doi:10.3772/j.issn.1006-6748.2018.01.010

Analysis of energy consumption characteristics of hydraulic system for wrecker truck based on CPR①

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Abstract

In order to solve problems associated with high heat production, high-energy consumption and low efficiency for the hydraulic system of the working device in a wrecker truck, a hydraulic energy-saving system based on common pressure rail (CPR) is proposed. A hydraulic transformer is utilized to control the actuators by analyzing energy-consumption characteristics of valve controlled hydraulic system in a wrecker truck. By analyzing the energy-saving principle of a hydraulic energy-saving system, a relevant mathematical model is established. A comparison is performed between the energy-saving hydraulic system and valve controlled hydraulic system in a wrecker truck within the same work period for operating efficiency and energy consumption. Results show that hydraulic energy-saving system of the wrecker truck has better controlling performance and efficiency of about 18% higher than the valve controlled hydraulic system. Energy-saving ratio for total energy consumption in this system reaches 51.46%, demonstrating energy-saving effect of the system.

Key words: common pressure rail (CPR), hydraulic transformer, energy saving, wrecker truck

0 Introduction

The wrecker truck is a type of armored security vehicle with various functions including loading & unloading, battlefield repair, etc. The working device of the wrecker truck is characterized by large inertia and complexity of load variation. Limited by traditional design methods and idea, these vehicles have some problems associated with high heat production, high-energy consumption and low efficiency. With increasing refinement in modern war techniques and diversification of battlefield tasks, a support vehicle which can realize maximal security and combat effectiveness of troops under limited resources is necessary. Therefore, there is an important military significance to carry out research to optimize the hydraulic system of wrecker truck. This is done by utilizing hydraulic energy-saving technology to reduce energy loss of the system, improving work performance, and enhancing support capability constantly and efficiently for each combat mission of troops in the battlefield.

In recent years, great progress has been made on the energy conservation using positive and negative flow technology, load sensitive technology and hybrid fuel-electric technology. Each of these plays a positive role in lowering energy consumption and increasing efficiency. However, some factors, such as larger throttling energy loss and multi-link energy conversion, has limited the energy-saving efficiency[1-4]. With in-depth research into secondary regulation technology for common pressure rail (CPR) in the field of hydraulic energy conservation and availability of a new-type of secondary regulation component, hydraulic transformer, advantages of the hydraulic energy conservation technology based on CPR for low energy consumption, energy recovery and efficiency enhancement becomes more obvious.

Energy saving system of the elevator with applications of CPR and the hydraulic transformer was discussed extensively[5]. Yao, et al.[6] proposed the use of a constant pressure and hydraulic transformer which has been applied to the ZL50 loader in the hydraulic system. Results showed that 55% power was saved in the new installed hydraulic system.

Configuration of the hydraulic hybrid excavator

① Supported by the National Natural Science Foundation of China (No. 51505289) and Scientific Research Projects of Army (No. 2014ZJ16, QT0703).

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Received on Apr, 17, 2017

based on CPR was proposed by Shen, et al.[7], eliminating throttling energy loss and effectively achieving the energy recovery. Wu, et al.[8] proposed a secondary regulation technology based on CPR. This was applied to wheel drives to control large torque with slow speed and constant power was realized to achieve high speed for vehicle.

Successful application of secondary regulation technology based on CPR in engineering machinery provides a new method for research in energy conservation for the hydraulic system of a wrecker truck. In this paper, a new type of hydraulic energy-saving system based on CPR & hydraulic transformer is proposed to improve system efficiency and reduce energy consumption. The work is focused on addressing characteristics of the working device such as energy consumption, efficiency and energy recovery of the hydraulic system in a wrecker truck.

1 Energy consumption analysis of the wrecker truck hydraulic system

1.1 Structure principle

A valve controlled hydraulic system of a working device in the wrecker truck is composed mainly of a hydraulic pump, master reversing valve, hydraulic cylinder, hydraulic motor and a balance valve. The schematic diagram of the system is shown in Fig. 1.

