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Throttling Loss Energy-Regeneration System Based on Pressure Difference Pump Control for Electric Forklifts

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Abstract: At present, the hydraulic systems of electric forklifts and traditional internal combustion forklifts are mostly valve-controlled speed-regulation systems, which have large throttling losses and potential energy waste. To further improve the energy-saving ability of electric forklifts, the forklift's common working conditions are analyzed in this paper. A throttling loss energy-regeneration system based on pressure difference pump control is designed, and the system's working principle is described. Aiming to deal with the problem that the pump−valve compound speed regulation with constant pressure difference could not realize high controllability and energy saving at the same time, a control strategy for variable pressure difference pump−valve compound speed regulation based on pressure balance control is proposed. The handle signal is positively related to the target speed of the oil cylinder. In the low-speed stage, the closed-loop control of the actual output torque of the motor/generator keeps the pressure difference across the proportional throttle valve unchanged, and the speed adjustment is realized by changing the opening of the proportional throttle valve. In the high-speed stage, the valve opening area is kept unchanged and the target pressure difference is changed to achieve the target speed. Finally, the feasibility of the control strategy is verified through an AMESim simulation, and the minimum pressure difference switching point is determined through experiments. The experiments show that the system's energy-saving efficiency can reach 21.5% under a 1 t load. With the increase in the load, the system's energy-saving efficiency can be further improved.

Keywords: electric forklift; throttling loss; energy regeneration; electric motor; pump/motor; variable pressure control

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1. Introduction

In recent years, global climate change has attracted the attention of all countries. With the transformation and upgrading of some high-emission industries and the increasing efforts to prevent and control motor vehicle pollution, the proportion of emissions from non-road mobile machinery has become more prominent. Non-road mobile machinery is mainly driven by internal combustion engines, which have high emissions, loud noise, and low efficiency. With the full implementation of the National IV emission standards in China, it is urgent to upgrade the technology related to energy conservation and emission reduction. With the continuous maturity of new energy technology for vehicles, hybrid drives and electric drives have gradually become the mainstream driving modes of new energy construction machinery and are providing excellent energy-efficient performances [1].

As the most common handling machinery in the logistics industry, the electrification of forklifts developed earlier, and there are many relatively mature products. However, the hydraulic systems are almost the same as those used in traditional internal combustion forklifts. Traditional forklifts use valve-controlled speed systems and there is a large amount of throttling loss and potential energy waste. The reduction of throttling loss and recovery of potential energy is important to realize energy savings and emission reduction [2].

In view of throttling loss, Prof. Yang used a variable voltage variable frequency (VVVF) hydraulic speed-control system in hydraulic elevators to realize speed control and potential energy recovery in elevators. The energy-saving efficiency was about 40%, but the dynamic performance of the system was slightly weaker [3]. Lin and Fu et al. employed variable speed volume speed regulation in hydraulic energy-saving systems, such as positive flow and load-sensing systems. The research results showed that energy-saving efficiency could reach 35.2% for positive flow systems and 19~70% for LUDV load-sensing systems [4–6]. Xu et al. combined variable speed pump controls and an independent metering system to cooperatively control the cylinder speed, and the system efficiency was increased by 21.3% [7]. This indicates that reducing the throttling loss could improve energy utilization. However, the valve-controlled speed control has high throttling losses and low energy-saving efficiency. Though variable speed pump-controlled speed control has a high energy-saving performance, the response speed and anti-interference performance are poor, and cannot well meet the common working condition requirements of forklifts.

