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

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ORIGINAL ARTICLE

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A multi-mode electronic load sensing control scheme with power limitation and pressure control for mobile machinery

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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. Combined with defined control priority, a switching rule including bilateral and unilateral switching for different modes is then established according to the actual requirements of mobile machinery. A comparative study was 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 control, which has good stability performance when switching between different control modes.

Keywords: Hydraulic control • Load sensing • Power limitation • Mobile machinery

1 Introduction

To reduce energy consumption in mobile machinery, variable displacement pumps have been widely used to meet flow requirements 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 [1], which leads to complex structures and limits control parameters.

In recent years, electrification of mobile machinery has been an urgent requirement to improve its automation level. Therefore, hydro-mechanical pumps are increasingly replaced by the 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 [3-4]. Advanced 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 [5–7]. Song et al. [8] 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 network or fuzzy algorithms [9-12]. 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 [13-15]. Moreover, integrating more control functions into ELS pumps has been continuously explored in academic and industrial areas. Ruggeri et al designed an electronic controller to integrate the control functions of power/torque limitation and variable load sensing [16]. Also, a fuzzy controller has been designed to improve the pump dynamic performance and fulfill the flow/power control demands [17]. Besides, some patents [18-20] about integrated load sensing control pumps can be 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,

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and only the switching stability about the pump itself is mentioned [13-14]. 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 [13, 21]. Our previous work has shown that the integral windup issue in ELS controllers probably leads to underlying instability due to improper operation [1]. 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 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 the system stability when switching between different modes. The paper is organized as follows: The system layout is briefly introduced in Section 2. Then, Section 3 depicts the MELS control framework. In Section 4, a comparative experimental study is carried out. Finally, Section 5 summarizes this study.

2 System Layout

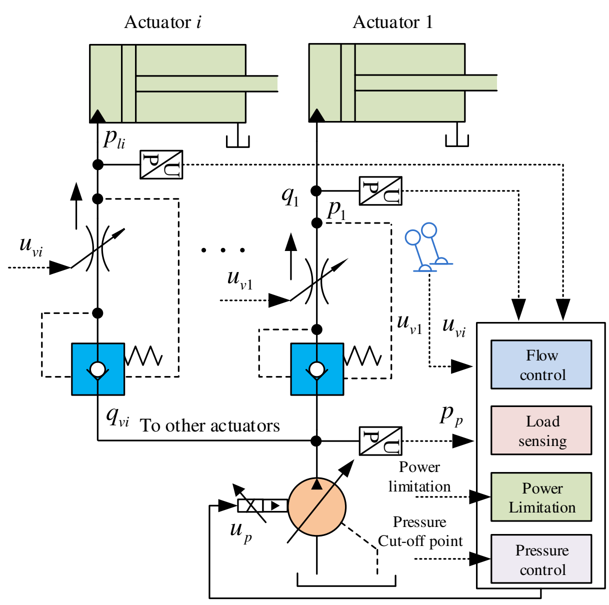

Figure 1 Hydraulic system with primary pressure compensation

As depicted 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. A 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 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 our previous work ([1], [20]), 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 (FC)

In the 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 [22-23]. Neglecting the pump leakage, the control signal can be written as

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

where C_d the discharge coefficient, u_{vi} the valve signal, Δp_{nom} the nominal pressure, ρ the oil density, n_p the pump rotational speed, and k_{pp} 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 τ_p is the time constant.

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 [21]. 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

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

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. e_{pr} is the pressure margin error, which can be expressed as

$$e_{pr}(t) = p_{dm} - p_{pm}(t) = p_{dm} - \left[p_p(t) - p_{lm}(t)\right] \quad (5)$$

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 is expressed by:

$$p_{lm}(t) = \max\left[p_{l1},..., p_{li},..., p_{ln}\right](i = 1,\ldots,n) \quad (6)$$

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 $q_p = P_n / p_p$. The power controller is then designed as

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

where u_{vlc} is the compensator based on dynamic pressure feedback to achieve active damping control [24], $\hat{G}_p$ is the identified pump model. The Laplace form of u_{vlc} is given by:

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

where s is the Laplace operator. k_{cp} and ω_{cp} are the control gain and the cut-off frequency of the compensator, respectively. The pump dynamic is modeled as a first-order term [1], which is given by:

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

where τ_c is the time constant.