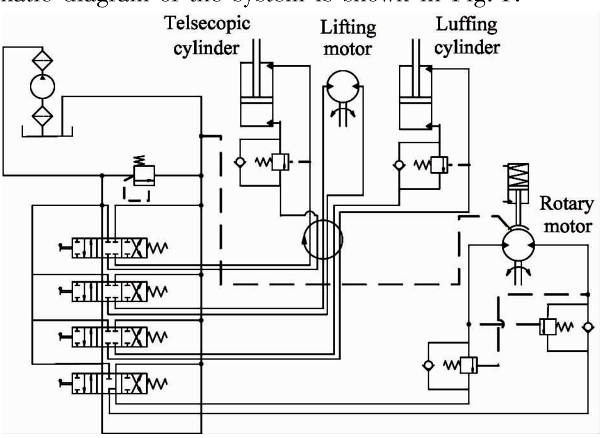


Fig. 1 Schematic diagram of the valve controlled hydraulic system

The wrecker truck provides a necessary system power by power transmission from the engine to the hydraulic pump. Hydraulic oil flows directly to the master-reversing valve. The oil is then provided to the executive components which drive actuators to implement actions of the work. These oil circuits are composed of four three-way and six-port reversing valves, which are independently used for the telescopic mechanism, lifting mechanism, luffing mechanism and rotary mechanism.

1.2 Mathematical models for main hydraulic components

The rotary subsystem and lifting subsystem are driven respectively by the rotary motor and lifting motor. Since their modeling processes are similar, the analysis of the rotary subsystem is taken as an example. Torque T_{sm} and flow Q_{sm} for the rotary motor are as follows[9]:

$$T_{sm} = (P_{sm1} - P_{sm2}) \cdot D_{sm} \cdot \eta_{smm} - J_{sm} \cdot \varepsilon_{sm} - B_{sm} \cdot \omega_{sm} \tag{1}$$

$$Q_{sm} = D_{sm} \cdot \omega_{sm} \cdot \eta_{smv} \tag{2}$$

where, P_{sm1} and P_{sm2} are pressures in two chambers. D_{sm} is displacement, B_{sm} is damping coefficient and η_{smm} is mechanical efficiency. η_{smv} is volume efficiency, J_{sm} is equivalent moment inertia and ω_{sm} is rotating angular velocity.

Analysis of the luffing subsystem is taken as an example for the luffing subsystem and telescopic subsystem. F_{dc} is the output force of the luffing hydraulic cylinder. Flow Q_{dc1} flows into the non-rod chamber and flow Q_{dc2} flows out of the sucker-rod chamber. The output force can be expressed as[9]

$$F_{dc} = P_{dc1} \cdot A_{dc1} - P_{dc2} \cdot A_{dc2} - M_{dc} \cdot \dot{v}_{dc} - B_{dc} \cdot v_{dc} F_{dcf} \tag{3}$$

$$Q_{dc1} = A_{dc1} \cdot v_{dc} + L_{ic} \cdot (P_{dc1} - P_{dc2}) \tag{4}$$

$$Q_{dc2} = A_{dc2} \cdot v_{dc} + L_{ic} \cdot (P_{dc1} - P_{dc2}) \tag{5}$$

where, P_{dc1} and P_{dc2}, A_{dc1} and A_{dc2} are pressures, effective working areas of non-rod and sucker-rod chambers respectively. M_{dc} is the mass of piston, while v_{dc} is the speed of the piston rod and B_{dc} is damping coefficient. F_{dcf} is the frictional force between the piston and cylinder body, and L_{ic} is the internal leakage coefficient.

Flow Q_{pm} for each channel in the multi-way valve is

$$Q_{pm} = C_d \cdot A_{pm} \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}} \tag{6}$$

where, C_d is flow coefficient, A_{pm} is the flow area of the channel, Δp is the differential pressure between both ends of the channel, and ρ is the density of the hydraulic oil.