For potential energy recovery, Minav had carried out a variety of researches for forklift, including hydraulic type and electrical type. Most of them adopted variable speed pump-controlled speed regulation, with a recovery efficiency of up to 66% [8–11]. Fu et al. proposed a potential energy-recovery scheme of dual hydraulic motors for heavy forklifts to avoid the low-efficiency zone of motors under low speed and heavy load conditions, and the system recovery efficiency ranged from 11.48% to 80.18% [12]. Wang et al. proposed a pump−valve compound speed-regulation control method based on pressure difference control for the boom potential energy recovery of the excavator. The potential energy was recovered by adding a hydraulic motor and a generator. Compared with traditional throttle speed regulation and variable speed pump-controlled speed regulation, compound speed regulation had a faster dynamic response and better controllability [13]. Xia applied a three-chamber hydraulic cylinder in the excavator, and the research showed that the energy-saving efficiency reached 51.1% [14]. Ahn et al. proposed a novel hydraulic hybrid forklift for energy saving stored in the accumulator. The results showed that the maximum energy-saving efficiency could reach 56% in one cycle and the total energy-saving efficiency was 19% [15]. Yu et al. proposed a novel hybrid regeneration system for electric forklifts. The gravitational potential energy and braking energy could be regenerated by the hydraulic accumulator and the battery, respectively. Through simulation analysis, the energy regeneration efficiency was 54% under typical unloading conditions [16]. Zhou et al. proposed a novel hybrid regeneration system with a battery and hydraulic accumulator to recover the potential energy and braking energy for forklifts. The simulation results showed that the energy regeneration efficiency was 54% under typical unloading conditions [17]. Zhao et al. proposed a forklift that uses a ball screw device for lifting instead of the hydraulic cylinder. The simulation results show that fuel consumption can be reduced by about 46.72% compared with the hydraulic forklift when the load is 3 t [18]. Recovering the potential energy can effectively improve the energy-saving ability of the system. However, in hydraulic recovery, a hydraulic accumulator is needed, which is not suitable for small- and medium-sized forklifts due to the limited installation space. Electrical recovery needs to add an additional set of hydraulic motor–generators, which will increase system complexity and cost.

In this paper, a lifting hydraulic system of forklifts is studied. The common working conditions of forklift trucks are analyzed. A variable pressure-difference pump–valve compound speed-regulating system and control strategy is proposed based on the balance of control and energy saving to realize the integration of lifting drive and potential energy recovery. The rest of this paper is organized as follows: In Section 2, the working conditions are analyzed and the lifting system speed-control scheme is designed. In Section 3, the variable pressure-difference pump−valve compound speed-regulation control strategy is developed in detail. In Section 4, the simulation model is set up and the performance characteristics are analyzed through simulation. In Section 5, the controllability, energy-saving ability, and proposed speed control are tested through the test rig. Conclusions are given in Section 6.

2. Throttling Loss Energy-Regeneration Scheme

2.1. Working Conditions Analysis

The usual route of forklift operation is shown in Figure 1. Among the pick-up point A, the stowage point B, the initial position C, and the transfer point D, the forklift is shifted according to lines 1 to 5 to realize cargo handling.

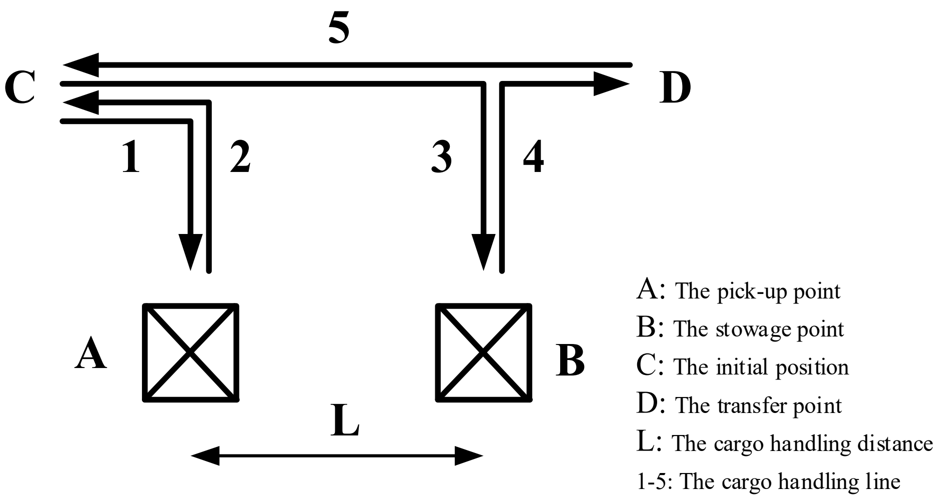


Figure 1. Forklift operation roadmap.

