y., Xia, L., Quan, L. et al. (2019) 'Research on characteristics of heavy-load lifting machinery based on hydraulic-gas energy storage and closed displacement controlled system', *Journal of Mechanical Engineering*, Vol. 55, No. 16, pp.213–219.

Helian, B., Chen, Z. and Yao, B. (2020) 'Precision motion control of a servomotor-pump direct-drive electrohydraulic system with a nonlinear pump flow mapping', *IEEE Transactions on Industrial Electronics*, Vol. 67, No. 10, pp.8638–8648.

Helian, B., Chen, Z., Yao, B. et al. (2021) 'Accurate motion control of a direct-drive hydraulic system with an adaptive nonlinear pump flow compensation', *IEEE/ASME Transactions on Mechatronics*, Vol. 26, No. 5, pp.2593–2603.

Hippalgaonkar, R. and Ivantysynova, M. (2013) 'A series-parallel hydraulic hybrid mini-excavator with displacement controlled actuators', *The 13th Scandinavian International Conference on Fluid Power*, Linköping, Sweden, 3–5 June, pp.31–42.

Kilic, E., Dolen, M., Caliskan, H. et al. (2014) 'Pressure prediction on a variable-speed pump controlled hydraulic system using structured recurrent neural networks', *Control Engineering Practice*, Vol. 26, pp.51–71, https://doi.org/10.1016/j.autcon.2020.103281.

Lin, T., Lin, Y., Ren, H. et al. (2020) 'Development and key technologies of pure electric construction machinery', *Renewable and Sustainable Energy Reviews*, Vol. 132, p.110080, https://doi.org/10.1016/j.rser.2020.110080.

Lin, T., Lin, Y., Ren, H. et al. (2021) 'A double variable control load sensing system for electric hydraulic excavator', *Energy*, Vol. 223, No. 27, p.119999.

Quan, L. (2008) 'Current status, problems and innovative solutions of electrohydraulic technology for pump control cylinders', *Journal of Mechanical Engineering*, Vol. 44, No. 2, pp.87–92.

Quan, L., Neubert, T. and Helduser, S. (2004) 'Principle to closed loop control differential cylinder with double speed variable pumps and single loop control signal', *Chinese Journal of Mechanical Engineering*, Vol. 17, No. 1, pp.85–88.

Quan, Z., Quan, L. and Zhang, J. (2014) 'Review of energy efficient direct pump controlled cylinder electro-hydraulic technology', *Renewable and Sustainable Energy Reviews*, Vol. 35, pp.336–346, https://doi.org/10.1016/j.rser.2014.04.036.

Rahmfeld, R. and Ivantysynova, M. (2001) 'Displacement controlled linear actuator with differential cylinder – a way to save primary energy in mobile machines', *The 5th International Conference on Fluid Power Transmission and Control*, Hangzhou, China, 3–5 April, pp.316–322.

Sarkar, A., Maji, K., Chaudhuri, S. et al. (2023) 'Actuation of an electrohydraulic manipulator with a novel feedforward compensation scheme and PID feedback in servo-proportional valves', *Control Engineering Practice*, Vol. 135, p.105490, https://doi.org/10.1016/j.conengprac.2023.105490.

Waheed, U.R., Wang, S., Wang, X. et al. (2016) 'Motion synchronization in a dual redundant HA/EHA system by using a hybrid integrated intelligent control design', *Chinese Journal of Aeronautics*, Vol. 29, No. 3, pp.789–798.

Wang, H. and Leaney, P. (2020) 'Modelling and energy efficiency analysis of a hybrid pump-controlled asymmetric cylinder drive system', *International Journal of Hydromechatronics*, Vol. 3, No. 1, p.214286649.

Wang, Y., Guo, S. and Dong, H. (2020) 'Modeling and control of a novel electro-hydrostatic actuator with adaptive pump displacement', *Chinese Journal of Aeronautics*, Vol. 33, No. 1, pp.365–371.

Williamson, C., Zimmerman, J. and Ivantysynova, M. (2008) 'Efficiency study of an excavator hydraulic system based on displacement-controlled actuators', *The Bath/ASME Symposium on Fluid Power and Motion Control*, Bath, UK, 10–12 September, pp.293–310.

Zhang, S., Li, S. and Minav, T. (2020) 'Control and performance analysis of variable speed pump-controlled asymmetric cylinder systems under four-quadrant operation', *Actuators*, Vol. 9, No. 4, p.123.

Zhang, S., Minav, T., Pietola, M. et al. (2019) 'The effects of control methods on energy efficiency and position tracking of an electro-hydraulic excavator equipped with zonal hydraulics', *Automation in Construction*, Vol. 100, pp.129–144, https://doi.org/10.1016/j.autcon.2019.01.003.

Zhu, Q., Mobayen, S., Nemati, H. et al. (2023) 'A new configuration of composite nonlinear feedback control for nonlinear systems with input saturation', *Journal of Vibration and Control*, Vol. 29, No. 6, pp.1417–1430.