A Novel Electro-Hydraulic Compound Driving System With Potential Energy Regeneration Capability for Lifting Device

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Abstract

In order to realize the high energy efficiency and high performance motion control of the lifting device of construction machinery, this paper proposes an electro-hydraulic compound driving system that can regenerate the potential energy. In this new system, the high-power density combination of hydraulic cylinder with accumulator is used to balance the self-weight of the lifting device and recycle the gravitational potential energy directly. The green electro-mechanical actuator with high energy efficiency and control performance is used to control the motion of the lifting device. In addition, control valves are added to dynamically adjust the pressure of the two chambers of the hydraulic cylinder and increase the stiffness of the hydraulic system, which can restrain the impact of acceleration and deceleration of large inertia lifting device and rapid charging and discharging of accumulator on electro-mechanical actuator. Finally, a typical construction machinery excavator is structurally modified to build a test platform. The test results show that, under the same working conditions, the proposed electro-hydraulic compound driving system can further reduce energy consumption by 31% and reduce peak power by 15% than that of the high energy efficiency three-chamber hydraulic cylinder driving system with potential energy regeneration. The boom runs smoothly, and the velocity overshoot and fluctuation time are significantly reduced, realizing high performance driving of the lifting device with low energy consumption. The research results also have broad application prospects in missile erection devices, cranes, presses, and other equipment.

Index Terms

Lifting device, electro-hydraulic compound driving, potential energy regeneration, high energy efficiency, excavator boom, high performance.

I. INTRODUCTION

With the rapid development of industry, fossil energy shortage and carbon emission has become serious problems, which lead to the continuous rise of global temperature. The interest in implementing energy efficiency actions has been adopted by many countries [1], [2]. For construction machinery, as the major user of fuel consumption and carbon pollution emission, energy conservation and emission reduction are urgent.

At present, the lifting device of construction machinery is generally driven by valve controlled hydraulic cylinder system [3], [4]. In order to ensure the operation quality, the opening of the control valve is small and there is a lot of throttling loss, causing a lot of energy waste [5], [6]. In addition, during the operation, the heavy lifting device is frequently lifted and lowered. In the process of lowering, the large amount of gravitational potential energy is converted into heat energy through the throttling of the control valve, resulting in huge energy waste and pollution emission [7], [8].

Among the existing methods for recovery and utilization of gravitational potential energy, electrical recovery ways have more energy transmission and conversion links, in which the potential energy is firstly converted into hydraulic energy to drive the hydraulic motor, then the hydraulic motor drives
the electric generator to convert the hydraulic energy into electrical energy [9], [10].

For hydraulic recovery ways, a balance hydraulic cylinder is added in parallel in the driving circuit of the hydraulic cylinder of the lifting device. The rodless chamber of the balance hydraulic cylinder is connected with the accumulator to balance the weight of the lifting device and realize the recovery and utilization of the gravity potential energy [11]. The combination of hydraulic cylinder with accumulator has the advantages of less energy conversion times, short transmission chain and less energy conversion loss [12]. Liebherr of Germany [13] has applied for the invention patent of hydraulic excavator using this technology, and the products have been manufactured and put on the market. Quan applied this method to the large hydraulic excavator boom, and the system energy consumption can be reduced by 52% [11]. Then, Quan further proposed a three-chamber hydraulic cylinder driving system, which integrates the driving hydraulic cylinder and balance hydraulic cylinder [14], [15]. With the three-chamber hydraulic cylinder driving system, the energy consumption of the excavator boom is reduced by 50.1% and the peak power is reduced by 64.9%. Although this hydraulic recovery energy system realizes efficient direct conversion and utilization of gravity potential energy, the drive circuit of lifting device still adopts valve control system, which still exist throttling loss. Hao et al. [16] and Zhou et al. [17] replaced the valve control drive circuit with the pump control drive circuit, further improving the energy efficiency of the system.

