

ORIGINAL ARTICLE

Open Access



# A Multi-mode Electronic Load Sensing Control Scheme with Power Limitation and Pressure Cut-off for Mobile Machinery

Min Cheng<sup>1</sup>, Bolin Sun<sup>1</sup>, Ruqi Ding<sup>2\*</sup>  and Bing Xu<sup>3</sup>

## Abstract

In mobile machinery, hydro-mechanical pumps are increasingly replaced by electronically controlled pumps to improve the automation level, but diversified control functions (e.g., power limitation and pressure cut-off) are integrated into the electronic controller only from the pump level, leading to the potential instability of the overall system. To solve this problem, a multi-mode electrohydraulic load sensing (MELS) control scheme is proposed especially considering the switching stability from the system level, which includes four working modes of flow control, load sensing, power limitation, and pressure control. Depending on the actual working requirements, the switching rules for the different modes and the switching direction (i.e., the modes can be switched bilaterally or unilaterally) are defined. The priority of different modes is also defined, from high to low: pressure control, power limitation, load sensing, and flow control. When multiple switching rules are satisfied at the same time, the system switches to the control mode with the highest priority. In addition, the switching stability between flow control and pressure control modes is analyzed, and the controller parameters that guarantee the switching stability are obtained. A comparative study is carried out based on a test rig with a 2-ton hydraulic excavator. The results show that the MELS controller can achieve the control functions of proper flow supplement, power limitation, and pressure cut-off, which has good stability performance when switching between different control modes. This research proposes the MELS control method that realizes the stability of multi-mode switching of the hydraulic system of mobile machinery under different working conditions.

**Keywords** Hydraulic control, Load sensing, Multi-Mode, Power limitation, Mobile machinery

## 1 Introduction

To reduce energy consumption in mobile machinery, variable displacement pumps have been widely used to meet flow requirements [1–3] by positive flow control, negative flow control, load sensing, etc. Axial piston pumps are

also integrated with hydro-mechanical regulating circuits to achieve auxiliary functions, such as power limitation and pressure cut-off. A corresponding valve regulator is needed for each control function of the pump, such as load sensing, pressure cut-off, or power limitation valve [4], which leads to complex structures and limits control parameters.

In recent years, the electrification of mobile machinery has been an urgent requirement to improve its automation level [5]. Therefore, hydro-mechanical pumps are increasingly replaced by electronically controlled pump which includes a general pump, a proportional valve, and an electronic controller. The benefit is to simplify the structure and improve flexibility [6–8]. Advanced

\*Correspondence:

Ruqi Ding  
[dingruqi0791@sina.com](mailto:dingruqi0791@sina.com)

<sup>1</sup> State Key Laboratory of Mechanical Transmissions, Chongqing University, Chongqing 400044, China

<sup>2</sup> Key Laboratory of Conveyance and Equipment, Ministry of Education, East China Jiaotong University, Nanchang 330052, China

<sup>3</sup> State Key Laboratory of Fluid Power and Mechatronic Systems, Zhejiang University, Hangzhou 310027, China

algorithms are constantly being proposed to achieve diversified control functions, such as load sensing, pressure control, and so on. Firstly, electronic load sensing (ELS) pumps are designed to replace hydro-mechanical load sensing (HMLS) pumps [9–11]. Song et al. [12] proposed a direct load sensing electric hydrostatic actuator to automatically adjust the supplied pressure and flow. Secondly, nonlinear pressure controllers have been developed to cope with nonlinearities and uncertainties, including neural networks or fuzzy algorithms [13–16]. Considering the switched characteristic of the self-supplied variable displacement pump, a few controllers have been also developed to ensure the pump stability itself under unknown time-varying flow disturbance [17–19]. Moreover, integrating more control functions into ELS pumps has been continuously explored in academic and industrial areas. Ruggeri et al. [20] designed an electronic controller to integrate the control functions of power/torque limitation and variable load sensing. Also, a fuzzy controller has been designed to improve the pump's dynamic performance and fulfill the flow/power control demands [21]. Besides, some patents [22–24] about integrated load sensing control pumps have been also authorized.

In the existing literature, more control functions have been integrated into variable displacement pumps. However, rare further discussion is discussed on the system stability, and only the switching stability of the pump itself is mentioned [17, 18]. However, it is well known that the load sensing system itself tends to oscillate due to a small stability margin and potential instability in local conditions. Moreover, the hydraulic control circuit switching between different modes is a typical nonlinear switched system, and the stability of the overall system cannot be ensured even if the subsystem stability is proven [17, 25]. Our previous work has shown that the integral windup issue in ELS controllers probably leads to underlying instability due to improper operation [4]. One practical solution in actual applications is to change the working points manually by modifying input commands of joysticks when oscillations or overshoots occur, but it is just a halfway solution so that full automation is hindered.

To fulfill the complex requirements of variable displacement pumps in mobile machinery, this paper is to develop a multi-mode electronic load sensing (MELS) control scheme that integrates pressure/flow/power control functions and guarantees system stability when switching between different modes.

Traditional multi-mode electronically controlled pump only focuses on the integration of pressure, power and load sensing control functions, but neglects the system stability when switching between different modes.

However, as described in Ref. [4], the stability performance in one mode cannot ensure the stability of the switched control system. Thus, the pump control scheme is designed in this paper from the view of system stability, which is not considered in the existing pump controller design. In summary, the main contributions are as follows.

- (1) The MELS control scheme is proposed for an electronically controlled pump of mobile machinery to achieve different control modes. The controller integrates four control modes: flow control, pressure control, load sensing control and power limitation. The priority of mode switching is designed so that the hydraulic system can be switched to the corresponding control mode under the pre-set conditions to ensure the safe operation of the system.
- (2) The stability of the system during different mode switching is analyzed, and the conditions and control parameters to satisfy the switching stability are obtained. In addition, the system stability during multi-mode switching is verified by conducting experiments and simulations on a 2-ton hydraulic excavator.

The paper is organized as follows: The system layout is briefly introduced in Section 2. Then, the MELS control scheme is designed in Section 3. In Section 4, the comparative experimental test and simulation are carried out. Finally, the conclusion is given in Section 5.

## 2 System Layout

As shown in Figure 1, the studied objective is a multi-actuator hydraulic system with primary pressure compensation for mobile machinery, in which the hydro-mechanical pump is replaced by an electronically controlled pump, and the pipe or shuttle valve for pressure feedback is removed by pressure sensors. In this system, primary pressure compensators decouple the flow across the control valve from the load. Thus, the steady-state flow rates are not related to the load variation, which only depends on the opening of the control valve. The pump displacement is controlled by a hydraulic proportional control valve with feedback of the swash plate angle to improve the dynamic performance. The MELS controller is designed for the pump to fulfill diversified control functions of mobile machinery. In the proposed controller, there are four modes including flow control, load sensing, power limitation, and pressure control, which are achieved by four independent controllers with a defined switching rule based on load conditions. The modes of flow control and load sensing are used to replace the HMLS module and supply the required flow

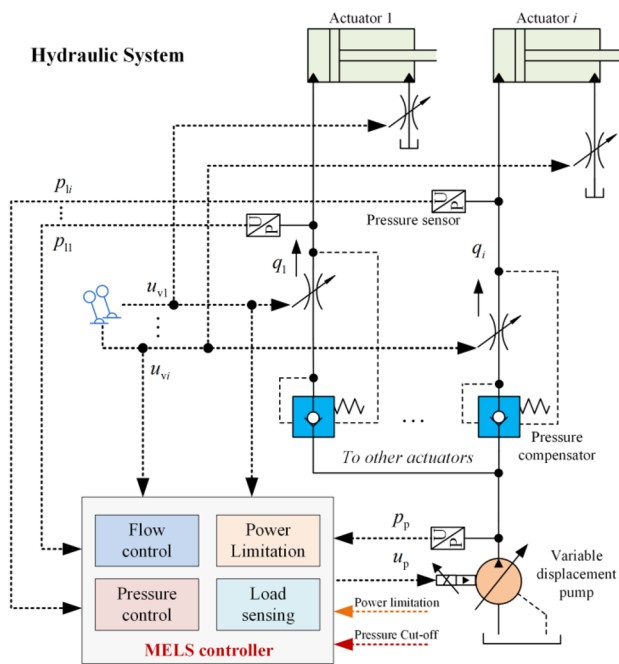


Figure 1 Hydraulic system with primary pressure compensation

to the actuators. Thus, the pump control signal can be expressed as

$$u_p = f(u_{pf}, u_{pl}, u_{pp}, u_{pr}), \tag{1}$$

where  $u_{pf}$ ,  $u_{pl}$ ,  $u_{pp}$ , and  $u_{pr}$  denote the outputs of the flow controller, the load sensing controller, the power controller, and the pressure controller, respectively.

