A new Electrohydraulic Load Sensing Control System for Hydraulic Excavators

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Abstract

In order to obtain an advanced operation performance and an improved energy efficiency of hydraulic systems in excavators, a new electrohydraulic load sensing system (EHLS) is introduced, which works in a synchronous of electro-proportional valves and electro-proportional pumps and open-loop control mode. An experimental prototype of 2-ton excavtor equipped with the EHLS system has been developed and a corresponding virtual prototype model has been built based on AMEsim and Adams. Compared with the hydraulic-mechanism load-sensing (HMLS) system, the stability, response and energy-saving performance of the EHMC system were investigated by simulation and experiment. A new method of pressure and flow compound control is proposed. Using the methods, the problems of overflow and energy loss were solved, and the pressure impact was effectively inhibited.

KEYWORDS: electrohydraulic load sensing, hydraulic excavators, energy consumption, response

1. Introduction

Hydraulic-mechanical load sensing (HMLS) control systems are widely equipped in hydraulic excavators owing to the improved operation performance and energy efficiency compared with constant pressure or constant flow systems. In HMLS systems, the pump pressure is continuously adjusted to match the highest load pressure. The pressure drops between the pumps and the highest load pressures are in the range of 2 MPa, even to 3 MPa normally, so the energy loss caused by the load-sensing is considerable. On the other hand, the HMLS systems may become oscillatory in some cases or even unstable.

In order to obtain an advanced operation performance and an improved energy efficiency of hydraulic systems in hydraulic excavators, a new electrohydraulic load sensing (EHLS) control system (**Figure 1**) is introduced, in which electro-proportional valves and electro-proportional pumps work in a parallel control mode. It improves the delayed response between the pump and valve in the traditional HMLS systems. The closed-loop feedback control of pressure as well as the preset pressure margin between pump and highest load pressure is not needed normally. The pump displacement is electrically controlled to provide the sum of requested flows, which are calculated according to the valve control signals.

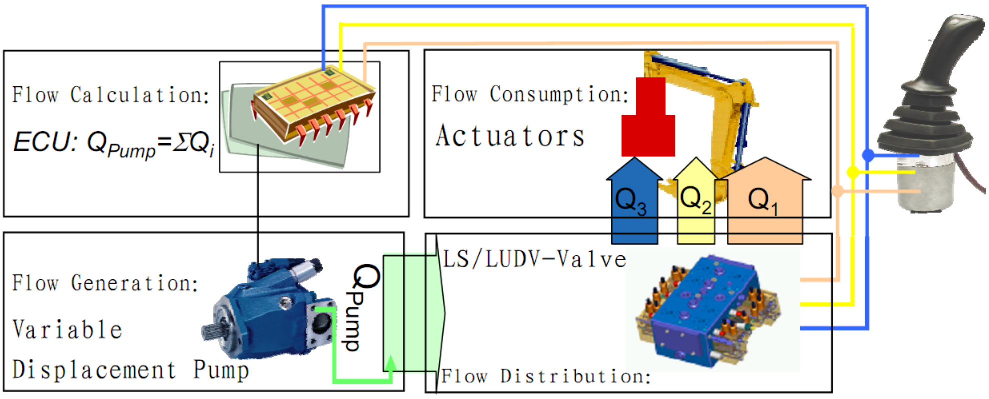


Figure 1: Principle of the EHLS system

The flow demand ocf each actuator is:

$$Q_i = C_q \cdot W_i \cdot X_{vi} \sqrt{\frac{2}{\rho}(\Delta P)} = C_q \cdot W_i \cdot X_{vi} \sqrt{\frac{2}{\rho}(\frac{F_0}{A_{c0}})} \tag{1}$$

The sum of requested flow of actuators is:

$$Q = \sum_{i=1}^{n} Q_i = C_q \cdot \sum_{i=1}^{n} (W_i \cdot X_{vi}) \cdot \sqrt{\frac{2}{\rho}(\frac{F_0}{A_{c0}})} \qquad i = [0, \cdots, n] \tag{2}$$