1.3 System energy consumption analysis

The working condition of the wrecker truck has a periodic characteristic. A standard work flow diagram is set up according to the practical working conditions of the wrecker truck, and the input signal of the system is set in one work cycle under standard workflow, which is shown in Fig. 2. At the same time, simulation is utilized to implement modeling, simulation and analysis of the wrecker truck under no-load condition. In this study, the output energy of the hydraulic pump is considered as total energy consumption of the hydraulic system. Additionally, overflow pressure of the system is set as 11MPa. Fig. 3, Fig. 4 and Fig. 5 show the speed and power characters of the valve-controlled hydraulic system under no-load condition.

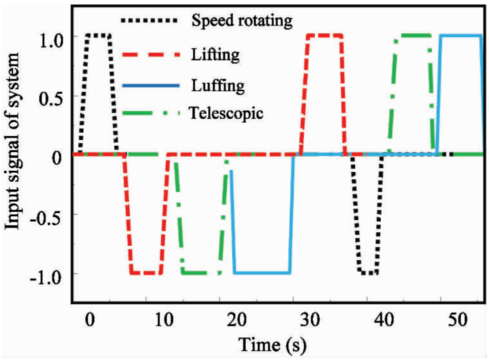
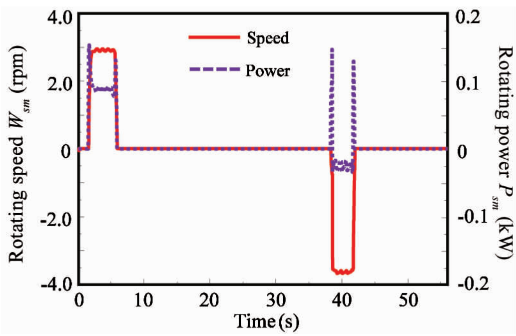
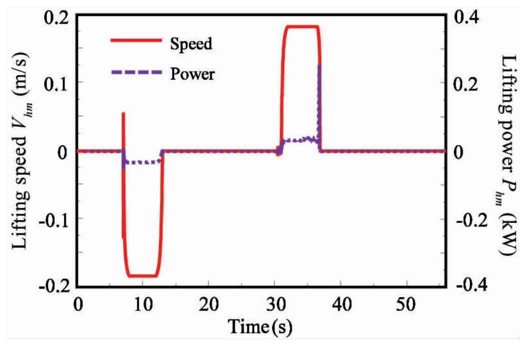


Fig. 2 Control signal in one work cycle

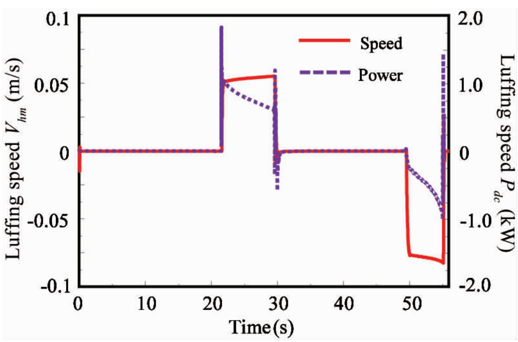
It can be seen from Fig. 3 that the rotation and return speeds are stabilized at 2.89rpm and −3.61rpm. The speed during the returning process is larger, because the driving load is reduced due to the inclination angle of the suspension arm. Speeds for reeling off and reeling in are kept at −0.183m/s. The speed of the lifting arm is increased from 0.051m/s to 0.053m/s. This is because the load increases with the increasing inclination angle of suspension arm. Speed of the drop arm increased gradually from −0.076m/s to −0.082m/s. This increase is related to the increase for acting force of gravity load on the luffing oil cylinder with the decrease in inclination angle of the suspension arm. Speeds of arm extension and arm return are 0.183m/s and −0.272m/s. Speed of the arm return is larger, which is due to a certain inclination angle of the suspension arm during the process of reeling off, the gravities of telescopic arm and lifted object has a positive action on the hydraulic cylinder.