Electro-mechanical actuator is a kind of green linear actuator with high control accuracy and energy efficiency, widely used by aerospace, precision machining, robots, and simulation platforms [18]–[20]. Compared with the hydraulic cylinder drive, the electro-mechanical actuator does not need to control the load movement through the throttling effect of the valve, and there is no throttling loss [21]. In addition, the electro-mechanical actuator is a rigid mechanical transmission, avoiding the influence of nonlinear factors such as pressure and leakage of hydraulic system on the control characteristics. At present, electro-mechanical actuator has been successfully applied to drive the landing gear and hatch of aircraft [22], [23]. Jang design a high force electro-mechanical actuator for computer numerical control lathe machines, which can replace the conventionally used hydraulically powered headstocks of the lathe machine [24]. Liu studied the vehicle mounted large missile launcher driven by electromechanical actuator, and realized the rapid vertical ejection of large missile driven by electricity [25]. Elduque reduced energy consumption by 15% through optimizing the process of electric injection molding machine [26]. However, the power density of electro-mechanical actuator is small and the bearing capacity is weak. It is mainly used for driving the light mechanisms, and difficult to directly drive the heavy device.

Starting from the purpose of efficient recycling gravitational potential energy of lifting device and efficient driving control of low-carbon construction machinery, in this paper, a novel electro-hydraulic compound driving system is proposed, based on the advantage fusion of electro-mechanical actuator and hydraulic system. The combination of hydraulic cylinder with accumulator is used to balance the self-weight of the lifting device and recycle the gravitational potential energy directly, making full use of the advantages of high-power density and efficient direct conversion of gravitational potential energy and hydraulic energy. Electro-mechanical actuator is used to control the motion of the lifting device, making full use of the advantages of high energy efficiency and control performance, realizing high performance driving of lifting device with low energy dissipation.

The rest of this paper is organized as follows. In section II, the electro-hydraulic compound driving system principle is proposed and analyzed. In section III, the mathematical modeling and theoretical analysis of the electro-hydraulic compound driving system are presented. In section IV, the hydraulic pressure regulation control strategy is designed to restrain the impact of heavy lifting device and rapid discharging of high-pressure accumulator on electro-mechanical actuator. In section V, taking the typical construction machinery excavator as the application object, relevant tests are performed and the analysis of system characteristics are expounded. The conclusion is given in section VI.

II. SYSTEM WORKING PRINCIPLE

An electro-hydraulic compound driving system for lifting device with potential energy regeneration capability is shown in Fig. 1.

As can be seen from Fig. 1, the heavy lifting device is jointly driven by an electro-mechanical actuator and two hydraulic cylinders arranged in parallel. The electro-mechanical actuator with high energy efficiency and control
performance is used to control the running velocity and position of the lifting device. There is no throttling loss in the motion control process. Under the active motion control of the electro-mechanical actuator, the hydraulic cylinders make the passive following motion. The rodless chamber of the hydraulic cylinder is connected with the hydraulic accumulator through the control valve 1 to balance the weight of the lifting device and realize the recovery and utilization of the gravity potential energy. Specifically, when the heavy lifting device lowering, its gravitational potential energy is converted into hydraulic energy through the hydraulic cylinder and stored in the hydraulic accumulator. When the heavy lifting device is lifted, the hydraulic energy stored in the accumulator is directly converted into the dynamic potential energy of the lifting device, assisting the electro-mechanical actuator to drive the heavy lifting device. The whole potential energy regeneration process only goes through one energy conversion, and the energy conversion link is the least, which can reduce energy loss. The power source of the whole system is the clean and non-pollution electric energy.

Further, under the control signals $u_1$ and $u_2$, the control valve 1 and control valve 2 are used to regulate the dynamic pressure difference between the hydraulic cylinder and the accumulator, oil tank or pressure oil source by changing the opening of the valve port. In this way, the hydraulic cylinder chambers pressure is adjusted to increase the stiffness of the hydraulic system, reducing the impact of acceleration and deceleration of large inertia lifting device and rapid charging and discharging of hydraulic accumulator on electro-mechanical actuator, and prolonging the service life of the electro-mechanical actuator.