The flow for actuators that supplied by variable displacement pump is:

$$Q_s = V \cdot n \cdot \eta \tag{3}$$

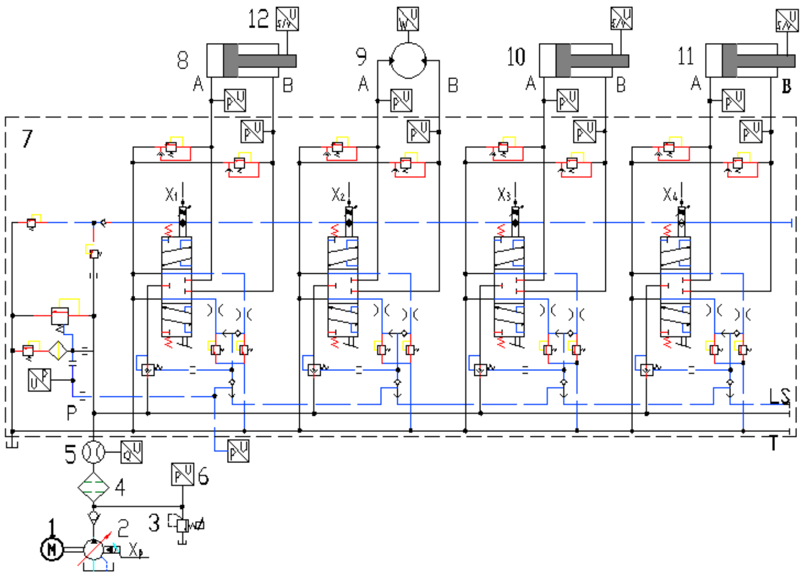
Under the ideal conditions, the flow for actuators supplied by pump is equal to the sum of requested flow of actuators.

Several investigations have been published already on this topic. Zähe developed the closed loop sum-of-flow control /1/, in which the velocities of the actuators constitute the main feedback signals for pump and valve control. The electronic flow matching is investigated using a non-flow-sharing valve by Djurovic /2,3/. However, secondary compensators and compensator position sensors are added in the system in order to provide feedback for stabilizing the system. In order to improve the energy efficiency of mobile machines, an electrohydraulic dual-circuit system has been proposed based on electronic flow matching by Finzel /4,5/, in which due to a separation of the often parallelly operated actuators in different hydraulic circuits the energy losses can be reduced. The alternating pump control for a load-sensing system was studied by Grösbrink /6,7/, in which a pressure difference sensor and an electronic swash plate angle controller were needed, and the dynamic system transfer behavior between the HMLS and the system with the electro hydraulic pump controller was investigated. In this paper, the objective of the new electro hydraulic load sensing control system is to obtain advanced operation performance and an improved energy efficiency by complete electronic control with minimum sensors based on the help of the pressure and flow closed-loop control pump.

2. Simulation model and test-rig

2.1. Test-rig

Figure 2 shows the schematic hydraulic circuit of the EHLS system with an electro-proportional multi-way valve and an electro-proportional variable displacement pump. In order to get a comparison between the EHLS system and a traditional hydro-mechanical LS-system, the electro-proportional variable displacement pump is replaced by a hydro-mechanical load sensing pump when testing the LS-system.



1. motor; 2. variable displacement pump with electronic control; 3.relief valve; 4.filter; 5.flow sensor; 6.pressure sensor; 7.PVG32 ; 8.arm cylinder; 9.swing motor; 10.bucket cylinder; 11. boom cylinder; 12. displacement /velocity sensor

Figure 2: Schematic diagram of the EHLS system

A 2-ton hydraulic excavator has been used as a test-rig, see **Figure 3**. The machine is equipped with a LS-system and the propose system to be tested on the same machine.

A real time control and data-acquisition system for hydraulic system control has been developed and constructed by using the software package (MATLAB/Simulink, xPC Target, Real-Time Workshop and VC++ compiler) and commercially available hardware (multifunction I/O boards and power amplifiers, etc.). By using xPC Target and the supported I/O boards, the complex system can be controlled in a neat way. Furthermore, there has no requirement on low level language programming nearly.