### 3 Control Scheme Design

The three controllers of flow control, load sensing, and power limitation have been designed in Refs. [4, 24], and the main work in this paper is to design the pressure controller and especially integrate the four controllers by defining the switching rules to guarantee system stability. Without loss of completeness, these three controllers are also briefly introduced as below.

#### 3.1 Flow Control

In the Flow Control (FC) mode, the purpose is to fulfill the flow requirements of hydraulic actuators. The flow feedforward control concept is introduced by directly calculating the flow rates across the valve based on the input signals. In contrast to existing HMLS systems, it has been proven that the flow feedforward method has the advantages of high energy efficiency and fast response [26, 27]. Neglecting the pump leakage, the control signal can be written as

$$u_{pf} = \sum_{i=1}^n \frac{C_d A(u_{vi}) \sqrt{2 \Delta p_n / \rho}}{n_p k_{pp}}, \tag{2}$$

where  $C_d$  is the discharge coefficient,  $u_{vi}$  is the valve signal,  $\Delta p_n$  is the nominal pressure drop,  $\rho$  is the oil density,  $n_p$  is the pump rotational speed, and  $k_{pp}$  is the displacement gain. Note that there exists a signal difference when switching from another mode to the FC mode, so a first-order inertial part is added to avoid signal jump, which is expressed as

$$\frac{u_{pf}(s)}{u_p(s)} = \frac{1}{\tau_p s + 1}, \tag{3}$$

where  $\tau_p$  is the time constant,  $s$  is the Laplace operator.

#### 3.2 Load Sensing (LS)

Although the flow controller has the aforementioned advantage, there is a remarkable disadvantage of pressure impact and energy consumption under flow excess [25]. Thus, a load sensing mode is designed to solve this issue. Like the HMLS system, the LS controller aims to maintain the pressure margin as a preset value. To avoid signal jump when switching from other modes, an incremental Proportional-Integral-Derivative (PID) controller is thereby designed to control the pressure margin, which can be expressed as

$$u_{pl}(t) = u_{pl}(t_q) + k_{lp} e_{pr}(t) + k_{li} \int_{t_q}^t e_{pr}(t) dt + k_{ld} \dot{e}_{pr}(t), \tag{4}$$

where  $t_q$  is the initial moment of switching to LS mode,  $u_{pl}(t_q)$  is the output of the MELS controller when switching into the LS mode,  $k_{lp}$ ,  $k_{li}$ , and  $k_{ld}$  are the PID parameters in the LS mode.  $e_{pr}$  is the pressure margin error, which can be expressed as

$$e_{pr}(t) = p_{dm} - p_{pm}(t) = p_{dm} - [p_p(t) - p_{lm}(t)], \tag{5}$$

where  $p_{dm}$  is the preset pressure margin,  $p_{pm}$  is the actual pressure margin,  $p_p$  is the system pressure.  $p_{lm}$  is the highest load pressure, which can be expressed as

$$p_{lm}(t) = \max \{p_{l1}, \dots, p_{li}, \dots, p_{ln}\} \quad (i = 1, \dots, n), \tag{6}$$

where  $p_{li}$  is the load pressure.

#### 3.3 Power Limitation (PL)

The purpose of the power controller is to limit the system output power to avoid engine stall. Under a limited value  $P_n$ , the derived pump flow rate can be expressed as  $q_p = P_n / p_p$  [4]. The power controller is designed as

$$u_{pp} = \frac{P_p}{n_p p_p(t)} \hat{G}_p^{-1}(t) - u_{vlc}(t), \tag{7}$$

where  $u_{vlc}$  is the compensator based on dynamic pressure feedback to achieve active damping control [28],  $\hat{G}_p$  is the identified pump model. The Laplace form of  $u_{vlc}$  is given by

$$u_{vlc}(s) = \frac{k_{cp}s}{(\omega_{cp} + s)} p_p(s) \hat{G}_p^{-1}(s), \tag{8}$$

where  $k_{cp}$  and  $\omega_{cp}$  are the control gain and the cut-off frequency of the compensator, respectively. The pump dynamic is modeled as a first-order term [4], which is given by

$$\frac{V_p(s)}{u_p(s)} = G_p(s) = \frac{k_{pp}}{1 + \tau_c s}, \tag{9}$$

where  $\tau_c$  is the time constant,  $V_p$  is the pump displacement.

### 3.4 Pressure Control (PC)

When the resistive load of the system is too large, the pressure controller is triggered to limit the system pressure and prevent affecting the hydraulic system adversely. Therefore, the pressure controller aims to maintain the outlet pressure of the pump at a certain value. Referring to the pressure cut-off circuit in current machinery, an electronic pressure controller is designed using incremental PID control to avoid signal jump when switching into the PC mode. The PC mode can be activated when the cylinder reaches the end stop, also it can work under the condition that the external load is too high. Moreover, to avoid faulty activation caused by pressure impact in the FC or LS mode, incremental control is defined to be active when the supplied flow is not excessive ( $p_{lm}(t) \leq p_{dm}$ ). Thus, the incremental PID controller is designed by

$$\dot{u}_{pr}(t) = \phi(t) [k_{rp}\dot{e}_{pc}(t) + k_{ri}e_{pc}(t) + k_{rd}\ddot{e}_{pc}(t)], \tag{10}$$

where  $k_{rp}$ ,  $k_{ri}$ , and  $k_{rd}$  are the PID parameters in the PC mode, and the function  $\phi(t)$  is defined by

$$\phi(t) = \begin{cases} 1, & p_{lm}(t) \leq p_{dm}, \\ 0, & p_{lm}(t) > p_{dm}. \end{cases} \tag{11}$$

The pressure error is defined as

$$e_{pc}(t) = p_{dc} - p_p(t), \tag{12}$$

where  $p_{dc}$  is the desired pressure, it is also the upper bound in PC mode (in Section 3.5 below). The initial value of the pressure controller is defined as the output of the MELS controller when switching into the PC

mode. Since the pressure controller does not require high dynamic in actual applications, a smaller control gain can be selected to reduce pressure overshoot and oscillation. To avoid that the power exceeds the limited value in the PC mode, an upper bound  $u_{pr} = P_p / n_p k_{pp} p_{dc}$  is set for the pressure controller.

### 3.5 Switching Rule

The control structure with four modes is simple and can be easily implemented in current control hardware. In addition, another concern is the switching rule to ensure the overall system stability. Based on Refs. [4, 25], a multi-mode switching rule is designed to avoid potential stability caused by continuous switching between different modes, as shown in Figure 2.

To simplify the switching rules and ensure the system stability, the bilateral switching rules are defined as “FC ↔ PC”, “FC ↔ LS” and “FC ↔ PL”, considering the open-loop structure of flow control, in which the symbol “↔” means the switching is bilateral. Moreover, some unilateral switching rules are defined. For instance, the switching between LS and PC is unilateral, and switching from LS to PC is not allowed. For this reason, the LS controller has a small stability margin, so potential instability due to continuous switching in local conditions is avoided. A similar unilateral switching rule is established between LS and PC. Since an upper bound has been configured in the PC mode, so a unilateral switching rule is also designed between PL and PC. The switching rules are summarized in Table 1, in which the modes in the first row mean those before switching, and the modes in the first column mean those after switching.