### III. MATHEMATICAL MODELING AND ANALYSIS

According to the analysis of the system working principle, the displacement equation of the lifting device is obtained as follows:

$$x = \frac{\alpha l}{2\pi i}$$  \hspace{1cm} (1)

The velocity equation of the lifting device is obtained as follows:

$$v = \frac{dx}{dt} = \frac{l}{2\pi i} \frac{d\alpha}{dt}$$  \hspace{1cm} (2)

where, $x$ is the displacement of the lifting device; $\alpha$ is the rotation angle of the servo motor of the electro-mechanical actuator; $l$ is the lead of the ball screw pair of the electro-mechanical actuator; $i$ is the transmission ratio of the reducer of the electro-mechanical actuator; $v$ is the velocity of the lifting device.

According to Newton’s second law and the mechanical balance relationship, the dynamic balance equation of the lifting device is as follows:

$$N_1F_e + N_2F_h = \frac{m d^2x}{dt^2} + B \frac{dx}{dt} + Kx + mg + f$$  \hspace{1cm} (3)

where, $N_1$ is the number of electro-mechanical actuators; $N_2$ is the number of hydraulic cylinders; $F_e$ is the output force of one electro-mechanical actuator; $F_h$ is the output force of one hydraulic cylinder; $m$ is the equivalent mass of the lifting device; $B$ is the system viscous damping coefficient; $K$ is the system stiffness; $f$ is the interference force including friction, etc.

The relationship between the output force of electro-mechanical actuator and the servo motor torque $T$ of the power source is as follows:

$$F_e = \frac{2\pi Ti}{T}$$  \hspace{1cm} (4)

The relationship between the output force of hydraulic cylinder and the hydraulic system pressure is as follows:

$$F_h = p_1A_1 - p_2A_2$$  \hspace{1cm} (5)

where, $p_1$ is the rodless chamber pressure of the hydraulic cylinder; $p_2$ is the rod chamber pressure of the hydraulic cylinder; $A_1$ is the effective working area of the rodless chamber of the hydraulic cylinder; $A_2$ is the effective working area of the rod chamber of the hydraulic cylinder.

According to equations (3) and (5), the output force of electro-mechanical actuator is also as follows:

$$F_e = \left[ \frac{m d^2x}{dt^2} + B \frac{dx}{dt} + Kx + mg + f + [mg - N_2 \times (p_1A_1 - p_2A_2)] \right] / N_1$$  \hspace{1cm} (6)

According to the “(6)”, in the proposed electro-hydraulic compound driving system, the hydraulic cylinder can output a large force to balance most of the gravity of the lifting device by adjusting the hydraulic system pressure $p_1$ and $p_2$. Whereas the electro-mechanical actuator with disadvantages of low bearing capacity and power density only needs to overcome the residual equivalent gravity of the lifting device, inertial force, system damping force, and friction force, which are very small compared with the original gravity of the lifting device. For example, when the acceleration of the lifting device is 1000 mm/s², its inertial force is only 10% of the gravity. Therefore, the electro-mechanical actuator only needs to output small force to realize high efficiency driving of the heavy lifting device motion.

### IV. HYDRAULIC CONTROL STRATEGY

In the actual working conditions, the large inertia lifting device starts and brakes frequently. When the motion state of the lifting device suddenly changes, its inertia force will directly act on the rigid electro-mechanical actuator, causing a certain impact, which affects the safety, reliability, and service life. Therefore, firstly, the hydraulic system is used to assist the electro-mechanical actuator to balance part of the gravity and inertia force, so as to reduce the peak force on the electro-mechanical actuator. Secondly, according to equation (6), reducing the impact force on
the electro-mechanical actuator is essentially to make the variable load force and inertial disturbance force overcome by the output force of the hydraulic cylinder as much as possible. Therefore, the control idea is to actively adjust the pressure of the hydraulic cylinder chambers or increase the stiffness of the hydraulic system by regulate control valves.