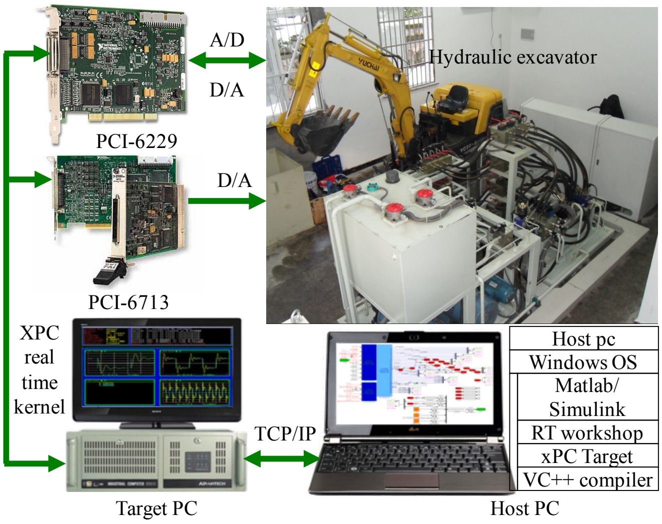


Figure 3: Test rig of 2-ton hydraulic excavator prototype

2.2. Simulation model

An excavator consists of several hydraulic components which include pumps, a main control valve and cylinders etc. It operates by the linkage of the hydraulic components and manipulator. The simulation model of hydraulic systems is built in AMESim software by combining the dynamics analysis of excavator's manipulator in ADAMS software. The AMESim model exchanges the data of force displacement and velocity with ADAMS through co-simulation interface shown in **Figure 4**.

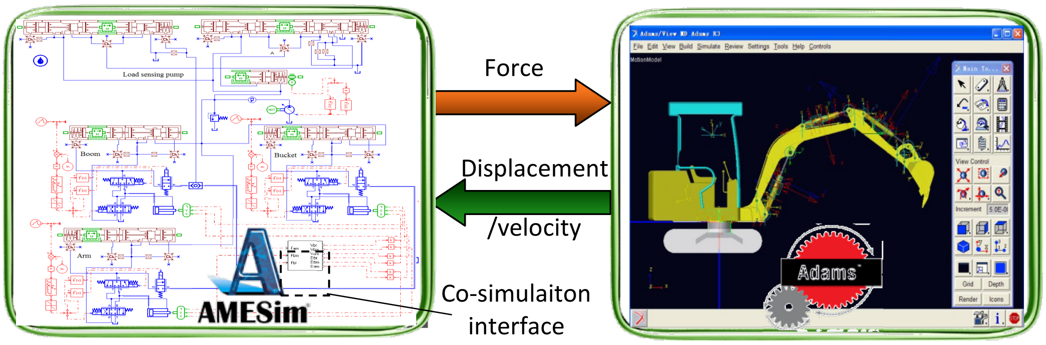


Figure 4: AMESim-Adams co-simulation model

3. Simulation and experiment results

3.1. Results of static and dynamic characteristics

The comparison between the simulated and experimental results of the HMLS system and EHLS system is shown in **Figure 5**. The simulated results are in accordance with the experimental results. It showed that the model was correct.

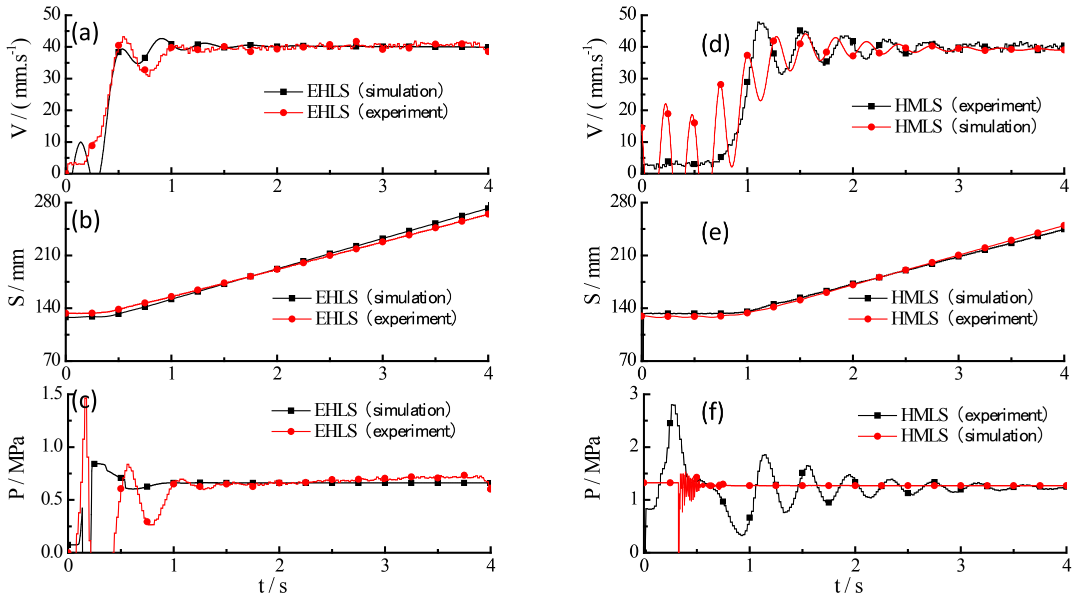
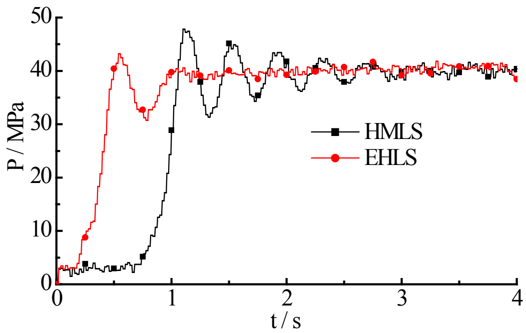
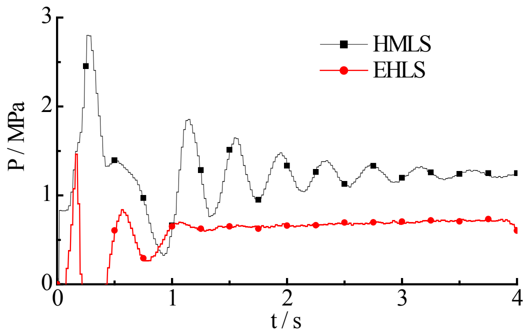