To ensure stability when switching between PC and FC, a hysteresis switching rule with dwell-time is designed. As given in Figure 3, the rule is defined as

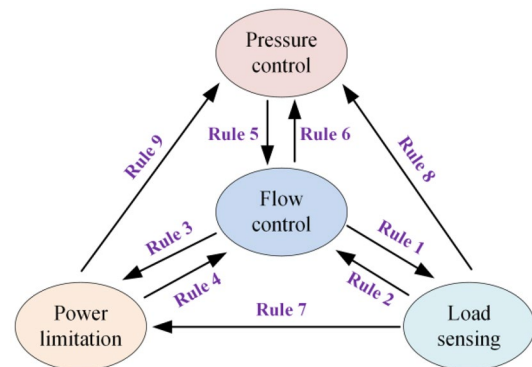
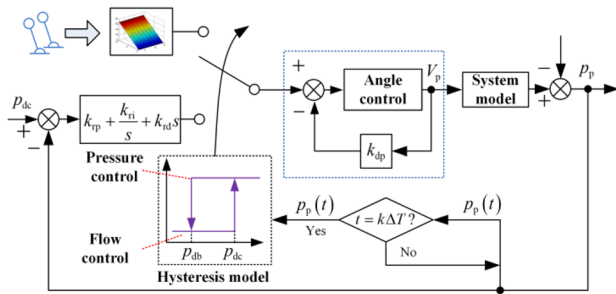


Figure 2 Switching rule of the MELS controllers

**Table 1** Switching rule of the controller

	FC	LS	PL	PC
FC	-	Given in Ref. [20]	$u_{pp}(t) \geq u_{pl}(t)$	Described by Eq. (13)
LS	Given in Ref. [20]	-	Not allowed	Not allowed
PL	$u_{pp}(t) < u_{pl}(t)$	$p_p(t) < p_{dc} \ \& \ u_{pl}(t) \geq u_{pp}(t)$	-	Not allowed
PC	Described by Eq. (13)	$p_p(t) \geq p_{dc}$	$p_p(t) \geq p_{dc}$	-



**Figure 3** Block diagram of hysteresis switching rule

**Table 2** Priority of the multi-mode controller

Control mode before switching	Priority sequence		
	1st	2nd	3rd
FC	No. 6	No. 3	No. 1
LS	No. 8	No. 7	No. 2
PL	No. 9	No. 4	-
PC	No. 5	-	-

$$u_p(t) = \begin{cases} u_{pr}(t), p_p(t) > p_{dc}, & t = t_0 + k\Delta T, k = 1, 2, \dots, n, \\ u_{pp}(t), p_p(t) < p_{db}, & \end{cases} \quad (13)$$

where  $p_{db}$  and  $p_{dc}$  are the lower bound and the upper bound for pressure control, respectively ( $p_{db} < p_{dc}$ ).  $t_0$  is the initial moment of mode switching. The dwell time is chosen at  $\Delta T = 0.5/f_{min}$ , where  $f_{min}$  is the lowest natural frequency of the overall system, which is measured when the load of excavator bucket is heaviest, and its calculation method can be referred to the Ref. [29].

The switching rule No. 7 is defined as  $p_p(t) < p_{dc}$  and  $u_{pl}(t) \geq u_{pp}(t)$ . The unidirectional switching rules No. 8 and No. 9 are defined as  $p_p(t) \geq p_{dc}$ . From the switching rules, it is predicted that there are some load conditions under which several rules are satisfied simultaneously. To determine the control mode in this case, the control priority is defined in Table 2. From Table 2, it is seen that the PC mode has the highest priority compared with other modes. That is, once the system pressure  $p_p$  is equal to or larger than  $p_{dc}$ , the controller will switch into the PC

mode. Moreover, the PL mode has the second priority: when the control signal in the LS or FC mode is greater than the power limitation value, it switches into the PL mode so that the system power is controlled within the setting range.

### 3.6 Stability Analysis

One concern is whether stability can be guaranteed when switching dynamically between different modes. Rules 1 and 2 between LS and FC are designed in the previous work based on multiple Lyapunov functions [24]. Rules 3 and 4 are to select the smaller value in the LS and FC modes, and the stability can be also ensured based on the analysis result using the describing function tool [4]. Thus, after designing the stable hysteresis switching rule between FC and PC, the remaining work is to analyze the stability under pressure control, which is carried out based on the linearization mathematical model in the Laplace form as below. The pump flow is expressed by

$$q_p(s) = n_p G_p(s) u_p(s) - k_{lp} p_p(s), \quad (14)$$

where  $k_{lp}$  is the pump leakage coefficient. Based on the flow continuity equation, the expressions are obtained as

$$p_p(s) = \frac{\beta_e}{V_{pi}s} [q_p(s) - q_1(s)], \quad (15)$$

$$p_1(s) = \frac{\beta_e}{V_1 s} [q_1(s) - A_1 v_1(s)], \quad (16)$$

where  $\beta_e$  is the effective bulk modulus,  $V_{pi}$  is the chamber volume between the pump and valve,  $V_1$  is the volume between the valve and cylinder,  $q_1$  is the flow rate out of the control valve,  $p_1$  is the pressure in the cap-side chamber,  $A_1$  is the effective area of cap-side chamber,  $v_1$  is the cylinder velocity. The flow equation of the valve is expressed by

$$q_1(s) = k_q k_v u_{v1}(s) + k_{pq} [p_p(s) - p_1(s)], \quad (17)$$

where  $k_q$  is the flow coefficient,  $k_v$  is the gain of the valve displacement,  $k_{pq}$  is the flow-pressure coefficient. Since the external load is relatively large in the PC mode, the

pressure in the rod-side chamber is neglected, so the force balance equation of the cylinder is simplified as

$$(m_1s + b_1)v_1(s) = A_1p_1(s) - F_e(s), \tag{18}$$

where  $m_1$  is the load mass,  $b_1$  is the viscous damping,  $F_e$  is the external force. As mentioned before, the PC mode can be active in two cases: ① the cylinder reaches the end stop; ② the external load is too large but the cylinder still moves forward. Actually, Case 1 can be considered as a special condition of Case 2. For the system pressure, the stability condition in Case 1 is more rigorous, since the moving cylinder provides equivalent damping to the system from Eq. (16). To obtain a conservative stability condition, the pump leakage is neglected, and the transfer function from the valve signal to the cylinder pressure in Case 1 is drawn from Eqs. (9)–(10) and (14)–(18) as

$$\frac{p_p(s)}{u_v(s)} = -\frac{k_qk_vV_1\beta_e s^2(1 + \tau_c s)}{N_0s^4 + N_1s^3 + N_2s^2 + N_3s + N_4} \tag{19}$$

where

$$\begin{aligned} N_0 &= V_1V_{pi}\tau_c \\ N_1 &= V_1V_{pi} + k_{pq}V_{pi}\beta_e\tau_c + k_{pq}V_1\beta_e\tau_c + k_{pp}k_{rd}n_pV_1\beta_e \\ N_2 &= k_{pq}V_{pi}\beta_e + k_{pq}V_1\beta_e + k_{pp}k_{pq}k_{rd}n_p\beta_e^2 + k_{pp}k_{rp}n_pV_1\beta_e \\ N_3 &= k_{pp}k_{pq}k_{rp}n_p\beta_e^2 + k_{pp}k_{ri}n_pV_1\beta_e \\ N_4 &= k_{pp}k_{pq}k_{ri}n_p\beta_e^2 \end{aligned}$$

Since the flow-pressure coefficient  $k_{pq}$  provide a positive effect on the stability, the extreme condition  $k_{pq} = 0$  is considered, and then Eq. (19) is simplified as

$$\frac{p_p(s)}{u_v(s)} = -\frac{k_qk_vV_1\beta_e s(1 + \tau_c s)}{N_0s^3 + N'_1s^2 + N'_2s + N'_3}, \tag{20}$$

where  $N'_1 = V_1V_{pi} + k_{pp}k_{rd}n_pV_1\beta_e$ ,  $N'_2 = k_{pp}k_{rp}n_pV_1\beta_e$  and  $N'_3 = k_{pp}k_{ri}n_pV_1\beta_e$ . Eq. (20) is a third-order system actually. Based on the Routh stability criterion, its stability condition can be expressed as

$$N_0, N'_1, N'_2, N'_3 > 0, \tag{21}$$

$$(N'_1N'_2 - N_0N'_3)/N'_1 > 0 \Leftrightarrow N'_1N'_2 - N_0N'_3 > 0. \tag{22}$$

Then, the stability condition in the PC mode can be drawn from Eqs. (21)–(22) as

$$0 < k_{ri} < \frac{V_{pi} + k_{pp}k_{rd}n_p\beta_e}{V_{pi}\tau_c} k_{rp}. \tag{23}$$

Based on Eq. (23), the PID parameters can be properly selected to ensure the system stability.