In addition, when the working device is suddenly lifted, the hydraulic accumulator would release high-pressure oil quickly, causing hydraulic impact and vibration to the electro-mechanical actuator and the whole system. Therefore, it is necessary to control the pressure oil discharge velocity of the hydraulic accumulator.

A hydraulic control strategy is shown in Fig. 2, which is characterized by taking the flow matching of hydraulic cylinder as the basis and the pressure regulation of two chambers of hydraulic cylinder as the means. The proposed hydraulic control strategy adopts the self-feedback torque signal of the servo motor to indirectly reflect the force state of the electro-mechanical actuator.

According to Fig. 2, the flow matching controller calculates and outputs the flow matching control signal \( u_q \) based on the system pressure and the operating velocity of the lifting device. Based on the difference between the setting torque \( T_e \) and the real-time actual torque signal \( T_i \) of the servo motor, the pressure regulation signal \( u_T \) is output by the PID controller and superimposed on the flow matching control signal. While ensuring the flow matching, the pressure of the hydraulic cylinder is regulated to meet the bearing requirements of the electro-mechanical actuator and avoid rigid impact. Therefore, the control signal of the control valve includes two parts: flow matching control signal \( u_q \) and pressure regulation signal \( u_T \):

\[
u = u_q + u_T \quad (7)
\]

Ignoring the hydraulic cylinder leakage, the flow equation of the control valve can be given as:

\[
q = C_d W u_q \sqrt{\frac{2 |\Delta p|}{\rho}} \quad (8)
\]

where, \( q \) is the flow of the control valve; \( C_d \) is the flow coefficient; \( W \) is the area gain of the control valve port; \( u_q \) is the flow matching control signal of the control valve; \( \Delta p \) is the pressure difference between the inlet and outlet of the control valve; \( \rho \) is the hydraulic oil density.

Under the motion control of the electro-mechanical actuator, the rodless chamber flow \( q_1 \) and the rod chamber flow \( q_2 \) of the hydraulic cylinder are respectively as follows:

\[
\begin{align*}
q_1 &= v A_1 \\
q_2 &= v A_2 \\
\end{align*}
\quad (9)
\]

Simultaneous equations (2), (8), and (9), under the motion control of the electro-mechanical actuator, the flow matching control signals \( u_{q1} \) and \( u_{q2} \) of the control valve in the lifting stage of the lifting device can be obtained as follows:

\[
\begin{align*}
u_{q1} &= \frac{60C_d W}{n A_1} \sqrt{\frac{2 (p_a - p_1)}{\rho}} \\
u_{q2} &= \frac{60C_d W}{n A_2} \sqrt{\frac{2 (p_2 - p_a)}{\rho}} \quad (10)
\end{align*}
\]

where, \( n \) is the rotation speed of the servo motor.

For the selected electro-mechanical actuator in the actual equipment, its lead \( l \) and transmission ratio \( i \) are fixed values. For the selected hydraulic cylinder, the effective action areas \( A_1 \) and \( A_2 \) of the two chambers are fixed values. And, for the selected control valve, the flow coefficient \( C_d \), the port area gain \( W \), and the oil density \( \rho \) are all can be considered as fixed values.

Defined: \( K_1 = \frac{60C_d W}{l A_1} \sqrt{\frac{2}{\rho}} \) and \( K_2 = \frac{60C_d W}{l A_2} \sqrt{\frac{2}{\rho}} \).

So, equation (10) can be simplified as follows:

\[
\begin{align*}
u_{q1} &= \frac{K_1}{n} \sqrt{p_a - p_1} \\
u_{q2} &= \frac{K_2}{n} \sqrt{p_2 - p_a} \quad (11)
\end{align*}
\]

Similarly, the flow matching control signals \( u_{q1} \) and \( u_{q2} \) of the control valve in the lowering stage of the lifting device can be obtained as follows:

\[
\begin{align*}
u_{q1} &= -\frac{K_1}{n} \sqrt{p_1 - p_a} \\
u_{q2} &= -\frac{K_2}{n} \sqrt{p_p - p_2} \quad (12)
\end{align*}
\]

where, \( p_a \) is the hydraulic accumulator pressure; \( p_l \) is the oil tank pressure; \( p_p \) is the oil source pressure.