Figure 5: Comparison between the simulated and experimental results of the HMLS system and EHLS system ((a), (d) velocity curves; (b), (e) displacement curves; (c), (f) pressure margin curves)



(a) Velocity curves



(b) Pressure curves

Figure 6: Comparison between the HMLS system and EHLS system (Experiment)

Figure 6 shows the experiment results of the comparison between the HMLS system and EHLS system when measured step response of the boom lifting. The load sensing is a feedback system with risk of instability. It is also a system with low damping. As shown in Figure 5 and Figure 6, the oscillations of velocities and system pressures occur in the load sensing system. Compared with the HMLS system, the systemic performance of dynamic response and stability in the EHLS system were greatly improved. The pressure margin of the EHLS system was reduced by more than 0.6-0.7 MPa, and energy efficiency was improved.

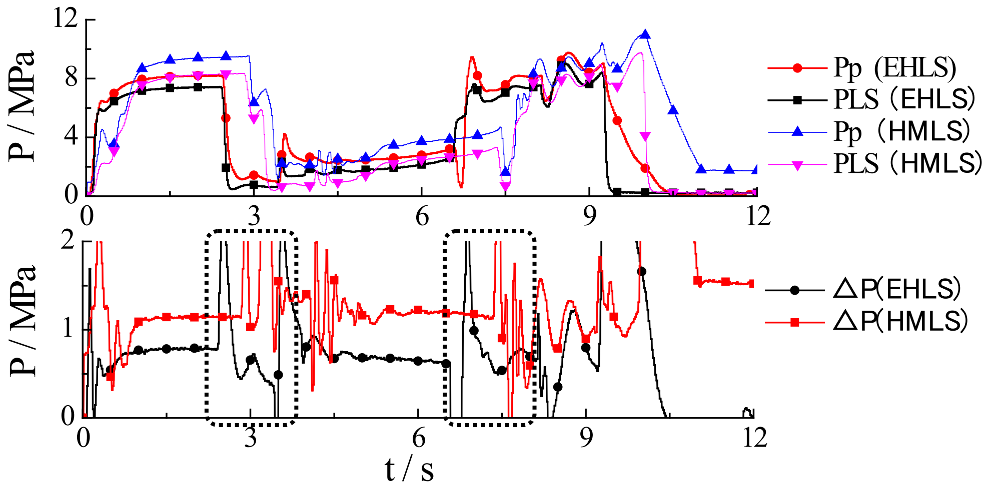
3.2. Energy consumption and work efficiency of defined excavating cycle

(a) (b) (c) (d) (e)

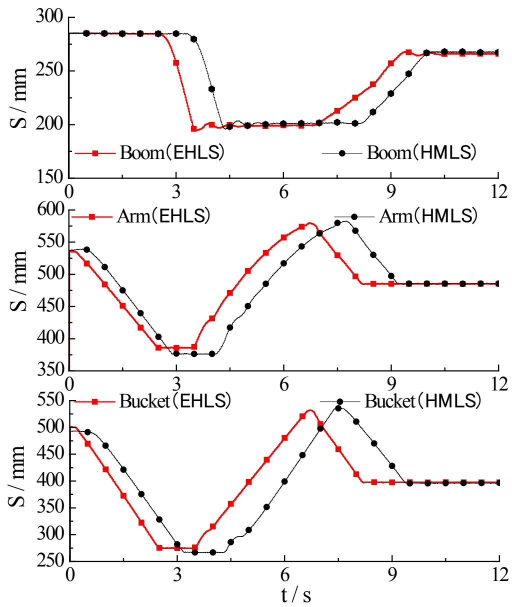
Figure 7: Excavating test cycle

The working cycle of digging and dumping were tested without swing function (**Figure 7 :(a)~(c)**). **Figure 8 (a)** shows the experimental results regarding the system pressures and the maximum load pressures. The results indicated that the EHLS system has a more improved capability than the HMLS system. When compared with the conventional LS system of the excavating cycle, the pressure margin is reduced by more than 0.6-0.7 MPa, the energy consumption is reduced by up to 10 % (According to equation (4)) in the EHLS system.