Figure 2 shows the operation flowchart of conventional forklifts, which is mainly composed of five processes: no-load pickup, lift pickup, drop delivery, lift stacking, and drop stacking. The process involves eight major working steps: load lifting/dropping, no-load lifting/dropping, fork leaning forward/backward, walking, and steering.

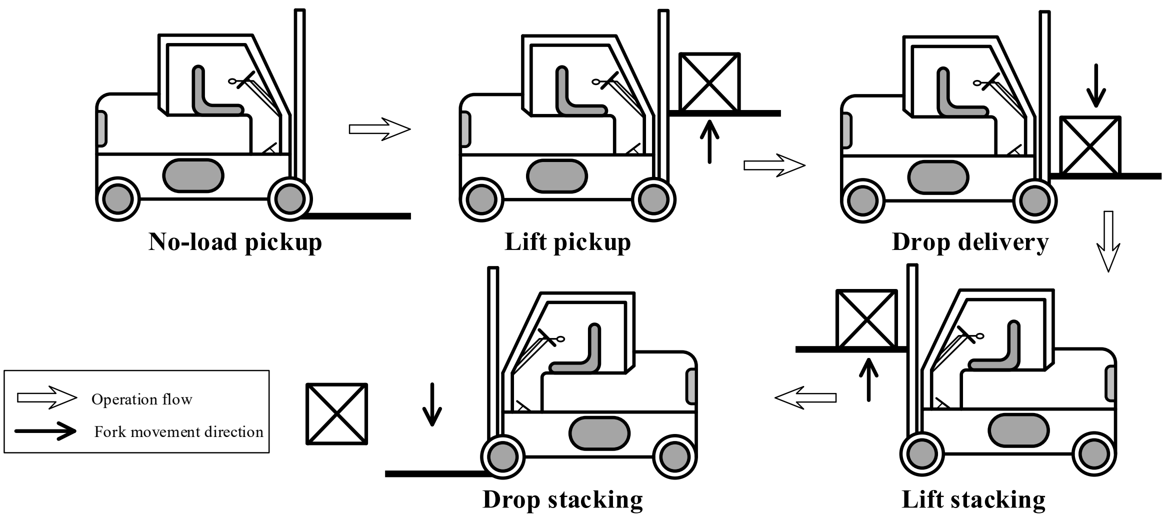


Figure 2. Forklift operation flowchart.

Qian et al. tested a traditional forklift with a 1 t load according to the test standard JB/T 3300-2010. The test data showed that the energy consumption of the walking system, the hydraulic working device (lifting system), and the steering system were 4110 kJ, 3361 kJ, and 907 kJ, accounting for 49.1%, 40.1%, and 10.8%, respectively [19]. Wang adopted the same test standard and achieved a similar result [20]. For traditional electric forklifts, the main energy-consuming mechanisms are lifting and walking, with the lifting mechanism accounting for more than 70% of the energy consumption of the whole hydraulic system.

For traditional valve-controlled electric forklifts, the hydraulic lifting cylinder's speed depends on the valve opening. Figure 3 is a schematic diagram of the energy consumption of the traditional valve-controlled system in the lifting process. The load pressure p_{L} determines the system pressure p_{p}, and the difference between p_{p} and p_{L} is the loss of valve port pressure difference caused by the throttle valve. q_{p} is the total system flow, and q_{L} is the actual flow into the lifting cylinder. P_{L}, P_{v}, and P_{T} are the load consumption power, throttling loss power, and throttling loss power of multiway valve return oil, respectively.