### 4 Experimental Validation

#### 4.1 Test Rig

A comparative experimental study is performed on a test rig with a 2-ton hydraulic excavator to validate the MELS controller, as shown in Figure 4. The hydraulic excavator is equipped with an electronically controlled pump (SYDFEE from Bosch Rexroth, Inc.) and a multi-way control valve (PVG 32 from Danfoss, Inc.). Further information about the test rig can be found in Ref. [30]. Motion control tests were then carried out and the main parameters are listed in Tables 3 and 4.

#### 4.2 Experimental Test: Boom Motion

To validate the effectiveness of the proposed MELS controller, the pure flow control method (Section 3.1) for the system is taken as a comparison.

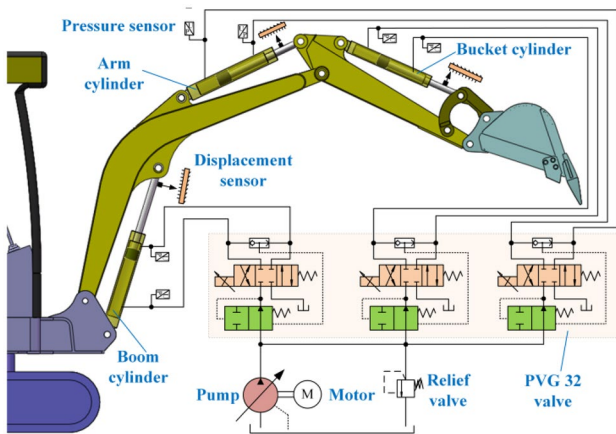
Firstly, valve control signals are given to complete the typical reciprocating movement of the boom. The movement process of the boom cylinder is: low-speed

**Table 3** Main parameters of the test rig

Parameters	Value
Maximum pump displacement (mL/r)	45.6
Motor rotation speed (r/min)	1500
Setting Relief pressure (MPa)	12
Cylinder/Piston diameter of the boom (m)	0.07/0.04
Cylinder/Piston diameter of the arm (m)	0.07/0.04
Cylinder/Piston diameter of the bucket (m)	0.06/0.035

**Table 4** Parameters in the proposed controller

Parameters	Value
Power limitation point $P_n$ (kW)	1.2
Pressure margin in the LS mode $p_{dm}$ (MPa)	1.5
PID parameters in the PC mode $k_{rp}/k_{ri}/k_{rd}$	0.1/0.6/0.01
Upper/lower pressure bound $p_{dc}/p_{db}$ (MPa)	11/9
Switch dwelling time $\Delta T$ (s)	0.5



**Figure 4** Hydraulic schematic of the test rig

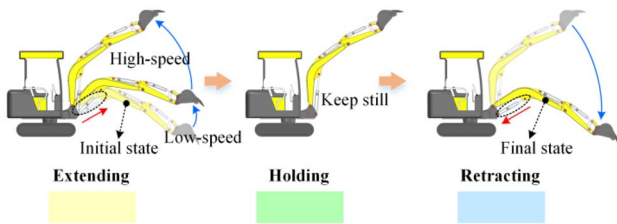


Figure 5 Schematic diagram of the boom movement

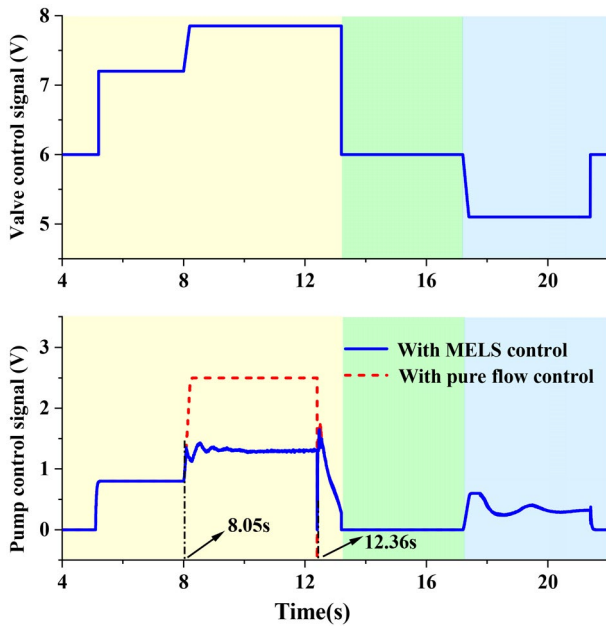


Figure 6 Control signals of the valve and pump

extending, high-speed extending, holding and retracting, as shown in Figure 5. The control signals of the valve and pump are also shown in Figure 6, and the test result is shown in Figures 7 and 8. The pump displacement and actuator velocity are shown in Figure 7, and the system pressure and power consumption are shown in Figure 8.

- (1) Low-speed extending (5.2–8 s): The speed of the boom cylinder is small, and the system with the MELS controller is in FC mode because the given valve control signal is not large. Therefore, the state of the system is essentially the same under the MELS or pure flow controller.
- (2) High-speed extending (8–13.2 s): It can be found that at  $t = 8.05$  s, the system with the MELS controller switches from the FC mode to the PL mode (Rule No. 3) because the system power exceeds the power limitation point, i.e., it exceeds 1.2 kW. Further, both the pump displacement and cylinder velocity decrease. At the same time, the flow

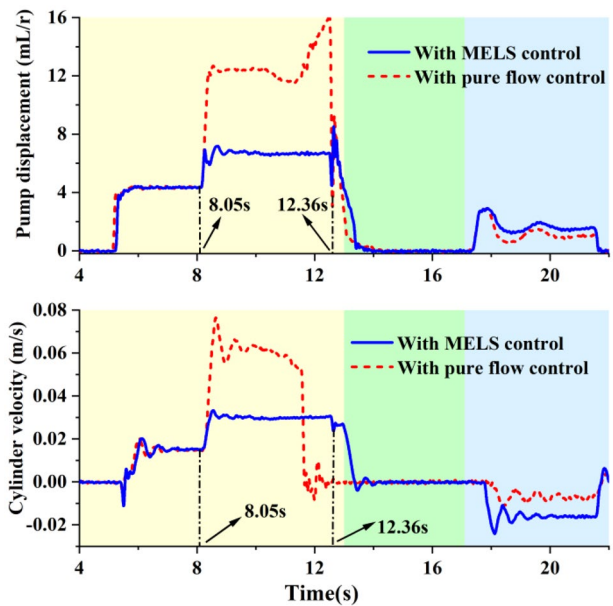


Figure 7 Pump displacement and actuator velocity

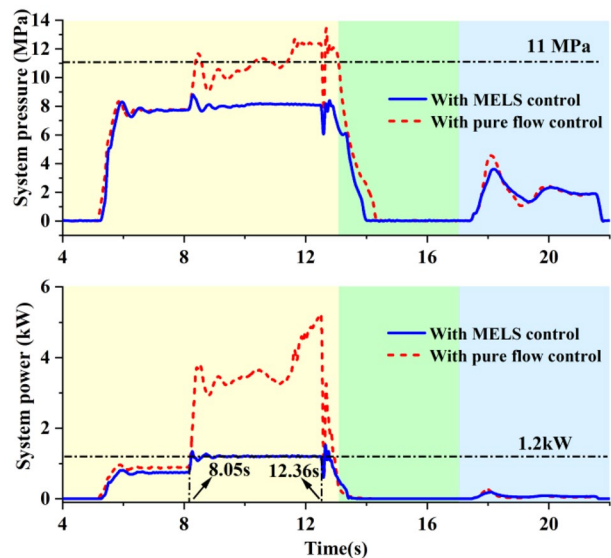


Figure 8 System pressure and Power consumption

through the proportional valve decreases, resulting in a pressure drop of the meter-in orifice and a corresponding decrease in the system pressure. Therefore, PC mode will not be active at this time. At  $t = 8.05$ – $12.35$  s, the power consumption using the MELS controller is controlled around 1.2 kW. During this stage, the root mean square (RMS) error of the system power is 0.035 kW, and the control accuracy can fulfill the actual requirement of mobile machinery. At  $t = 12.36$  s, the system switches from

PL into FC mode (Rule No. 4). At 12.36–13.2 s, if the system power exceeds the set point, or the system pressure margin exceeds the set value, the system will switch to power limitation (Rule No. 3) or load sensing control (Rule No. 1) mode.