On the basis of flow matching, the pressure regulation signal based on servo motor torque is further added to compensate it to realize the pressure regulation of hydraulic system. The PID control signals \( u_{r1} \) and \( u_{r2} \) are respectively:

\[
\begin{align*}
u_{r1} &= G_{p1} \left[ T_e - T_{r1} + \frac{1}{T_{i1}} \int |T_e - T_{r1}| dT + T_{d1} \frac{dT_e - T_{r1}}{dT} \right] \\
u_{r2} &= G_{p2} \left[ T_e - T_{r2} + \frac{1}{T_{i2}} \int |T_e - T_{r2}| dT + T_{d2} \frac{dT_e - T_{r2}}{dT} \right] \quad (13)
\end{align*}
\]

where, \( T_e \) is the setting torque threshold of the servo motor; \( T_i \) is the real-time actual torque signal of the servo motor; \( G_{p1} \), \( G_{p2} \),
$T_{i1}$, and $T_{d1}$ are the proportional, integral, and differential coefficients in PID closed-loop control of control valve 1, respectively; $G_{p2}$, $T_{i2}$, and $T_{d2}$ are the proportional, integral, and differential coefficients in PID closed-loop control of control valve 2, respectively.

In particular, when the working device is in a stable operation state with acceleration of 0, the control valves ports are fully open to minimize the throttling loss of the system.

V. TEST AND CHARACTERISTIC ANALYSIS

In order to verify and evaluate the actual performance of the proposed electro-hydraulic compound driving system, this paper takes the excavator boom as the lifting device research object. Appropriate structural modification of the laboratory 6-ton hydraulic excavator is carried out, and the electro-hydraulic compound driving excavator boom test prototype is constructed as shown in Fig. 3. The test system principle is shown in Fig. 4.

As shown in Fig.3, the driving mode of the excavator boom is changed from the hydraulic cylinder driving to electro-mechanical actuator and two hydraulic cylinders compound driving. A hydraulic accumulator with a nominal volume of 20 L is added to balance the self-weight of the boom and recycle the gravitational potential energy. The control valves and hydraulic pipeline are added, dynamically regulating the pressure of the hydraulic cylinder. During the test, WT3000 power meter is used to measure the power of servo motor of electro-mechanical actuator. Pressure sensors are installed to detect the pressure inside the hydraulic cylinder and accumulator. The system control concepts and test data collecting are being realized by the DSPACE hardware in the loop computer control system ds1103.

A. BOOM OPERATING CHARACTERISTICS

The load sensing driving system of the original excavator boom and the proposed electro-hydraulic compound driving system are tested respectively. The velocity and displacement characteristic curves of the excavator boom under two different driving systems are shown in Fig. 5.

As shown in Fig. 5(a), under the hydraulic cylinder load sensing driving system, due to the large inertia of the boom and the influence of nonlinear factors such as hydraulic oil compression and leakage, boom velocity exists large overshoot and fluctuation at the initial stage of boom lowering. The maximum velocity overshoot is 50 mm/s, and...
it tends to be stable after 1.5 s of large velocity fluctuation. In addition, the boom velocity has overshoot of 14 mm/s, 22 mm/s, and 38 mm/s in the stages of lowering deceleration braking, accelerating lifting, and lifting deceleration braking respectively, resulting in poor operation stability.