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, the incremental control is defined to be active when the supplied flow is not excessive ($p_{lm}(t) \le p_{dm}$). Thus, the incremental PID controller is designed by:

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

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

$$\phi(t) = \begin{cases} 1, p_{lm}(t) \le p_{dm} \\ 0, p_{lm}(t) > p_{dm} \end{cases} \quad (11)$$

The pressure error e_{pc} is defined as

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

where p_{dc} is the desired pressure. 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 our previous work [1] and [21], a multi-mode switching rule is designed to avoid potential stability caused by continuous switching between different modes, as shown in Figure 2.

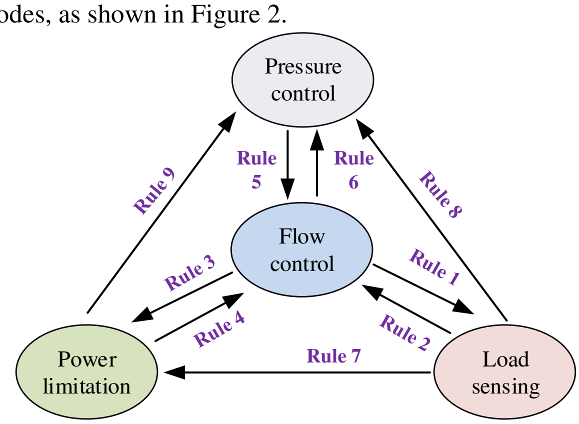

Figure 2 Switching rule of the MELS controllers

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 the switching from LS to PC is not allowed. For the 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, the modes in the first column mean those after switching.

Table 1 Switching rule of the controller

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

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

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

where p_{dc} and p_{db} 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.

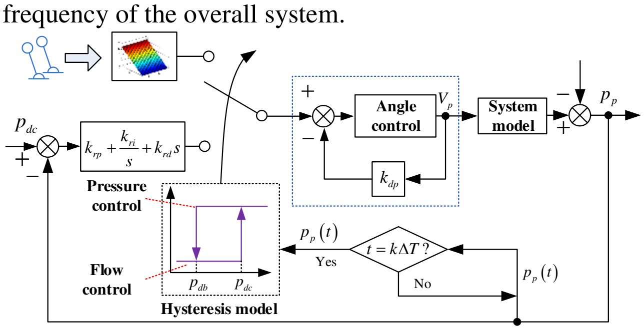

Figure 3 Block diagram of hysteresis switching rule

The switching rule No. 7 is defined as $p_p(t) < p_{dc}$ and $u_{pl}(t) \ge u_{pp}(t)$. The unidirectional switching rules No. 8 and No. 9 are defined as $p_p(t) \ge p_{dc}$. 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.

Table 2 Priority of the multi-mode controller

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

3.6 Stability Analysis

One concern is whether the 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 [20]. Rules 3 and 4 is 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 [1]. 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:

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

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

where β_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 capside chamber, A_1 is the effective area of capside chamber, v_1 is the cylinder velocity. The flow equation of the valve is expressed by:

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

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 rodside chamber is neglected, so the force balance equation of the cylinder is simplified as

$$(m_1 s + b_1) v_1(s) = A_1 p_1(s) - F_e(s) \tag{18}$$

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 Equations (9)-(10) and (14)-(18) as:

$$\frac{p_p(s)}{u_v(s)} = -\frac{k_q k_v V_1 \beta_e s^2 (1+\tau_c s)}{N_0 s^4 + N_1 s^3 + N_2 s^2 + N_3 s + N_4} \tag{19}$$

where

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

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

$$\frac{p_p(s)}{u_v(s)} = -\frac{k_q k_v V_1 \beta_e s (1+\tau_c s)}{N_0 s^3 + N_1' s^2 + N_2' s + N_3'} \tag{20}$$

where $N_1' = V_1 V_{pi} + k_{pp} k_{rd} n_p V_1 \beta_e$, $N_2' = k_{pp} k_{rp} n_p V_1 \beta_e$ $N_3' = k_{pp} k_{ri} n_p V_1 \beta_e$. Eq. (20) is a third-order system actually. Based on the Routh stability criterion, its stability condition can be expressed as

$$N_0, N_1', N_2', N_3' > 0 \tag{21}$$

$$(N_1' N_2' - N_0 N_3') / N_1' > 0 \Leftrightarrow N_1' N_2' - N_0 N_3' > 0 \tag{22}$$

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. 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 [25]. Motion control tests were then carried out and the main parameters are listed in Tables 3 and 4.