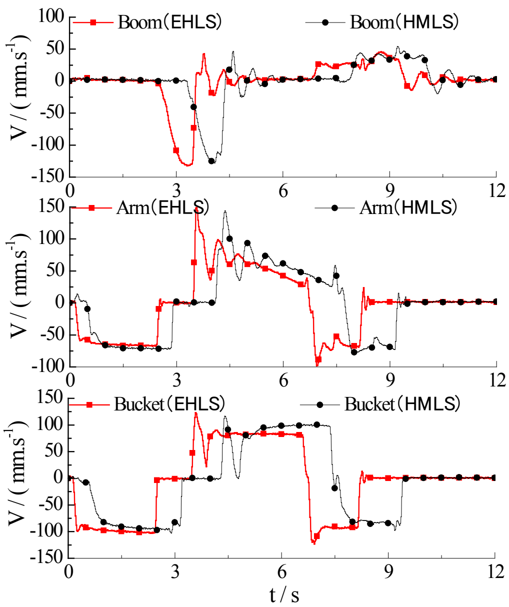
$$\eta = \frac{(\Delta p_{LS1} - \Delta p_{LS2}) \bullet Q}{P_{p1} \bullet Q} \tag{4}$$



(a) Pressure curves: pump pressure, load sensing pressure, and pressure margin that pressure difference between pump and the maximum load pressure



(b) Displacement curves



(c) Velocity curves

Figure 8: Experimental results of excavating test cycle: comparison between the HMLS system and EHLS system

Figure 8 (b)~(c) show the displacements and velocities of actuators. The response of displacements and velocities in the HMLS system are delayed compared with the EHLS system. Therefore, the work efficiency of the experiment cycle is increased by about 10 % in the EHLS system.

4. EHLS based on pressure and flow compound control

4.1. Problems of overflow and pressure impact

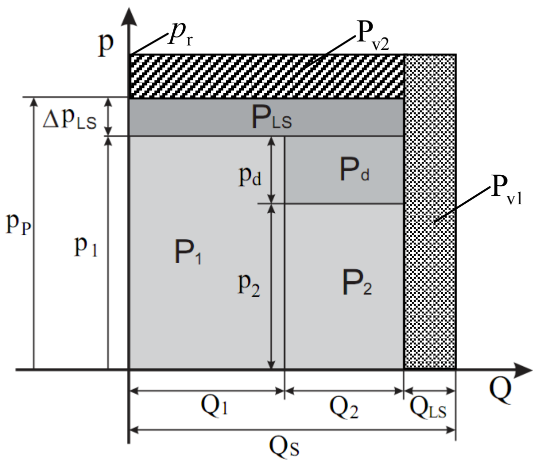


Figure 9: The characteristics of energy consumption

In the EHLS system, if the flow (Q in equation (2)) for actuators supplied by pump is more than the sum of requested flow (Q_s in equation (2)) of actuators, overhead flow matching occurs. Then, the system pressure increases and overflow occurs. **Figure 9** shows the characteristics of energy consumption when the flow is overhead in the EHLS system.

The power consumption due to overflow through the relief vale is:

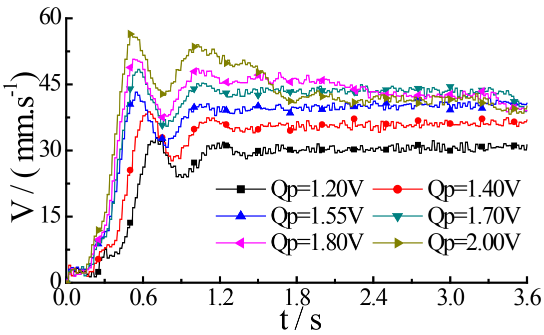
$$P_{\mathrm{v1}} = p_{\mathrm{r}} \cdot (Q_{\mathrm{s}} - Q_1 - Q_2) \qquad (5)$$

The power consumption due to pressure loss in pressure compensator is:

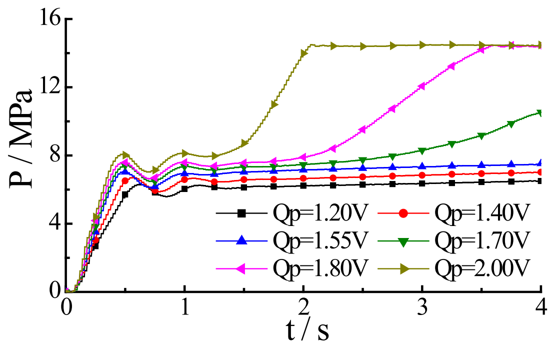
$$P_{\mathrm{v2}} = (p_{\mathrm{r}} - p_{\mathrm{p}}) \cdot Q_{\mathrm{s}} \qquad (6)$$

The sum of power consumption due to overhead flow matching is:

$$P_v = P_{v1} + P_{v2} \qquad (7)$$



(a) Velocity curves



(b) System pressure curves

Figure 10: Experimental results of overhead flow matching

The experimental results with different relations of flow matching are shown in **Figure 10**. When $Q_{\mathrm{p}} > 1.55$, overhead flow matching occurs. With overhead flow of a pump increasing, the overshoot and oscillations increases together. The stability of the EHLS system decreases. Moreover, the system pressure reaches to the set pressure of the relief valve. Therefore, the energy consumption and heat of the system increase greatly. When $Q_{\mathrm{p}} < 1.55$, flow saturation occurs, the required velocity isn't met. On the other hand, pressure impact will happen when system actuators instantaneously stop or quickly switch, as shown in **Figure 8 at** the time of about 3 s and 7 s.

4.2. Principle of pressure and flow compound control

A new method of pressure and flow compound control is proposed to solve the problems of overflow, energy loss and the pressure impact. By the control method, the EFMC system combined the advantages of open-loop flow control in general EFMC systems with the advantages of closed-loop pressure control in electro hydraulic LS systems. Therefore, the closed-loop pressure control and open-loop flow control can be switched under different working conditions. When flow was excessive, the system used closed-loop pressure control, otherwise open-loop flow control was adopted.