As shown in Fig. 5(b), under the electro-hydraulic compound driving system, the maximum overshoot of boom velocity is only 12 mm/s, reduced by 76% than hydraulic cylinder load sensing driving system. After a small velocity fluctuation of 0.5 s, it will stabilize at the target velocity, the fluctuation time is reduced by 66%. Boom operation characteristics are significantly improved than traditional hydraulic cylinder. The reason is that in the electro-hydraulic compound driving system, the motion of the boom is controlled by the electro-mechanical actuator. The rigid mechanical transmission and the servo motor drive with good control characteristics of the electro-mechanical actuator make it have better control characteristics and better boom operation characteristics than the partial flexible drive of the traditional hydraulic cylinder.

B. SYSTEM PRESSURE CHARACTERISTICS
Fig. 6 is the pressure characteristic curves of the electro-hydraulic compound driving system. The pressure difference curve in Fig. 6 is the dynamic pressure difference between the rodless chamber of the hydraulic cylinder and the accumulator.

According to the pressure characteristic curves in Fig. 6 and system principle in II section, when the boom is lowering, the high-pressure oil in the rodless chamber of the hydraulic cylinder flows into the accumulator. The gravitational potential energy of lifting device converts into the hydraulic energy, causing rodless chamber pressure and accumulator pressure gradually increases. During the initial phase of boom acceleration lowering of 0.6 s to 1.5 s, due to the regulation of control valve 1, the maximum dynamic pressure difference between rodless chamber and accumulator is 2 MPa, and the maximum pressure of rodless chamber is 11 MPa, which can balance the gravity of working device and form a certain impedance effect on the large inertia of boom acceleration lowering, reducing the impact on electro-mechanical actuator. After the boom runs stably in 1.6 s, the valve port of control valve 1 is fully opened. At this time, the pressure difference between the rodless chamber and the accumulator is 1.1 MPa, and the throttling loss is the smallest. According to equation (14), in the boom lowering stage, the throttling loss at control valve 1 is only 1.3 kJ.

\[ E = \int \Delta p \cdot vA_1 dt \] (14)

Under the regulation of control valve 1, the accumulator slowly releases high-pressure oil in the boom lifting stage, and the rodless chamber pressure of the hydraulic cylinder is lower than that of the accumulator, so as to avoid the impact of the rapid release of accumulator high-pressure oil on the boom and electro-mechanical actuator. According to equation (14), the throttling loss at control valve 1 is about 1 kJ in the lifting stage.

According to the pump pressure curve in Fig. 6, the oil source pump is always maintained at a low working pressure, which is only used to supply flow to the system. In the whole operation process, the pressure of the hydraulic system has no obvious oscillation and the pressure stability is good.

C. FORCE CHARACTERISTICS OF ACTUATORS
Fig. 7 shows the force characteristic curves of each actuator in the electro-hydraulic compound driving boom system. The force ratio is the ratio of the electro-mechanical actuator force to the total output force of the system, which is calculated as follows.

\[ \varepsilon = \frac{F_e}{F_e + 2(p_1A_1 - p_2A_2)} \] (15)

As shown in Fig. 7, in the boom lowering stage, the average output force of hydraulic cylinder is about 50 kN, which is used to balance the gravity of working device. The output force of electro-mechanical actuator is only about 5 kN, accounting for 9% of the total output force of the system. In particular, at the stage when the boom motion state changes, in order to reduce the impact of the boom inertia.
on the electro-mechanical actuator, the peak output force of the hydraulic cylinder is about 59 kN, which is used to form a certain impedance effect on the boom inertia, by using the hydraulic control strategy proposed in IV section. At the same time, the output force of electro-mechanical actuator is maintained between −10 kN and 17 kN to compensate the hydraulic cylinder output force and control the motion of the boom. The rated load of the electro-mechanical actuator adopted in test system is 40 kN, and the fluctuation range of output force is 25% to 43%, no obvious impact force on the electro-mechanical actuator in the whole process. After three dynamic fluctuations, the output force of the electro-mechanical actuator gradually tends to be stable.

**D. ENERGY EFFICIENCY CHARACTERISTICS**

Fig. 8 shows the power and energy characteristic curves of accumulator under one cycle of boom lowering and lifting.