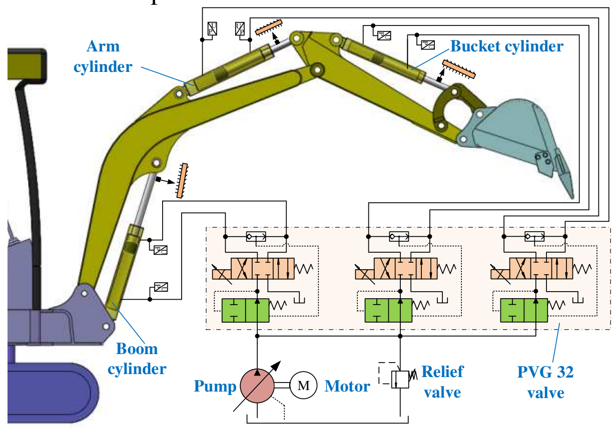

Figure 4 Hydraulic schematic of the test rig

Table 3 Main parameters of the test rig

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

Table 4 Parameters in the proposed controller

Parameters	Symbol	Value	Unit
Power limitation point	P_n	1.2	kW
Preset pressure margin in the LS mode	p_{dm}	1.5	MPa
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}	11/9	MPa
Switch dwelling time	ΔT	0.5	s

4.2 Experimental Test: Boom Motion

Firstly, valve control signals are given to complete the typical reciprocating movement of the boom. The boom cylinder carried out the movement of low-speed extending, high-speed extending, holding, and retracting. The system with pure flow control is taken as a comparison to validate the proposed controller. The schematic diagram of the movement is given in Figure 5. The control signals of the valve and pump are also shown in Figure 5, and Figures 6 and 7 show the test result.

The system with pure flow control is the first one that reaches the cylinder end and stop moving because of its fast speed. It can be found that at t=8.05 s, the system with the MELS controller switches into the PL mode, so both the pump displacement and cylinder velocity decrease. At the same time, the flow rate across the multi-way control valve decreases, which reduces the system pressure accordingly. Until t=12.36 s, the system switches from PL into LS mode. In the period t=8.05~12.36 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.

During the lowering movement of the boom, the system pressure is relatively low and the PL mode is not active. However, due to large pressure fluctuations, the switching condition of the LS controller is triggered (the pressure margin is greater than the setting value of 1.5 MPa), so the system is implemented in the LS mode.