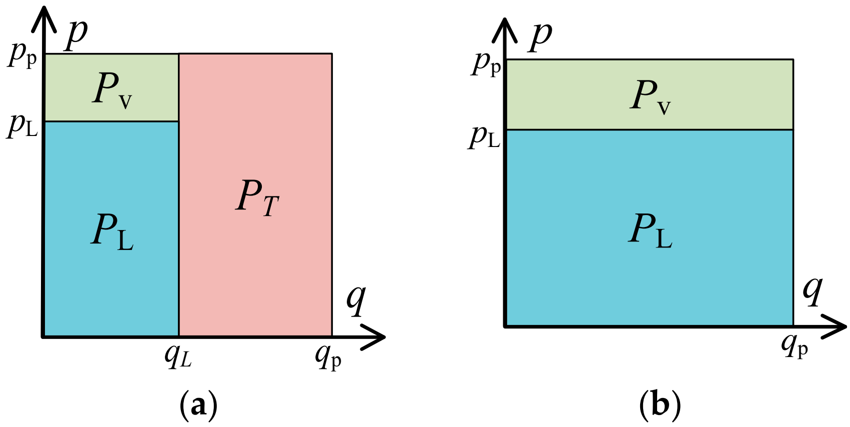


Figure 3. Diagram of energy consumption of traditional valve-controlled lifting. (**a**) Non-full speed; (**b**) full speed.

As can be seen in Figure 3, when lifting at non-full speed, the pump output flow is fixed due to the constant speed-control method adopted by the motor. A small part of the flow enters the lifting cylinder, and most of the flow returns to the oil tank from the middle position of the multiway valve, which leads to large return oil throttling losses. This part of energy is converted into heat, which affects the service life and safety performance of the hydraulic system. When lifting at full speed, although all the oil enters into the lifting cylinder, there are still throttling losses through the throttle valve port.

In addition, in the descending condition, since there is no potential energy recovery device in the traditional hydraulic system, this part's energy is finally absorbed by the hydraulic oil through the throttle valve in the form of heat, and the high-temperature hydraulic oil further affects the safety performance of the hydraulic system.

2.2. Throttling Loss Energy-Regeneration Design

To further improve the energy-saving ability of electric forklifts, a throttling loss energy-regeneration system based on pressure difference pump control is proposed. A schematic diagram is shown in Figure 4. The control performance of the cylinder is ensured by the proportional throttle valve. However, the original valve-controlled hydraulic speed regulation is replaced by the pump−valve compound speed regulation based on pressure difference control. A set of electric motor/generator−hydraulic pump/motor is used for lifting drive and throttling loss energy recovery by controlling the pressure difference of the proportional throttle valve. The recovered throttling-loss energy is converted from the gravitational potential energy.

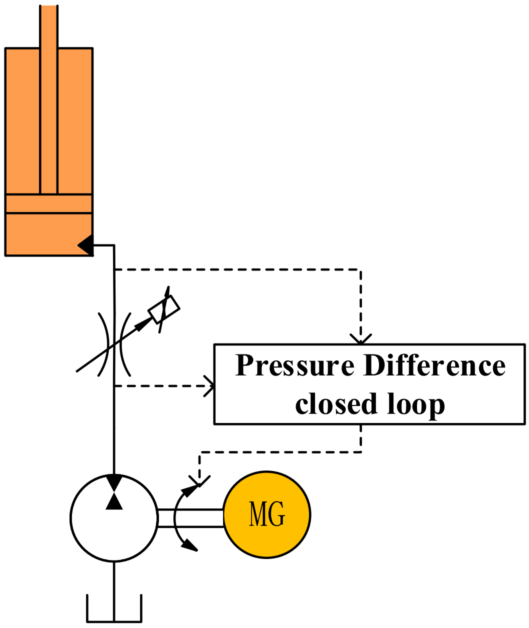


Figure 4. Throttling loss energy-regeneration system based on pressure difference pump control.

When the weight is lifted, the electric motor/generating−hydraulic pump/motor works in the electric motor−hydraulic pump mode. When the weight is lowered, it turns into the generator−hydraulic motor mode, and the gravitational potential energy is converted into throttling loss energy in the proportional throttle valve, which is recovered by the electric motor/generator−hydraulic pump/motor. The recovered energy can release during the lifting process. The proportional throttle valve is retained in the system to improve the system's damping characteristics. The electric motor/generator is used to control the pressure difference of the proportional throttle valve when the actuator is moving. The larger the valve opening and the pressure difference of the proportional throttle valve are, the faster the actuator speed can be achieved. Therefore, this kind of control method can theoretically get better speed-control performance than the traditional valve control, and better response speed than the variable-speed pump control.