However, in this stage, the actuator reaches the end of the boom cylinder of the system with pure flow control and stops moving because of its high speed. This resulted in an excessive system power of more than 1.2 kW and an excessive system pressure of more than 11 MPa.

- (3) Holding (13.2–17.2 s): In this phase, the boom cylinder remains stationary. The system with the MELS controller is controlled in FC mode.
- (4) Retracting (17.2–21.4 s): During the low movement of the boom cylinder, the system pressure is relatively low, and the PC and PL modes are not active. However, due to large pressure fluctuations, the switching condition of the LS controller is triggered, i.e., the pressure margin is greater than the setting value of 1.5 MPa. Therefore, the system is switched from FC mode to LS mode (Rule No. 1).

### 4.3 Experimental Test: Arm Motion

Similarly, the typical reciprocating movement of the arm cylinder is tested. The movement process of the arm cylinder is: high-speed retracting, low-speed retracting, holding and extending, as shown in Figure 9. The control signals of the valve and pump are shown in Figure 10, and the results are shown in Figures 11 and 12.

- (1) High-speed retracting (5–9.2 s): At the beginning of the movement, the system is in FC mode. At  $t = 5.7$  s, the system power exceeds the set point of 1.2 kW, but the system pressure does not exceed the set point of 12 MPa. Therefore, the system with the MELS controller switches from FC mode to PL mode (Rule No. 3), and the pump displacement and cylinder velocity are reduced, so the system power maintains near the setting value of 1.2 kW, until the end of the high-speed retracting phase. At the

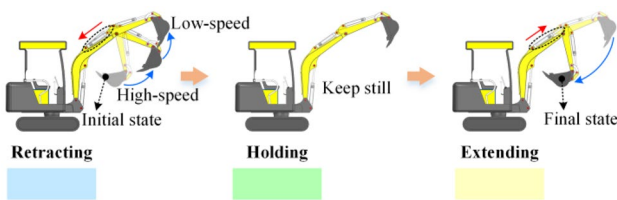


Figure 9 Schematic diagram of the arm movement

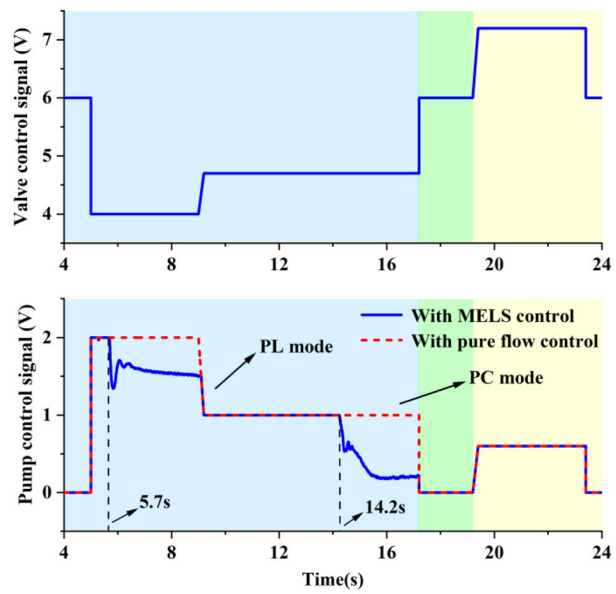


Figure 10 Control signals of the valve and pump

end of high-speed retracting, the system power decreases and the system switches from PL mode to FC mode (Rule No. 4). However, at this stage (5.7–9.2 s), the power of the system with pure flow control is constantly in excess of the set point.

- (2) Low-speed retracting (9.2–17.2 s): During the initial phase of low-speed retracting, the system maintains FC mode. After the arm cylinder is retracted at a low speed for a certain interval, the arm cylinder reaches the end position at  $t = 14.2$  s, and the

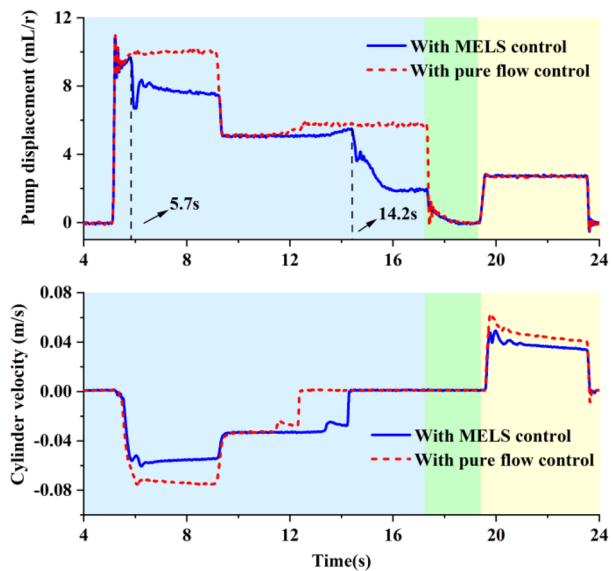
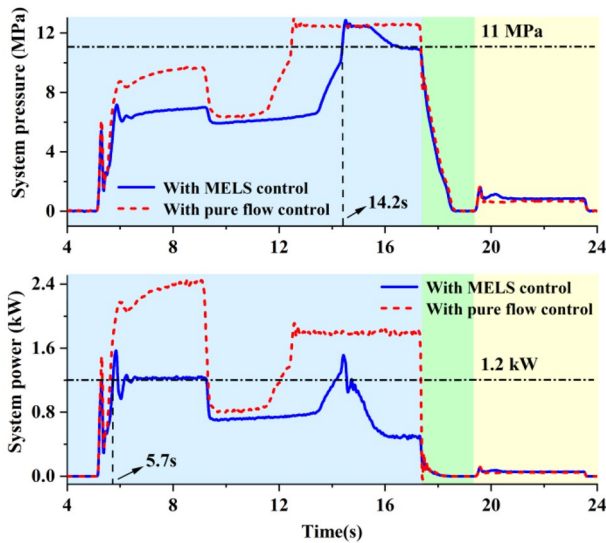


Figure 11 Pump displacement and cylinder velocity

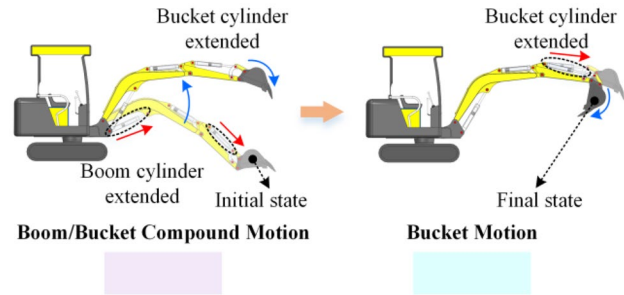




**Figure 12** System pressure and power consumption

system pressure rises until it reaches 11 MPa (upper bound in the PC mode). Therefore, the system with the MELS controller switches from FC mode to PC mode (Rule No. 6), and the system pressure is controlled at around 11 MPa. Accordingly, the pump displacement is reduced. At  $t = 15.2\text{--}17.2$  s, the control signal of the pump is about 0.2 V to compensate for the pump leakage and maintain the system pressure. However, with pure flow control, after the arm cylinder reaches the end stop, the system pressure rises to the setting value of the relief valve (i.e., 12 MPa). The relief valve is open and most supplied oil flows back to the tank. The system pressure is controlled at 12 MPa. Thus, the MELS controller achieves the control functions of power limitation and pressure cut-off, thereby avoiding engine stall and excess pressure.

- (3) Holding (17.2–19.2 s): In this phase, the arm cylinder remains stationary, and the system pressure gradually decreases. When the system pressure falls below 9 MPa (lower bound in the PC mode), the system switches from PC mode to FC mode (Rule No. 5). Subsequently, the system with the MELS controller remains in FC mode.
- (4) Extending (19.2–23.4 s): In this phase, the PC and PL modes are not active because the system pressure and power are low. In addition, the pressure margin of the system does not exceed the set



**Figure 13** Schematic diagram of the boom/bucket compound movement

point of 1.5 MPa, so the LS mode is also not active. Therefore, the system with the MELS controller is controlled in FC mode.