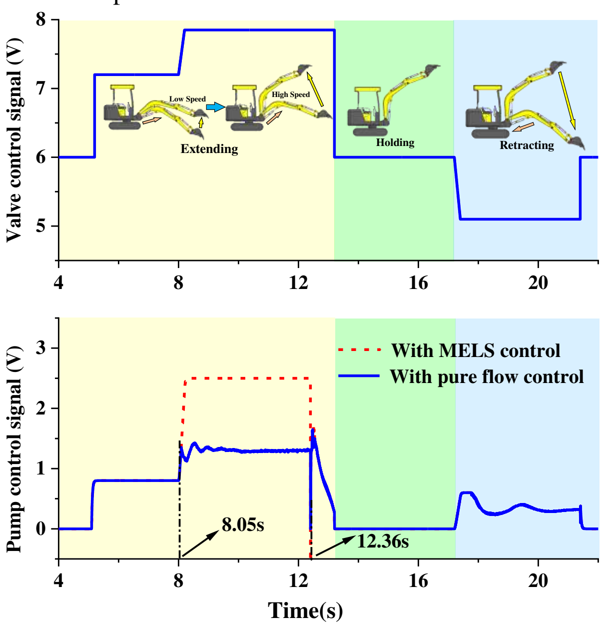

Figure 5 Control signals of the valve and pump

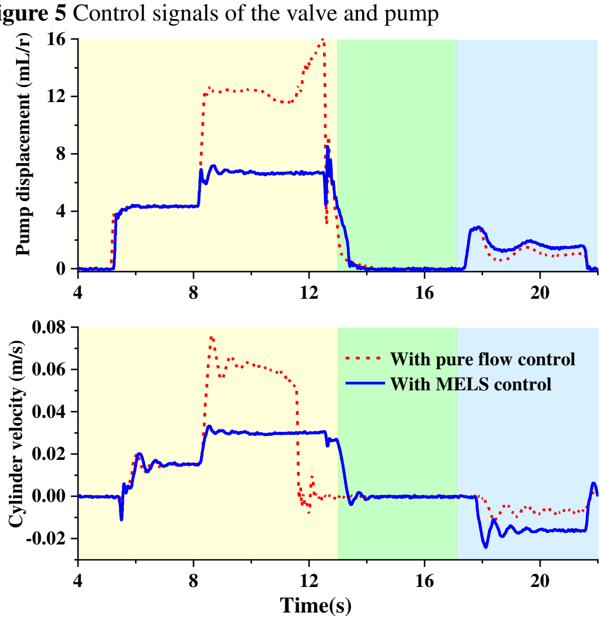

Figure 6 Pump displacement and actuator velocity

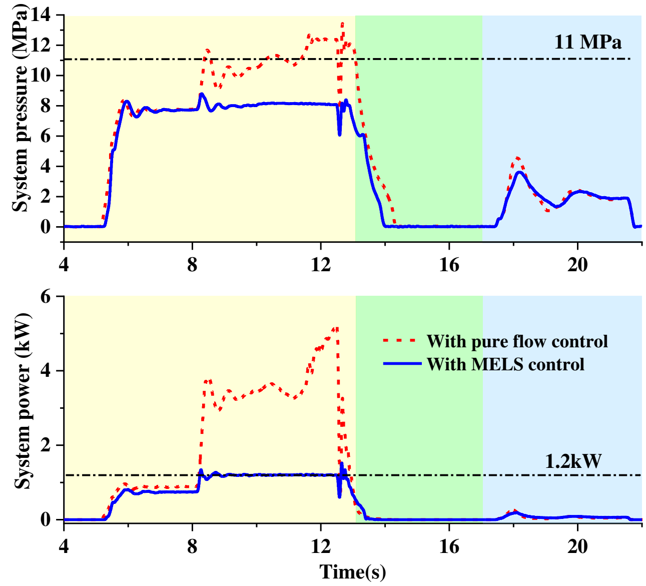

Figure 7 System pressure and Power consumption

4.3 Experimental Test: Arm Motion

Similarly, the typical working cycle of the arm cylinder is tested, which takes a cyclic movement of high speed retracting, low-speed retracting, holding, and extending. The cylinder movement is shown in Figure 8 as well as the control signals. The results are shown in Figures 9 and 10.

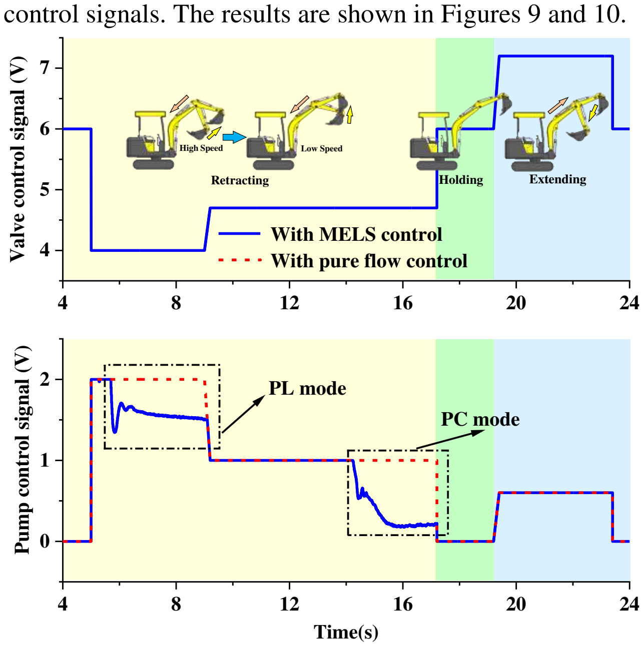

Figure 8 Control signals of the valve and pump

When the arm cylinder is retracted at a high speed, the power controller is triggered, and the pump displacement is reduced, so the system power maintains near the setting value 1.2 kW. After the arm cylinder is retracted at a low speed for a certain interval, the arm cylinder reaches the end position, and the system pressure rises until it reaches

11 MPa (upper bound in the PC mode). Therefore, the PC mode is active, and the pump displacement is reduced, so the system pressure is controlled around 11MPa. At this time, the control signal is about 0.2V to compensate the pump leakage and maintain the system pressure. Without the MELS controller, after the arm cylinder reaches the end stop, the system pressure rises to the setting value of the relief valve (12 MPa), and most supplied oil flows back to the tank. Thus, the MELS controller achieves the control functions of power limitation and pressure cut-off, thereby avoiding engine stall and excess pressure.