The main differences between the new system and traditional valve-controlled speed regulation are that the throttling loss of return oil is eliminated and energy saving is realized. Different from variable speed pump-controlled speed regulation, the proportional throttle valve is retained in the proposed system, which can reduce the pressure fluctuation caused by instantaneous on–off of the oil circuit, ensure the safety of the machine, and improve the controllability of the system.

3. Variable Pressure-Difference Pump−Valve Compound Speed-Regulation Control Strategy

According to the principle of pressure-difference closed-loop pump−valve compound speed regulation, the cylinder speed can be adjusted either through the setting value of the target pressure difference across the throttle valve or the throttle valve opening area. When the target pressure difference setting value is small, the system throttling loss is lower and the energy saving is higher. However, to meet the maximum speed requirements of the system, it is necessary to choose the proportional throttle valve with a larger flow area, which will increase the cost and installation space. While the target pressure difference setting value is large, and the system speed is small, there is a large throttling loss, which leads to the energy-saving reduction.

To solve the above contradictions, considering controllability and energy-saving ability, and combined with the advantages of electric control, a variable pressure-difference pump−valve compound speed-regulation system based on the pressure balance of control and energy saving is proposed, as shown in Figure 5.

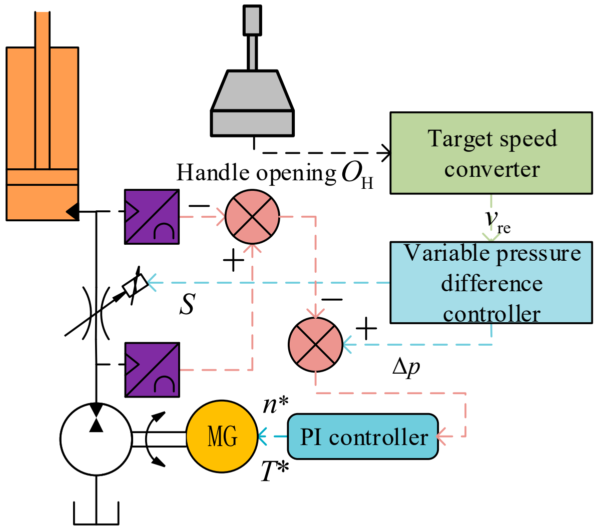


Figure 5. Variable pressure-difference pump−valve compound speed-regulation diagram.

The handle opening is no longer directly related to the proportional throttle valve opening, and the target pressure difference across the throttle valve is no longer fixed. Instead, they are closely related to the current throttle valve opening area and the cylinder target speed. Figure 6 shows the flowchart of the variable pressure-difference pump−valve compound speed-regulation control strategy. After conversion, the handle opening OH directly corresponds to the cylinder target speed v_{re}. The larger the handle opening is, the higher the target speed can be achieved. If the handle is forward, the target speed is positive and the cylinder rises accordingly. While the handle is backward, the target speed is negative and the cylinder drops accordingly.