#### 4.4 Experimental Test: Boom/ Bucket Compound Motion

The boom/bucket compound motion test is also carried out to validate the proposed controller. The schematic diagram of the boom/bucket compound movement is shown in Figure 13, which contains two movement phases. The control signals are given in Figure 14. The test results are shown in Figures 15 and 16. In the compound motion test, the power limitation point is set at 3.4 kW.

- (1) Boom/bucket compound motion (5–10 s): In this stage, the boom cylinder extends and the bucket cylinder retracts. At the beginning of the movement, the system is in FC mode. But it can be found that the system pressure rapidly rises. At  $t = 5.26$  s, the system pressure is greater than the set point of 11 MPa. Therefore, the system with the MELS controller switches from FC mode to PC mode (Rule No. 6), the system pressure is maintained at about 11 MPa. At  $t = 6.12\text{--}10$  s, it can be found that the system power is reduced accordingly. This ensures that the system power control is near the value of 3.6 kW, and the root mean square error of power control under steady-state conditions is 0.12 kW, with a control accuracy of 0.17 kW/ 4.7%. However, the power of the system with pure flow control consistently exceeds the set point of 3.4 MPa.
- (2) Bucket motion (10–18 s): In this stage, the boom cylinder movement stops and the bucket cylinder continues to extend until it reaches the end of the

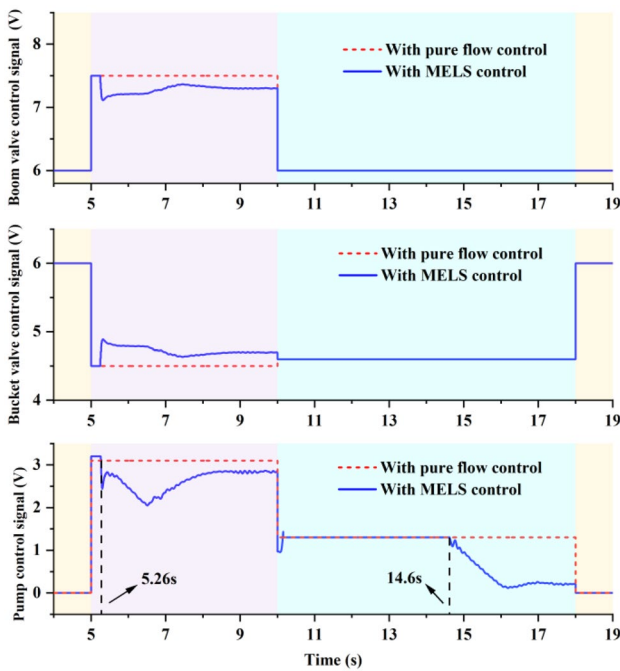


Figure 14 Control signals of the valves and pump

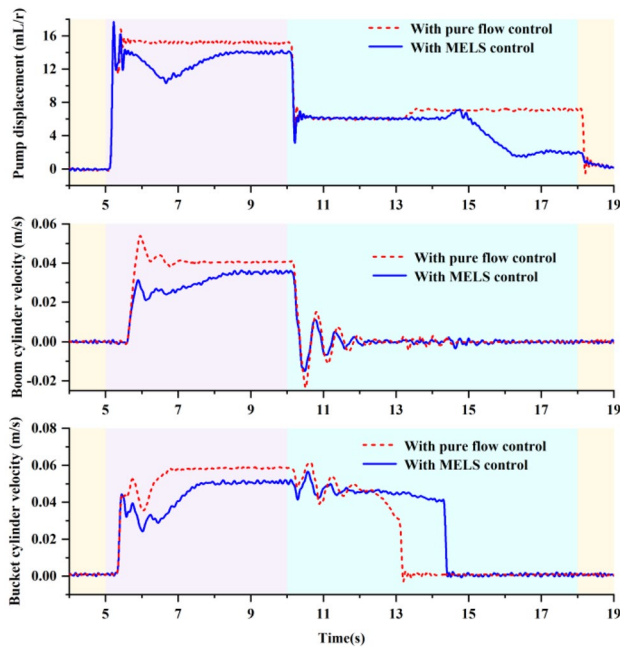


Figure 15 Pump displacement and cylinders velocity

cylinder. At  $t = 10\text{--}14.6\text{ s}$  the bucket cylinder continuously extends, the system pressure gradually decreases. When the system pressure falls below 9 MPa (lower bound in the PC mode), the system switches from PC mode to FC mode (Rule No. 5).

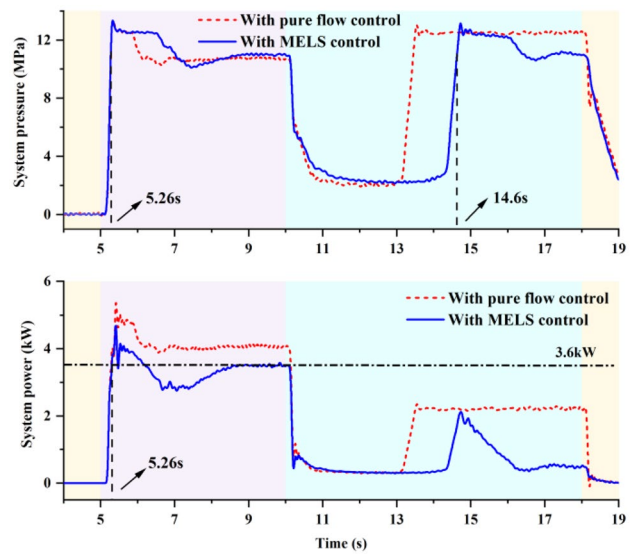


Figure 16 System pressure and system power

Subsequently, the system is in FC mode. After the bucket cylinder piston reaches the end of the cylinder at  $t = 14.6\text{ s}$ , the system pressure rises because the pump displacement is not reduced in time. When the system pressure is greater than the set point of 11 MPa, the PC mode is active. The system switches from FC mode to PC mode (Rule No. 6) to make the pump displacement gradually decreases. Thus, the system pressure is maintained at about 11 MPa. At this stage (i.e., at  $t = 14.9\text{--}18\text{ s}$ ), it can be found that the system power is reduced accordingly. However, with pure flow control, after the bucket cylinder reaches the end of the cylinder, the system pressure rises to the setting value of the relief valve (i.e., 12 MPa). The relief valve is open and most supplied oil flows back to the tank. Then, the system pressure is controlled at 12 MPa. In addition, the power consumption of the system with pure flow control is also higher compared to the proposed controller.

#### 4.5 Simulation Test: Boom Motion under External Load

A comparison simulation under variable external load is also carried out to validate the proposed MELS controller, and the other controller used for comparison is the existing ELS control with pressure control. To ensure the consistency of the external load when using different controllers, a simulation model is conducted by using grey box modeling [31] in the AMESim Software, as shown in Figure 17. The actual parameters of the pump

and valve in the test rig above are identified and utilized to establish the simulation model, including static/dynamic characteristics of the pump and valves (given in Ref. [25]), dimensions of the cylinder, and the mechanical arm (further information can be found in Ref. [30]). As shown in Figure 18, it can be seen that the results of valve flow mapping, pump dynamic response, and boom velocity response in the simulation are consistent with those in the actual test with satisfying accuracy.

Then, this model is used to take a comparative study under the variable external load. The boom cylinder carried out the sequential movement of extending, holding, and retracting, as shown in Figure 19. In

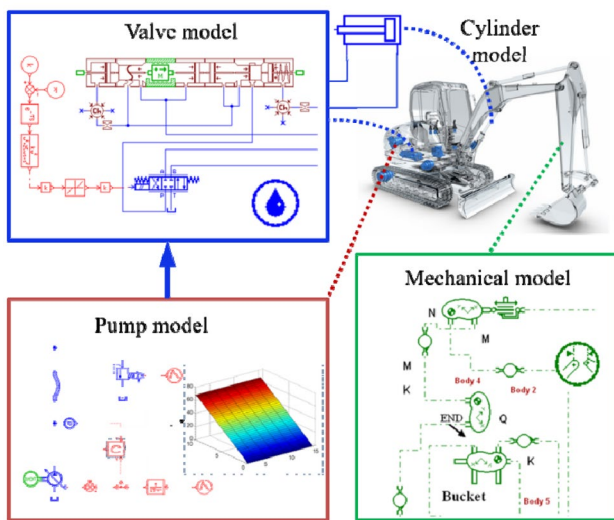


Figure 17 Simulation model in the AMESim Environment

addition, a variable external load force is applied at the endpoint of the bucket at  $t = 15.5\text{--}23\text{ s}$  when the boom cylinder is retracted, and its variation trend is also shown in Figure 19. The simulation results (the system pressure and the cylinder velocity) are shown in Figure 20, respectively.