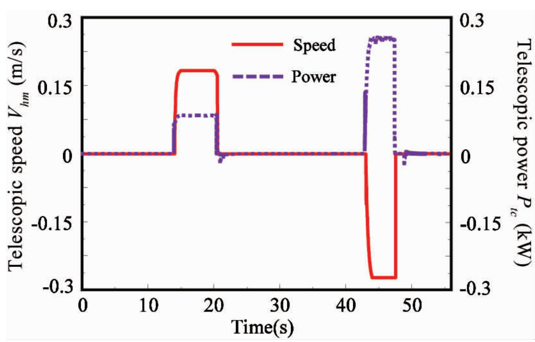
(a) Rotating speed & power curves



(b) Lifting speed & power curves



(c) Luffing speed & power curves



(d) Telescopic speed & power curves

Fig. 3 Speed and power curves of actuator in valve-controlled hydraulic system

The change rule of power is the same as that of speed. However, negative power states are observed for the rotary, lifting and luffing mechanisms, which is the key to energy recovery. This is mainly due to the load force acting in the same direction as actuator motion. However, it is typically converted into heat energy. Power change is large when rotary mechanism performs working and braking. The main reason is that the inertia of the working device is large, pressure of the system is increased when the rotary mechanism starts working and braking.

Power of the lifting mechanism is increased significantly when the lifted object reaches a terminal point. At this point the reeling speed is zero, which is caused by the lifting mechanism ratchet gear, which applies a large torque in the opposite direction. As a result, pressure of both lifting motor chambers increases instantaneously. At the same time, internal discharge flow increases sharply. During the process of lifting arm, power of the luffing mechanism decreases gradually. This is due to the gradual reducing of the load of luffing cylinder. In the telescopic mechanism, power of drop arm is slightly larger than that of the arm extension. The reason is that the speed of the drop arm is larger compared to the extension arm. Moreover, flow of the system during the process of drop arm is larger.

Simulation results in Fig. 4 show that the peak values for output power of the pumps and loss power typically occur in stages when the actuator starts or stops working. Correspondingly, powers decrease when the operating speeds of the actuators become stable. Meanwhile, the loss power of overflow is caused mainly at the time when the actuator starts or stops working and control signal input exists when the actuator reaches stroke limit of displacement.

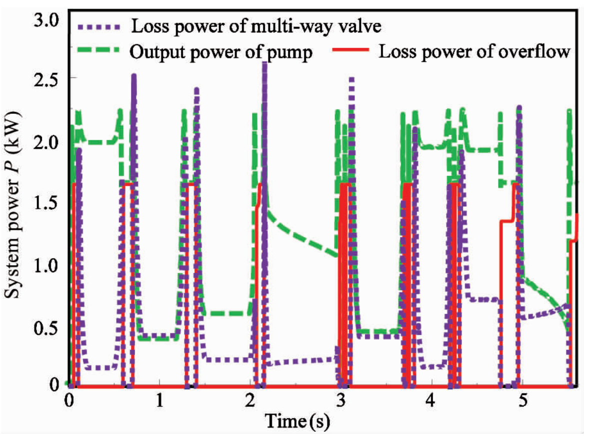


Fig. 4 Main power curves of valve-controlled hydraulic system

Distribution diagram of the energy consumption is shown in Fig. 5. The loss power of the multi-way valve and overflow are 32.99%, and 19.72% respectively. Effective power of the executive component is 17.58%, and loss of other powers such as pipeline is 29.71%. Analysis results show that less than one fifth of the system power is used for the actuator. Half of the system power is occupied by the loss of the multi-way valve and overflow. Therefore, adopting an effective method to lower loss powers of multi-way valve and overflow offers a better energy-saving effect.