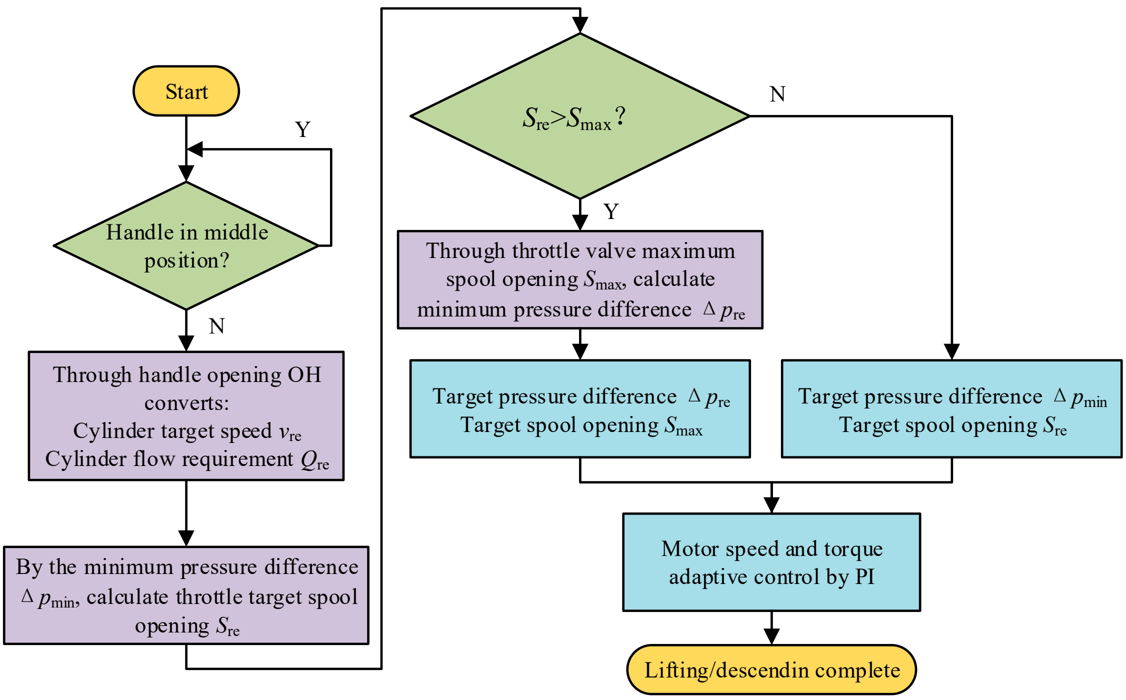


Figure 6. Variable pressure-difference pump−valve compound speed-regulation control strategy flowchart.

When the variable pressure difference controller receives the target speed value v_{re}, it first calculates the demand flow rate to obtain the target flow rate Q_{re}, then takes the minimum pressure difference as the target pressure difference $\Delta p_{\min}$ across the throttle valve, and then calculates the target flow area S_{re} (valve opening u_{re}) of the throttle valve according to the orifice flow formula, which is compared with the maximum flow area $S_{\max}$ (maximum valve opening $u_{\max}$) of the proportional throttle valve. If the target flow area S_{re} of the throttle valve is less than the maximum flow area $S_{\max}$, then it enters the constant pressure difference stage. At this stage, the minimum pressure difference $\Delta p_{\min}$ is output to the PI controller, and the actual flow area of the proportional throttle valve is adjusted to the target flow area S_{re}. Then, the PI controller outputs target speed n^* and target torque T^* of the motor after calculation, controlling the positive and negative rotation of the motor to realize the cylinder lifting or descending. It should be noted that in the simulation, the flow coefficient C_{d} is an idealized and fixed value, but in practice, the flow coefficient C_{d} changes dynamically with the opening and the pressure difference of the proportional throttle valve [21]. In this study, the actual flow area curve of the proportional throttle valve is fitted through the pre-test data, so the flow coefficient C_{d} can be regarded as a constant value, and its change is compensated by the actual flow area of the throttle valve.

If the target flow area of the throttle valve (S_{re}) is greater than the maximum flow area of the throttle valve ($S_{\max}$), it indicates that the flow area of the throttle valve is not enough to meet the cylinder target speed at this time, and it needs to enter into the variable pressure difference stage. At this stage, the maximum flow area $S_{\max}$ is taken as the actual flow area, and then the target pressure difference Δp_{re} across the throttle valve is calculated according to the orifice flow formula. At this time, the target pressure difference across the throttle valve satisfies $\Delta p_{\mathrm{re}} \geq \Delta p_{\min}$ and changes with the cylinder target speed. Then, the target pressure difference Δp_{re} and the maximum flow area $S_{\max}$ are output to the PI controller. Then, the PI controller outputs target speed n^* and target torque T^* of the motor after calculation, controlling the positive and negative rotation of the motor to realize the cylinder lifting or descending.