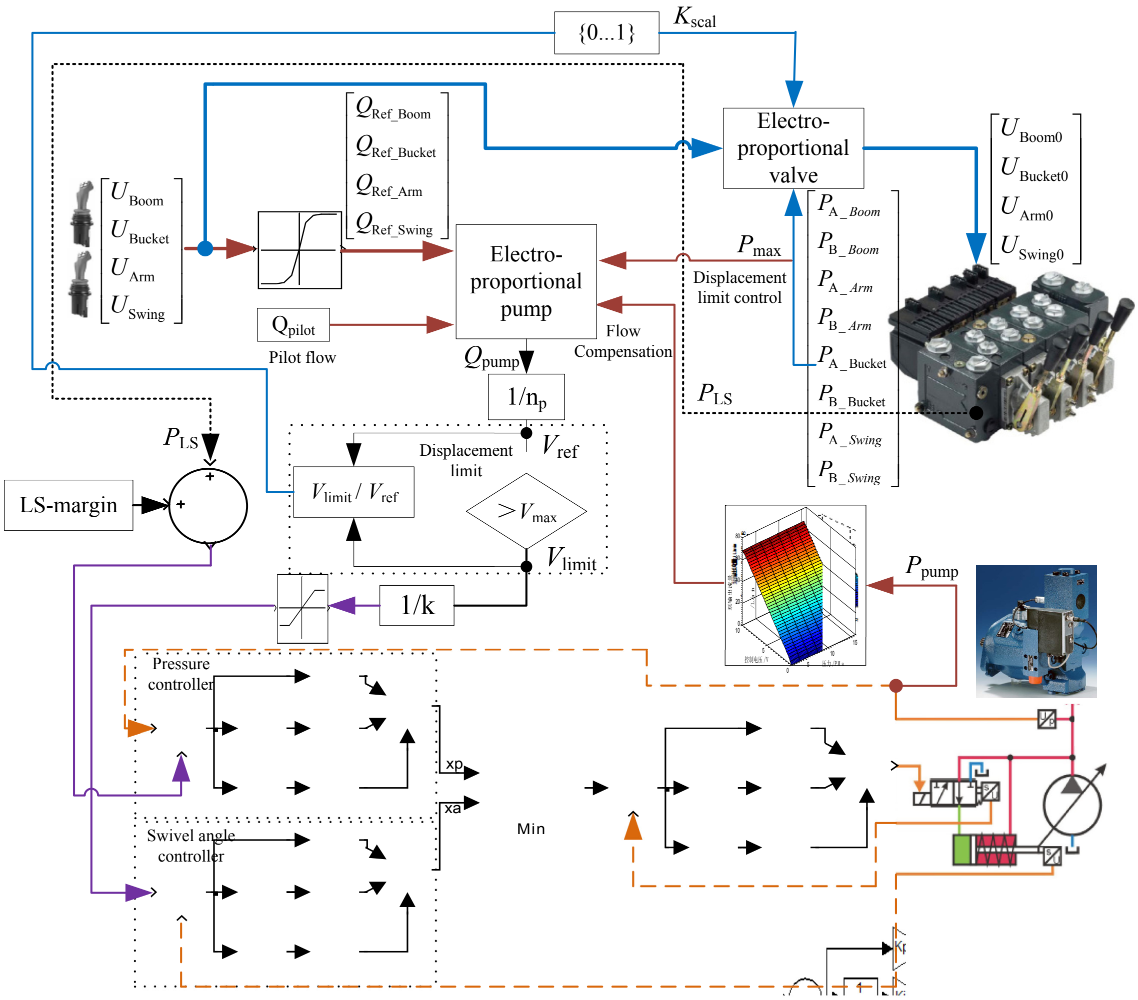


Figure 11: Diagram of pressure and flow compound control

The electro-proportional pump controls pressure and flow on the basis of electro hydraulic closed-loop control. The actual value for pressure is picked up by a pressure transducer. A displacement transducer picks up the actual pump swivel angle. The amplifier card compares these actual values with the command inputs. A minimum value generator assures the activation of the appropriate controller. The output signal of this generator is used as the input signal to the proportional valve solenoid on the pump. The function of flow control is used in the EHLS systems normally. However, if the flow provided by the pump is overhead or a consumer is blocked, the pressure of the consumer will increase. To solve these problems, the function of pressure control is working with the pressure command by the sum of load sensing pressure feedback and certain pressure margin. As you can see in **Figure 11**, the module of flow compensation based on system pressure, the module of anti-saturation and the module of end position detection are added in the system controller.

4.3. Experimental results

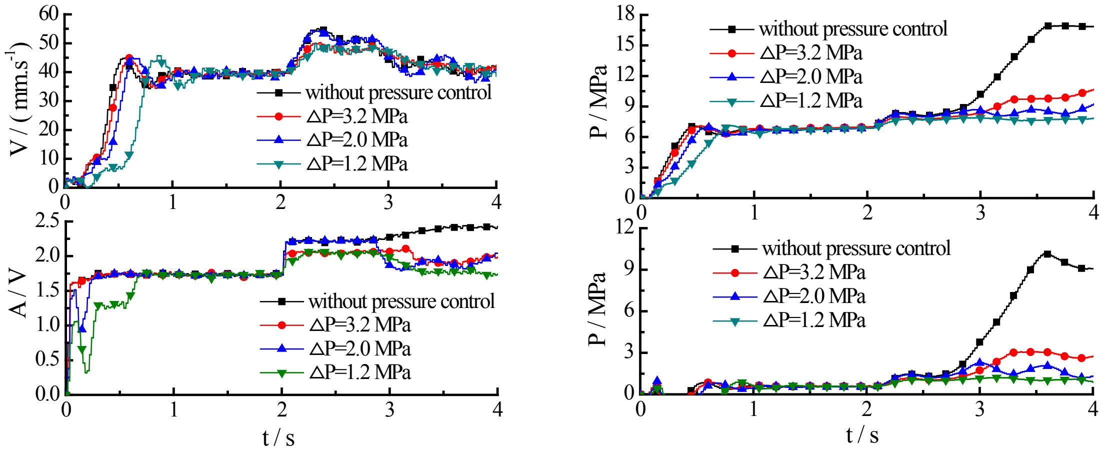
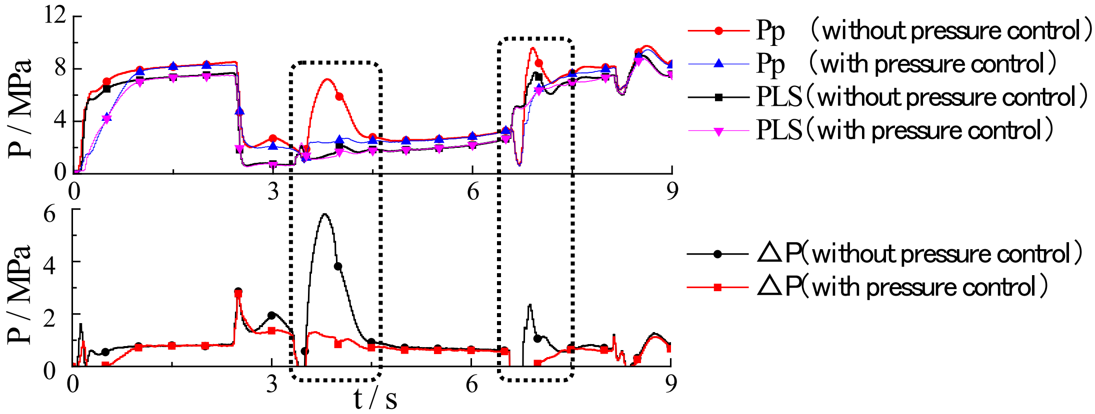