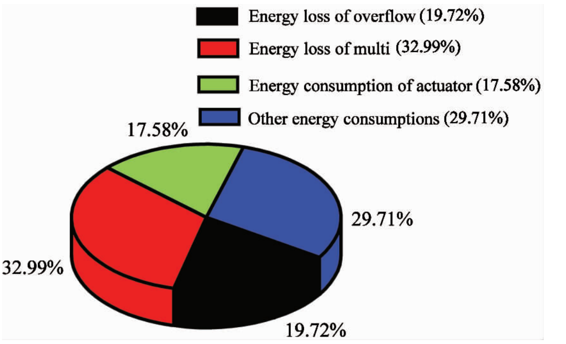


Fig. 5 Distribution diagram of energy consumption in a valve-controlled hydraulic system

2 Energy-saving system analysis of the wrecker truck hydraulic system

2.1 Energy-saving principle

A new-type of wrecker truck hydraulic system is proposed to maximize reduction of power loss for the system, and sufficiently absorb and utilize the potential energy during the drop arm process. This is achieved based on the operating characteristics of the wrecker truck working unit as well as theories and features of CPR and hydraulic transformer, as shown in Fig. 6. The new-type of hydraulic energy-saving system consists mainly of high and low pressure pipelines; executive components, a hydraulic transformer and a high-speed switching valve. Oil is supplied to the high-pressure pipeline by a constant pressure variable pump (master pump) and a hydraulic reservoir. The low-pressure pipeline is connected to the oil tank.

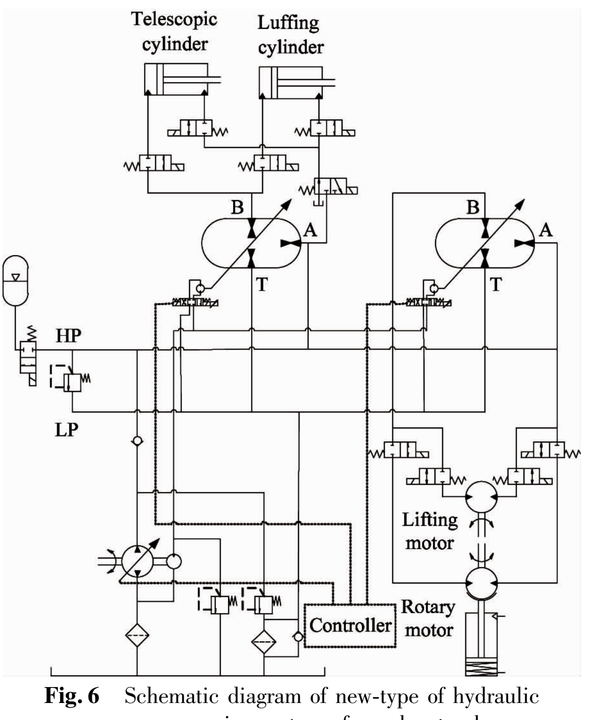


Fig. 6 Schematic diagram of new-type of hydraulic energy-saving system of wrecker truck

When the system is in operation, the output torque and acting force of the motor and the cylinder are matched to the load by adjusting the hydraulic transformer. When the actuator is in braking or declining state, braking energy or gravitational potential energy is recycled and stored in the hydraulic reservoir through the action of hydraulic transformer. At this time, the engine and hydraulic reservoir act as a hybrid power source. This new hydraulic energy-saving system allows power loss of multi-way valve to be eliminated.

2.2 Mathematical models of the new key components

A constant pressure variable pump is used to maintain a quasi-constant system pressure and provide sufficient flow. Discharge pressure p_s and flow q_s of constant pressure variable pump are separately expressed as follows[10,11]:

$$p_s \cdot A_1 = F_k + F_h \tag{7}$$

$$q_s = q_{sv} - q_{fc} - q_i \tag{8}$$

where, A_1 is the constant pressure valve element area and F_k is the constant pressure valve spring force. F_h is the acting force of liquid oil in the constant pressure valve, and q_{sv} is the theoretical flow. Meanwhile, q_{fc} is the flow flowing into the constant pressure valve of constant pressure variable pump and variable hydraulic cylinder, and q_i is flow leakage.