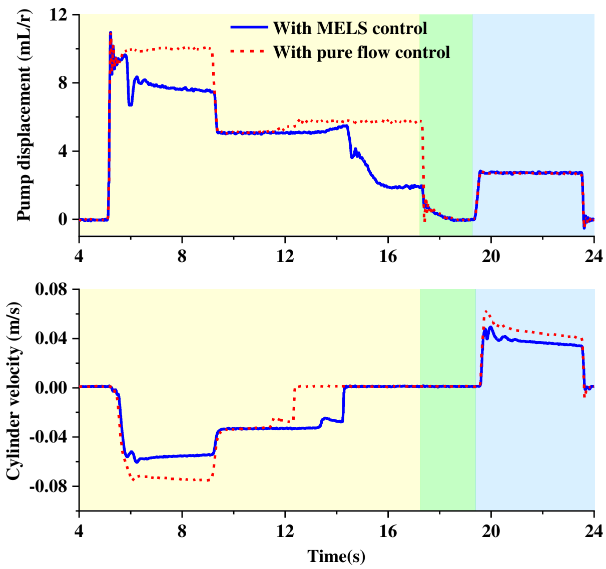

Figure 9 Pump displacement and cylinder velocity

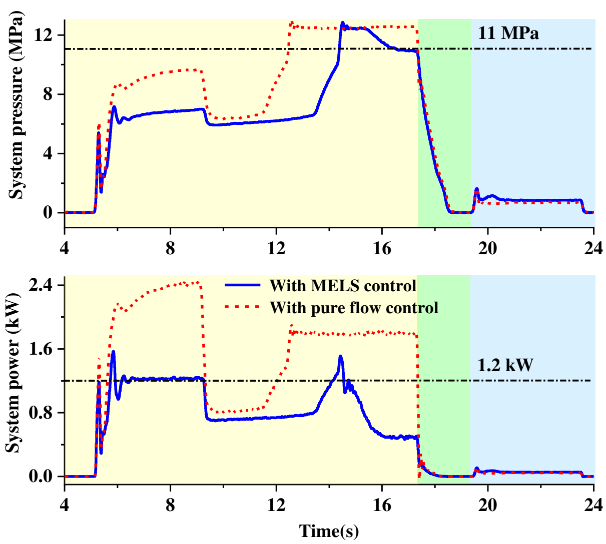

Figure 10 System pressure and power consumption

4.4 Simulation Test: Motion Under External Loads

A comparison study under different external loads is also carried out to validate the proposed controller. To ensure the consistency of the external loads when using different controllers, we conducted a simulation model using grey box modeling [26] by the AMESim Software, as shown in Figure 11. 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 characteristic of the pump and valves (given in [21]), dimensions of the cylinder and the mechanical arm (further information can be found in [25]). As depicted in Fig. 12, it is 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.