Figure 12: Experimental results of boom up test cycle: comparison between the EHLS system with pressure control and that without pressure control ((a) velocity curves (b) swivel angle curves (c) pump pressure curves (d) pressure margin curves)

The experimental comparison between the EHLS system with pressure control and that without pressure control is shown in **Figure 12**. In the EHLS system with pressure control, the referenced pressure margin is set to be 1.2 MPa, 2.0 MPa and 3.2 MPa. When overhead flow matching occurs after the time of 2 s, compared with the EHLS system without pressure control, the system pressure and pressure margin of the EHLS system with pressure control is reduced greatly and the requested flow is met by closed-loop pressure control. It is similar to traditional hydraulic-mechanism load sensing system, but the mechanism feedback for pressure signal is replace by the electric feedback.



Pressure curves: pump pressure, load sensing pressure, and pressure margin that pressure difference between pump and the maximum load pressure

Figure 13: Experimental results of excavating test cycle: comparison between the EHLS system with pressure control and that without pressure control

Figure 13 shows experimental results of excavating test cycle regarding pressure characteristics. When the control valve quickly switches, the pump flow is overhead owing to the response difference between the pump and valve. Due to the function of pressure control, the pressure impact is effectively inhibited when system actuators quickly switch.

5. Conclusions

The principle and characteristics of EHLS systems were analyzed in the 2-ton class crawler type excavator prototype which installed an EHLS system and a HMLS system. Energy consumption and dynamic characteristics were comparatively discussed between the HMLS system and the EHLS system by simulation and experiment. To solve the problems of overhead flow matching and pressure impact, the method of pressure and flow compound control was proposed, which was validated to be effective by experimental investigation.

6. Acknowledgement

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7. References

/1/ Zahe B. Energiesparende Schaltungen hydraulischer Antriebe mit veränderlichem Versorgungsdruck and ihre Regelung[D]. Aachen: RWTH Aachen University, 1993.

/2/ DUROVIC M, HELDUSER S. New control strategies for electrohydraulic load sensing[C]. Proceedings of the Bath Workshop on Power Transmission and Motion Control. BATH, UK: PTMC, 2004: 201-210.

/3/ HELDUSER S, DJUROVIC M. Control strategies for load-sensing in mobile machinery[C]. Proceedings of the 6th International Conference on Fluid Power Transmission and Control. Hangzhou,China: ICFP, 2005:32-42.

/4/ Finzel R. Elektrohydraulische Steuerungssysteme für mobile Arbeits-maschinen[D]. Technische Universität Dresden, 2010.

/5/ Finzel R, Helduser S, Jang D S. Electro-Hydraulic Dual-Circuit System to Improve the Energy Efficiency of Mobile Machines[C]. The 7th International Fluid Power Conference, Aachen, Germany, 2010.

/6/ Grösbrink B, Harms H H. Control concept for an advanced load-sensing system[C]. The Seventh International Conference on Fluid Power Transmission and Control, Hang Zhou,China, 2009:201-204.

/7/ Grösbrink B, Harms H H. Alternating Pump Control for a Load-Sensing System[C]. The 7th International Fluid Power Conference, Aachen, Germany, 2010.

8. Nomenclature

v	velocity	mm/s
$\triangle p$	difference pressure /pressure margin	MPa
ρ	density	Kg/m^3
p_p	pump pressure	MPa
p_LS	maximum load pressure	Mpa
P	power	kW
A_c0	area	mm^2
F_0	preset spring fore	N
W_i	area gradient	mm
X_i	valve opening	mm
Q_s	flow	L/min
V	displacement	ml /rev
n	speed	rpm
η	efficiency	
p_r	set pressure of relief valve	MPa