![FIGURE 8. Power and energy characteristic curves of accumulator.](image)

In the boom lowering stage, the gravity potential energy of working device is directly converted into hydraulic energy and stored in the accumulator. The accumulator recovers energy 13 kJ and the peak power is 5.2 kW. In the boom lifting stage, the accumulator releases energy 13.2 kJ to assist electro-mechanical actuator drives the boom, realizing the reuse of recovered energy.

Fig. 9 shows the power and energy consumption characteristic curves of the proposed electro-hydraulic compound driving system.

![FIGURE 9. Power and energy consumption curves of electro-hydraulic compound driving system.](image)

According to the servo motor power and energy consumption curves in Fig. 9, during the lowering process of the boom, the peak power of the servo motor is 1.84 kW and only 1.5 kJ of electric energy is consumed in the lowering process. The reason is that the hydraulic cylinder outputs a large force to balance the partial weight of the boom, and the electro-mechanical actuator only outputs small power for its motion control. According to the pump power and energy consumption curves in Fig. 9, in the boom lowering stage, the pump peak power is 1.28 kW and the energy consumption is 2.8 kJ for suppling oil to the rod chamber of the hydraulic cylinder. In the process of boom lifting, the peak power of the servo motor is 3.9 kW and the electric energy consumption is 8.34 kJ.

Under the same working conditions of one cycle of boom lowering and lifting, Fig. 10 shows the comparison of energy consumption and power during one cycle of excavator boom lowering and lifting between the original hydraulic cylinder load sensing driving system, the three-chamber hydraulic cylinder driving system which can realize potential energy recycling, and the proposed electro-hydraulic compound driving system.

As shown in Fig. 10, the original load sensing driving system consumes energy 38.5 kJ and the peak power is 13 kW during one cycle of excavator boom lowering and lifting. The load sensing driving system does not recycle the gravitational potential energy of the boom, so the potential energy is wasted. Under the same working conditions, the energy consumption of the three-chamber hydraulic cylinder driving system which can realize potential energy recycling, and the proposed electro-hydraulic compound driving system.

The proposed electro-hydraulic compound driving system can not only realize the efficient recovery and utilization of the gravitational potential energy of the boom, but also adopt...
the electro-mechanical actuator without throttling loss for the driving circuit. The energy consumption of the electro-hydraulic compound driving system is only 12.8 kJ, which is 31% lower than that of the three-chamber hydraulic cylinder driving system, and the peak power of the servo motor is 3.9 kW, reduced by 15% than that of the three-chamber hydraulic cylinder driving system. The energy efficiency of the electro-hydraulic compound driving system has been significantly improved.

VI. CONCLUSION

(1) Aiming at the problems of serious waste of gravitational potential energy of lifting device and large throttling loss of valve-controlled driving system, a novel electro-hydraulic compound driving system with potential energy regeneration capability is proposed in this paper. The high-power density combination of hydraulic cylinder with accumulator is used to balance the self-weight of the lifting device and recycle the gravitational potential energy directly. The electro-mechanical actuator without throttling loss outputs small power to control the movement of the lifting device.

(2) The test results show that the energy consumption of the electro-hydraulic compound driving system is reduced by 67% than that of the traditional load sensing system without gravitational potential energy regeneration capability, and the peak power is reduced by 70%. Compared with the high energy efficient three-chamber hydraulic cylinder driving system which can realize potential energy recovery, the electro-hydraulic compound driving system can further reduce energy consumption by 31%, and reduce peak power by 15%, significantly improving system energy efficiency.

(3) Compared with the hydraulic cylinder driving system, the velocity overshoot of the electro-hydraulic compound driving excavator boom system is reduced by 76%, the velocity fluctuation time is reduced by 66%. In addition, under the proposed hydraulic control strategy, the force characteristics of actuators are relatively stable, and the electro-mechanical actuator has no impact.

(4) Our future work will focus on the impact of system parameter matching on energy efficiency characteristics and its application in other mechanical equipment such as loaders, cranes, and presses.

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