- (1) Extending and holding: The system of the proposed MELS controller is in FC or LS mode (the system can be switched from FC to LS mode under Rule No. 1 or from LS to FC mode under Rule No. 2), the system with the ELS and pressure control is in LS mode. In these two phases, the system state is relatively smooth under the action of two controllers.
- (2) Retracting: It can be seen that the pressure of the system under the ELS and pressure control fluctuates frequently after  $t = 15.5\text{ s}$ . Because the control mode of the system switches back and forth between LS mode and PC mode, resulting in system pressure oscillation. In contrast, the system with the proposed controller is more stable. After  $t = 16.6\text{ s}$ , the system pressure is kept under 11 MPa to ensure the smooth operation of the system until the end of the boom movement. In addition, the oscillation of the cylinder velocity at the beginning and end of the retracting phase of the boom cylinder is also smaller by using the proposed controller. This further validates the control effectiveness and the smoothness of the switching of the proposed controller.

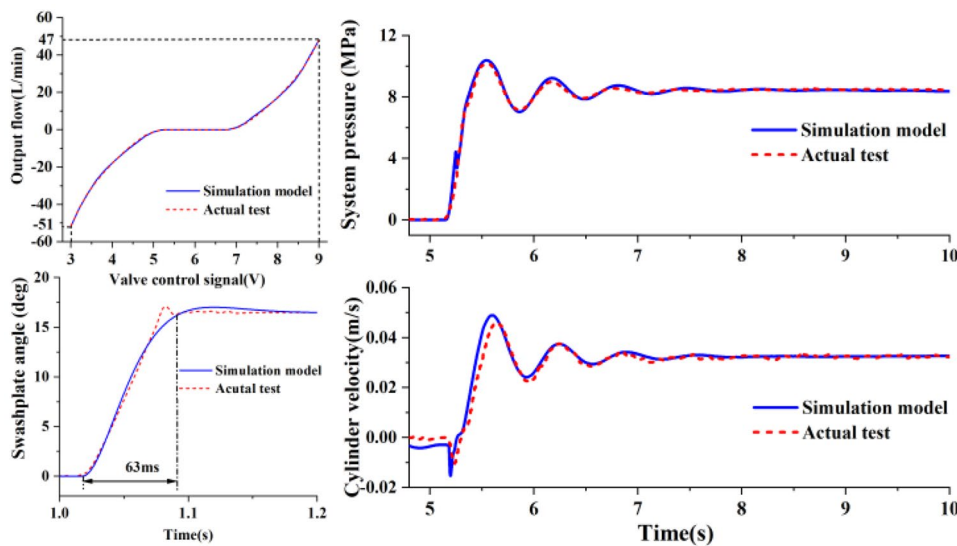
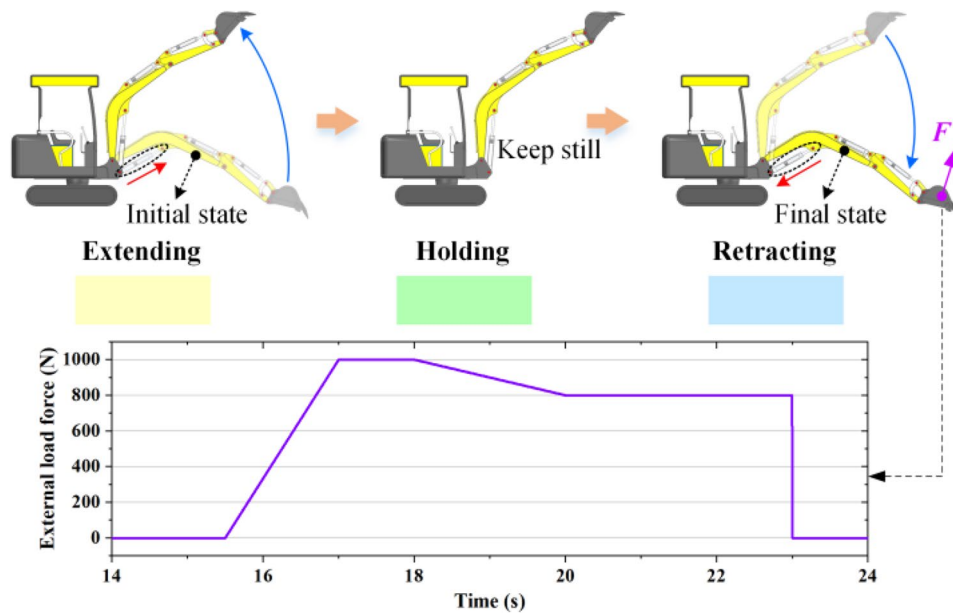
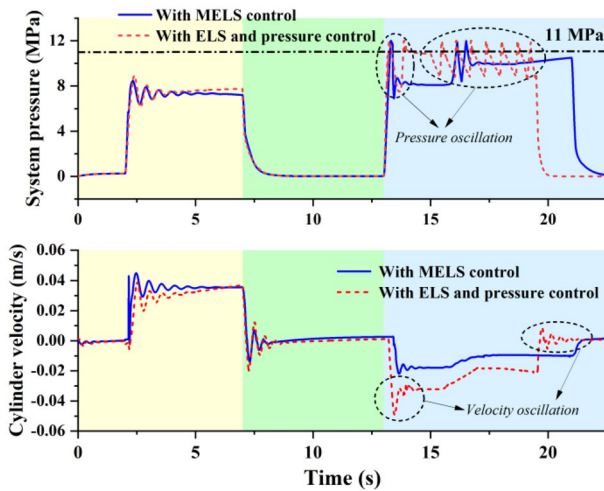


Figure 18 Comparison of simulation and test result



**Figure 19** Schematic diagram of the boom movement with variable external load



**Figure 20** System pressure and cylinder velocity under variable external load

- (2) Combined with defined control priority, an overall switching rule with bilateral and unilateral switching is established for four control modes, including flow control, load sensing, power limitation, and pressure control. A hysteresis switching rule with dwell-time is designed to stably switch between flow control and pressure control mode, and the system stability condition is drawn under pressure control.
- (3) Simulation and experimental tests are conducted on a 2-ton hydraulic excavator. The test results validate the feasibility of the proposed controller under different load conditions to achieve proper flow supplement, power limitation, and pressure cut-off. In addition, pressure fluctuation caused by continuous mode switching can be also avoided.
- (4) Future work will be focused on extending the multi-mode controller into energy-saving hydraulic systems, such as the individual metering system.

## 5 Conclusions

- (1) To meet the complex requirements of variable displacement pumps, this paper develops a multi-mode electronic load sensing control framework for mobile machinery based on our previous work, which integrates three control functions, those are, proper flow supplement, pressure cut-off, and power limitation.

### Acknowledgements

Not applicable.

### Author Contributions

MC and BS wrote the manuscript. RD assisted with sampling and laboratory analyses. BX was in charge of the whole trial. All authors read and approved the final manuscript.

### Authors' Information

Min Cheng, born in 1987, is currently an associate professor at State Key Laboratory of Mechanical Transmissions, Chongqing University, China. He received his Ph. D. degree from Zhejiang University, China, in 2015. His research

interests energy-saving and motion control of electrohydraulic systems, and mechatronic systems design.

Bolin Sun, born in 1998, is currently pursuing the Ph.D. degree in mechanical engineering at the *State Key Laboratory of Mechanical Transmissions, Chongqing University, China*. His research interests include electrohydraulic control systems of mobile machinery and motion control of hydraulic systems.