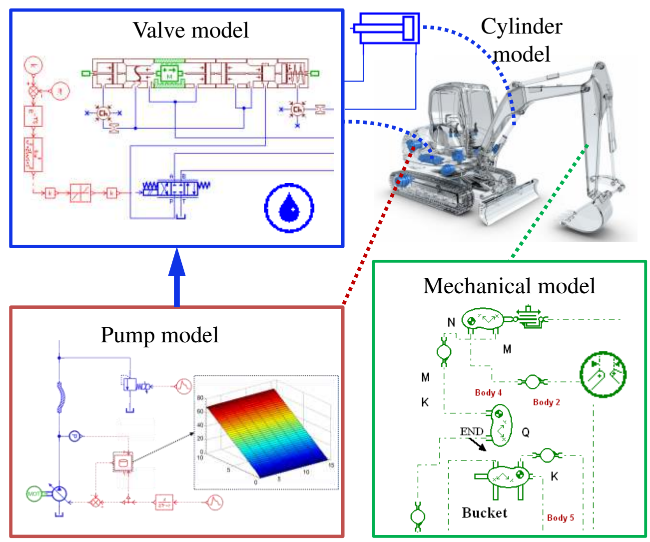

Figure 11 Simulation model in the AMESim Environment

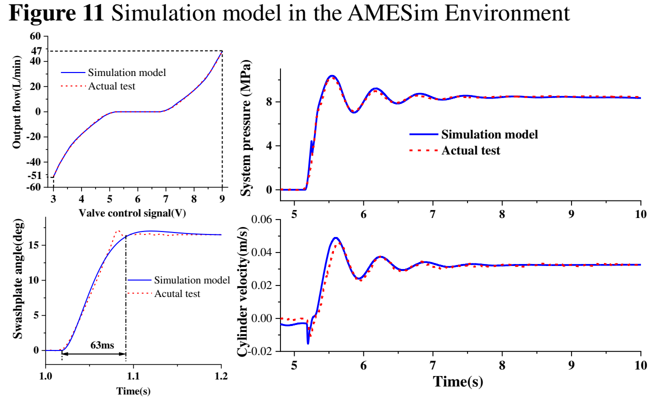

Figure 12 Comparison of simulation and test result

Then, this model is utilized to take a comparative study under different external loads. The boom cylinder carried out the sequential movement of extending, holding, and retracting, as shown in Figure 13. A loading force F=800 N is applied at the endpoint of the bucket at 13 s and then changes to 1000 N at 15.5 s when the boom cylinder is retracted. Two controllers are used for comparison: one is the proposed controller, and the other is the existing ELS control with pressure control. Figures 13 and 14 show the system pressure and the cylinder velocity, respectively. It is seen that the system pressure fluctuates frequently when the boom cylinder is retracted after 15.5 s, since the controller switches between ELS and pressure control continuously. In contrast, the hydraulic system is more stable when with the proposed ELS controller. The velocity using the two controller seems more stable because the flow that goes into the cylinder is limited by the pressure compensator.

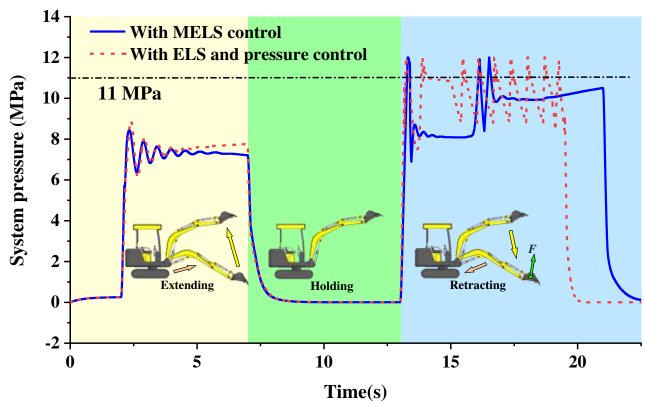

Figure 13 System pressure under different loads

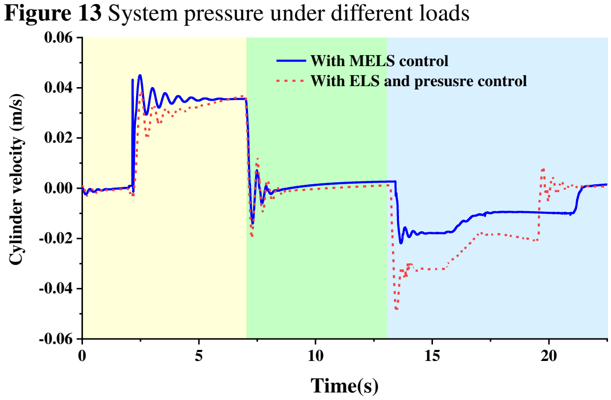

Figure 14 Cylinder velocities under different loads

5 Conclusion

To meet 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 integrating three control functions of proper flow supplement, pressure cut-off, and power limitation. 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 and pressure control, and the system stability condition is drawn under pressure control. Simulation study and experimental tests were 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. Future work will be focused on extending the multi-mode controller into energy-saving hydraulic systems, such as individual metering systems.

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6 Declaration

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Availability of data and materials

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

Authors' contributions

The authors' contributions are as follows: 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.

Competing interests

The authors declare no competing financial interests.

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