The hydraulic transformer is a secondary regulation component that can be theoretically used to adjust linear and rotational loads in four quadrants without any throttling energy loss. Torques T_A, T_B and T_T of three hydraulic fluid ports A, B and T of hydraulic transformer as well as the mathematical model of transformer ratio λ can be expressed as follows[6,12,13]:

$$T_A = p_A \cdot V_{HT_A} = \frac{p_A \cdot V_{HT}}{2\pi} \cdot \sin\frac{\alpha}{2} \cdot \sin\delta \tag{9}$$

$$T_B = p_B \cdot V_{HT_B} = \frac{p_B \cdot V_{HT}}{2\pi} \cdot \sin\frac{\beta}{2} \cdot \sin\left(\delta - \frac{\alpha}{2} - \frac{\beta}{2}\right) \tag{10}$$

$$T_T = p_T \cdot V_{HT_T} = \frac{p_T \cdot V_{HT}}{2\pi} \cdot \sin\frac{\gamma}{2} \cdot \sin\left(\delta + \frac{\alpha}{2} + \frac{\gamma}{2}\right) \tag{11}$$

$$\lambda = \frac{p_B}{p_A} = \frac{\sin\delta \cdot \sin\frac{\alpha}{2} + \frac{p_T}{p_A} \cdot \sin\frac{\gamma}{2} \cdot \sin\left(\delta + \frac{\alpha}{2} + \frac{\gamma}{2}\right)}{\sin\left(\frac{\alpha+\beta}{2} - \delta\right) \cdot \sin\frac{\beta}{2}} \tag{12}$$

where, V_{HT} is the discharge capacity of the axial-piston component, $V_{HT\text{-}A}$, $V_{HT\text{-}B}$ and $V_{HT\text{-}T}$, α, β and γ, p_A, p_B and p_T are the corresponding discharge capacities, the nominal arc lengths of kidney-shaped slots in the valve plates correspond the pressures of three hydraulic fluid ports A, B and T, respectively. δ is the control angle of the valve plate.

The hydraulic transformer torque that acts on the cylinder body and rotating parts is determined by the sum of turning moments on three ports of hydraulic transformer. Its dynamic characteristic is expressed as follows[12]:

$$\Delta T = J_{HT} \cdot \dot{\omega}_{HT} = T_A + T_B + T_T \tag{13}$$

where, J_{HT} is the inertia of the hydraulic transformer's cylinder body and related components, $\dot{\omega}_{HT}$ is the angular acceleration of hydraulic transformer's cylinder body.

The hydraulic reservoir is mainly used to store energy of the recovery system and forms a hybrid power source[14] with the engine. The thermodynamic equation of hydraulic reservoir is given as

$$p_{a0} \cdot V_{a0}^n = p_a \cdot V_a^n \tag{14}$$

where, p_{a0} is the pressure of the hydraulic reservoir at a stable working point and p_a is the gas pressure inside the hydraulic reservoir. V_{a0} is the gas volume of the hydraulic reservoir at a stable working point, V_a is the gas volume inside the hydraulic reservoir, and n is the gas index, assuming that $n = 1.25$.

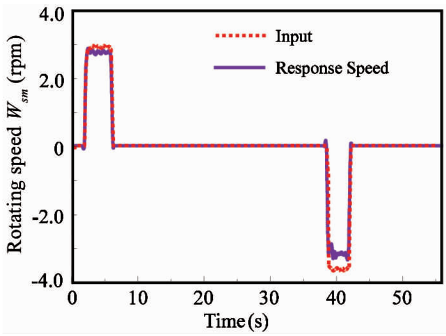
Stored-energy equation of the hydraulic reservoir is set to be positive when doing external work. Otherwise, it is set to be negative. The stored energy is expressed as

$$E = \pm \int_{V_{a0}}^{V_a} p_a \mathrm{d}V_a = \pm \int_{V_{a0}}^{V_a} p_0 \left(\frac{V_a}{V_{a0}}\right)^n \mathrm{d}V_a \tag{15}$$