Ruqi Ding, born in 1987, is currently an associate professor at *East China Jiaotong University, China*. He received his Ph. D. degree from *Zhejiang University, China*, in 2015. His research interests include energy-saving and motion control of electrohydraulic systems in mobile machinery and mechatronic system design.

Bing Xu, born in 1971, is currently a professor and the director at the *State Key Laboratory of Fluid Power and Mechatronic systems, Zhejiang University, China*. He received his Ph. D. degree from *Zhejiang University, China*, in 2001. His research interests include fluid power components and systems, mechatronic systems design, energy-saving, and motion control for mobile machinery. He has authored more than 200 papers and authorized 49 patents. His research interests include fluid power components and mechatronic systems design. Prof. Xu is currently a Chair Professor of the Yangtze River Scholars Program, and a science and technology innovation leader of the Ten Thousand Talent Program.

#### Funding

Supported by National Key Research and Development Program of China (Grant No. 2020YFB2009702), National Natural Science Foundation of China (Grant Nos. 52075055, U21A20124 and 52111530069), Chongqing Natural Science Foundation of China (Grant No. cstc2020jcyj-msxmX0780).

#### Availability of Data and Materials

The datasets supporting the conclusions of this article are included within the article.

#### Competing Interests

The authors declare no competing financial interests.

Received: 7 June 2021 Revised: 7 November 2022 Accepted: 8 February 2023

Published online: 24 February 2023

#### References

- [1] Z M Tong, J Z Miao, Y S Li, et al. Development of electric construction machinery in China: a review of key technologies and future directions. *Journal of Zhejiang University-Science A*, 2021, 22(4): 245–264.
- [2] Z M Tong, S S Wu, S G Tong, et al. Energy-saving technologies for construction machinery: a review of electro-hydraulic pump-valve coordinated system. *Journal of Zhejiang University-Science A*, 2020, 21(5): 331–349.
- [3] H K Wang, P G Leaney. Modelling and energy efficiency analysis of a hybrid pump-controlled asymmetric (single-rod) cylinder drive system. *International Journal of Hydromechanics*, 2020, 3(1): 1–25.
- [4] M Cheng, J H Zhang, B Xu, et al. Anti-windup scheme of the electronic load sensing pump via switched flow/power control. *Mechatronics*, 2019, 61: 1–11.
- [5] S Q Luo, M Cheng, R Q Ding, et al. Human–robot shared control based on locally weighted intent prediction for a teleoperated hydraulic manipulator system. *IEEE/ASME Transactions on Mechatronics*, 2022, <https://doi.org/https://doi.org/10.1109/TMECH.2022.3157852>.
- [6] B Xu, J Shen, S H Liu, et al. Research and development of electro-hydraulic control valves oriented to Industry 4.0: A review. *Chinese Journal of Mechanical Engineering*, 2020, 33: 29.
- [7] S G Ye, J H Zhang, B Xu, et al. A theoretical dynamic model to study the vibration response characteristics of an axial piston pump. *Mechanical Systems and Signal Processing*, 2021, 150, 107237.
- [8] R H Hansen, T O Anderson, H C Pederson. Development and implementation of an advanced power management algorithm for electronic load sensing on a telehandler. *ASME Symposium on Fluid Power and Motion Control*, 2010: 537–550.
- [9] T Lin, Y Lin, H Ren, et al. A double variable control load sensing system for electric hydraulic excavator. *Energy*, 2021, 223(27): 119999.
- [10] L Ge, L Quan, X G Zhang, et al. Power matching and energy efficiency improvement of hydraulic excavator driven with speed and displacement variable power source. *Chinese Journal of Mechanical Engineering*, 2019, 32: 100.
- [11] G P Jayaraman, S V Lunzmann. Modeling and analysis of an electronic load sensing pump. *2011 IEEE International Conference on Control Applications (CCA)*, Denver, USA, September 28–30, 2011: 82–87.
- [12] Z N Song, Z X Jiao, Y X Shang, et al. Design and analysis of a direct load sensing electro-hydrostatic actuator. *2015 International Conference on Fluid Power and Mechatronics (FPM)*, Harbin, China, August 5–7, 2015: 624–627.
- [13] J Wei, K Guo, J Fang, et al. Nonlinear supply pressure control for a variable displacement axial piston pump. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 2015, 229(7): 614–624.
- [14] W Shen, J Wang. A robust controller design for networked hydraulic pressure control system based on CPR. *Peer-to-Peer Networking and Applications*, 2019, 12(6): 1651–1661.
- [15] S H Park, J M Lee, J S Kim. Robust control of the pressure in a control-cylinder with direct drive valve for the variable displacement axial piston pump. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*. 2009, 223(4): 455–465.
- [16] S Wang. Generic modeling and control of an open-circuit piston pump-part ii: control strategies and designs. *Journal of Dynamic Systems Measurement and Control*, 2016, 138(4): 041005.
- [17] W Kemmetmüller, F Fuchshumer, A Kugi. Nonlinear pressure control of self-supplied variable displacement axial piston pumps. *Control Engineering Practice*, 2010, 18(1): 84–93.
- [18] K Guo, Y Xu, J Li. A switched controller design for supply pressure tracking of variable displacement axial piston pumps. *IEEE Access*, 2018, 6: 3932–3942.
- [19] Y Wang, T Shen, C Tan, et al. Research status, critical technologies, and development trends of hydraulic pressure pulsation attenuator. *Chinese Journal of Mechanical Engineering*, 2021, 34: 14.
- [20] M Ruggeri, M Guidetti. Variable load sensing and anti-stall electronic control with sliding mode and adaptive PID. *Proceeding of the JFPS international symposium on fluid power*, 2008(7–2): 301–306.
- [21] Y H Anis, S A Kassem. Performance of constant power operated swash plate axial piston pumps with fuzzy logic controllers. *Proceedings of the ASME 2013 International Mechanical Engineering Congress and Exposition*, California, USA, November 15–21, 2013, V04AT04A016.
- [22] A Nobuei, I Kouji, K Hideo, et al. Pump torque control system for hydraulic construction machine. US, 8424298B2, 2013–04–23. <https://www.freepatentsonline.com/8424298.html>.
- [23] K Ligenfelter, G Lafayette. Electronic torque and pressure control for load sensing pumps. WO, 2015140622A1, 2015–09–24, <https://www.freepatentsonline.com/WO2015140622.html>.
- [24] K R Ligenfelter, A Bruns, C Daley, et al. Electronic load sense control with electronic variable load sense relief, variable working margin, and electronic torque limiting. US, 9759212B2, 2017–09–12. <https://www.freepatentsonline.com/9759212.html>.
- [25] B Xu, M Cheng, H Yang, et al. A hybrid displacement/pressure control scheme for an electrohydraulic flow matching system. *IEEE/ASME transactions on Mechatronics*, 2015, 20(6): 2771–2782.
- [26] M Axin, B Eriksson, P Krus. Flow versus pressure control of pumps in mobile hydraulic systems. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 2014, 228(4): 245–256.
- [27] R Q Ding, M Cheng, L Jiang, et al. Active fault-tolerant control for electro-hydraulic systems with an independent metering valve against valve faults. *IEEE Transactions on Industrial Electronics*, 2021, 68(8): 7221–7232.
- [28] M Cheng, S Q Luo, R Q Ding, et al. Dynamic impact of hydraulic systems using pressure feedback for active damping. *Applied Mathematical Modelling*, 2021, 89: 454–469.

- [29] H E Merritt. *Hydraulic control systems*. New York: John Wiley & Sons, Inc., 1967.
- [30] W Liu. *Investigation into the characteristics of electrohydraulic flow matching control systems for excavators*. Hangzhou: Zhejiang University, 2011. (in Chinese).
- [31] P Casoli, A Anthony. Gray box modeling of an excavator's variable displacement hydraulic pump for fast simulation of excavation cycles. *Control Engineering Practice*, 2013, 21(4): 483–494.

**Submit your manuscript to a SpringerOpen<sup>®</sup> journal and benefit from:**

- ▶ Convenient online submission
- ▶ Rigorous peer review
- ▶ Open access: articles freely available online
- ▶ High visibility within the field
- ▶ Retaining the copyright to your article

---

Submit your next manuscript at ► [springeropen.com](https://www.springeropen.com)

---