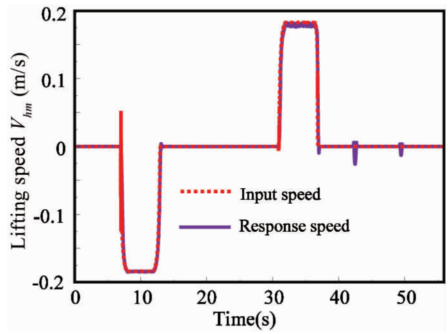
2.3 System performance & energy consumption analysis

To fully validate the feasibility and energy-saving effect of the proposed hydraulic energy-saving system, system performance and energy consumption are analyzed, which is realized by measuring the actuator speed of the valve-controlled hydraulic system as the input signal of hydraulic energy-saving system. Fig. 7, Fig. 8 and Fig. 9 show the speed and power characters of the hydraulic energy-saving system.

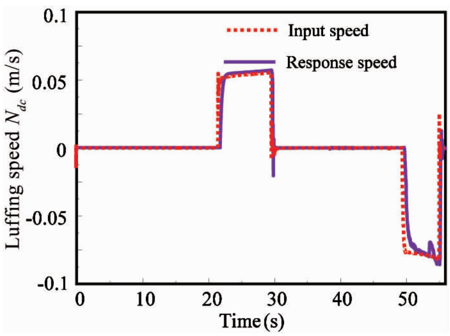
It can be seen from Fig. 7 that it is feasible for the hydraulic energy-saving system to track the input speed signal. Although overshoot and oscillation are observed for the response speed of the actuator compared to input speed, values are within acceptable range. This validates feasibility of the hydraulic energy-saving system.



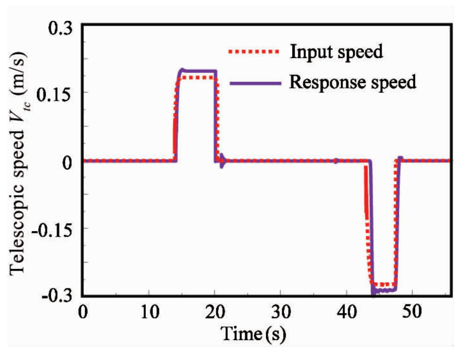
(a) Curve of rotating speed



(b) Curve of lifting speed



(c) Curve of luffing speed



(d) Curve of telescopic speed

Fig. 7 Velocity curve of actuator in the hydraulic energy-saving system

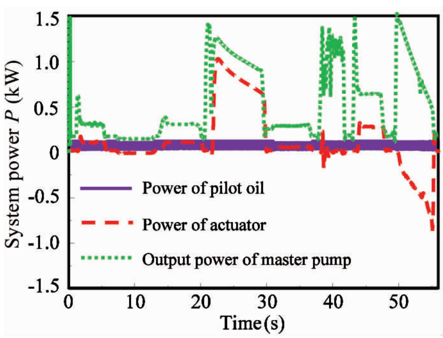


Fig. 8 Main power curve of hydraulic energy-saving system

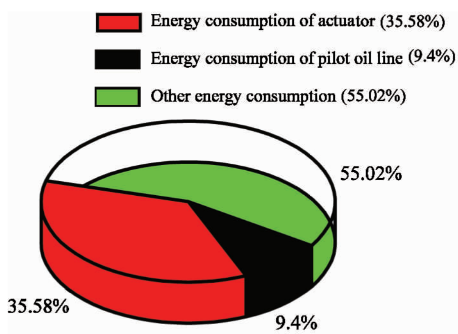


Fig. 9 Distribution diagram of energy consumption in

Fig. 8 shows that the output power of the pump is always below 1.5kW. This is consistent with the power change law of the actuator. The pilot oil line is mainly used to supply oil to the oscillating motor when required energy is lower than 0.1kW.

It can be seen from Fig. 9 that more than one third of the hydraulic energy-saving system energy is used to power the actuator. The energy consumed by the actuator is 35.58%. Meanwhile, energy consumption of the pilot oil line is the lowest at 9.4%. However, other energy consumption like the pipeline is 55.02%.

3 System efficiency & energy-saving performance analysis

3.1 Analysis of system efficiency

In order to study the energy usage of the system comparatively, the operating efficiency η_e of the system defined as the ratio between total energy consumption W_p of the system and energy consumption W_e of the actuator. Operating efficiency η_e is expressed as

$$\eta_e = \frac{W_e}{W_p} \tag{16}$$

Table 1 lists operating efficiency of the hydraulic system. It can be seen that operating efficiency of the valve-controlled hydraulic system is 17.58%, and for hydraulic energy-saving system it is 35.58%. Compared to the valve-controlled hydraulic system, operating efficiency of the hydraulic energy-saving system is improved by 18%. It can be seen that the total energy consumption is significantly reduced, and energy consumption of the actuator in the system loop is increased, so that the efficiency is improved.

Table 1 Operating efficiency of hydraulic system

Hydraulic System	Energy Type	Energy (kJ)	Operating Efficiency
Valve-controlled hydraulic system	Energy consumption of actuator	12.01	17.58%
	Total energy consumption	68.3	
Hydraulic energy-saving system	Energy consumption of actuator	11.79	35.58%
	Total energy consumption	33.15	

3.2 Analysis of the system energy-saving performance

High operating efficiency of the hydraulic energy-saving system means high utilization ratio of system energy. However, since there is difference in energy consumption characteristics between the hydraulic system and hydraulic energy-saving system, energy-saving performance of the system cannot be completely and effectively expressed by operating efficiency. Therefore, energy-saving ratio η_v [14,15] of the system is defined to evaluate energy-saving effect of the hydraulic energy-saving system, and is expresses as

$$\eta_v = \frac{W_f - W_h}{W_f} \tag{17}$$

where, W_f and W_h are the energy consumed by the valve-controlled hydraulic system and the hydraulic energy-saving system respectively.

Table 2 lists energy-saving ratio of the hydraulic system. It can be seen that energy-saving ratio of the total energy consumption in the hydraulic energy-saving system reaches 51.46%, which proves the energy-saving effect. Since the load and work cycles of both systems are consistent, energy consumption of actuators in the hydraulic energy-saving system and valve-controlled hydraulic system are consistent. Energy-saving ratios are relatively small. However, energy-saving ratio of the hydraulic energy-saving system reaches 100% in the energy loss of the multi-way valve and overflow, which suggests that there is no energy consumption by the multi-way valve or overflow in the hydraulic energy-saving system. Pilot control oil line is increased in the hydraulic energy-saving system, which increases energy consumption by the pilot line. In the new hydraulic energy saving system, components of the balance valve etc. are removed and components of the hydraulic transformer are added. Other energy loss due to the pipeline is also changed.

Table 2 Energy-saving ratio of the hydraulic system

Energy type	Valve-controlled hydraulic system	Hydraulic energy-saving system	Energy-saving ratio
Total energy consumption	68.3kJ	33.15kJ	51.46%
Energy consumption of actuator	12.01kJ	11.79kJ	1.83%
Energy loss of multi-way valve	22.53kJ	0kJ	100%
Energy loss of overflow	13.47kJ	0kJ	100%
Energy consumption of pilot oil	0kJ	3.12kJ	--
Other energy consumption	20.29kJ	18.24kJ	10.1%

4 Conclusions

(1) Based on CPR, due to the multi-way valve energy loss is eliminated and the system is simplified in the hydraulic energy-saving system of the wrecker truck.

(2) Hydraulic energy-saving system of the wrecker truck can be used to optimize track input signal of the system, which shows good control performance.

(3) Operating efficiency of the hydraulic energy-saving system in the wrecker truck is improved by 18% compared to the valve-controlled hydraulic system in the wrecker truck.

(4) During one work cycle, energy-saving ratio of the total energy consumption in the hydraulic energy-saving system of wrecker truck reaches 51.46%. Energy-saving effect using the proposed system is significant.

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