The speed-regulation process is divided into two stages: the constant pressure difference stage and the variable pressure difference stage. In the constant pressure difference stage, the pressure difference across the throttle valve remains unchanged, and the flow area of the throttle valve is changed to adjust the cylinder speed. For the electric proportional throttle valve, the spool displacement is only related to the input electrical signal theoretically, while in practice, the pressure difference will lead to the difference in the hydraulic force on the valve spool. Then, its actual flow area is also affected by the pressure difference. In the constant pressure difference stage, because the pressure difference is fixed, then the hydraulic force on the valve spool is almost unchanged, so the control precision of the electric proportional throttle valve will be improved. In the variable pressure difference stage, due to the limited displacement of the throttle valve spool, its maximum flow area is fixed, so the speed can only be further improved by increasing the target pressure difference across the throttle valve. At this time, the pressure difference is no longer fixed, and the hydraulic force on the valve spool is no longer fixed as well. However, the displacement of the valve spool cannot be further increased and the influence of the hydraulic force on the control accuracy of the displacement of the valve spool can be eliminated.

Theoretically, the smaller the minimum pressure difference is, the lower the throttling loss caused by the throttle valve and the higher the energy saving of the system can be. However, in practice, the minimum pressure difference cannot be infinitesimal; its value needs to be further obtained through simulations and tests.

4. Simulation Analysis

4.1. Simulation Model

To verify the feasibility of the control strategy, a simulation model of a 3 t forklift lifting system was established based on software AMESim 2020 [22,23], as shown in Figure 7. The main submodels and parameters of the simulation model are shown in Table 1.

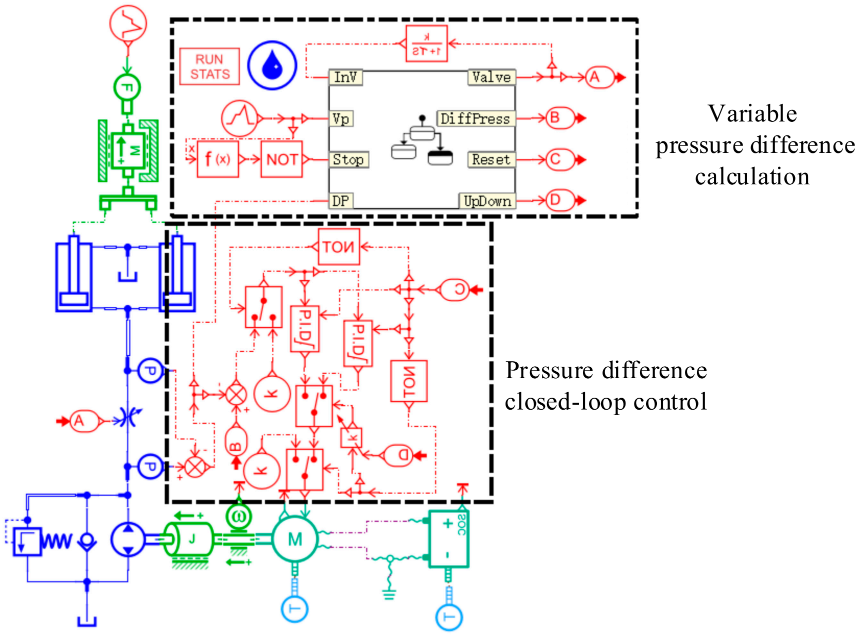


Figure 7. Simulation model of variable pressure difference pump—valve compound speed regulation.

Table 1. Main submodels and parameters in simulation.

Component	Submodel	Main Parameter	Value
Mass block	MECMAS21	Mass	720 kg
		Dynamic friction	3000 N
		Motion viscosity coefficient	5000 N/(m/s)
		Stroke	1.5 m
Hydraulic cylinder	HJ020	Cylinder diameter	54.9 mm
		Rod diameter	44.9 mm
Hydraulic pump	HYDFPM01	Displacement	25 mL/r
Electric motor	DRVEM01	Maximum torque	88 Nm
		Maximum power	13.5 kW
		Efficiency	0.9
Rotating block	MECRL0	Inertia moment	0.015 kg·m^2
		Rotary damping	0.001 Nm/(r/min)

4.2. Simulation Analysis

4.2.1. Comparison of Constant and Variable Pressure Difference Speed Regulation

Set the minimum pressure difference as 0.8 MPa and the load as 1 t. The target speed of the system is shown in Figures 8 and 9. During 2~11 s, the cylinder is lifting, and during 12~22 s, the cylinder is descending. The cylinder speed curves of constant pressure-difference speed control and variable pressure-difference speed control are shown in Figures 8 and 9, respectively.

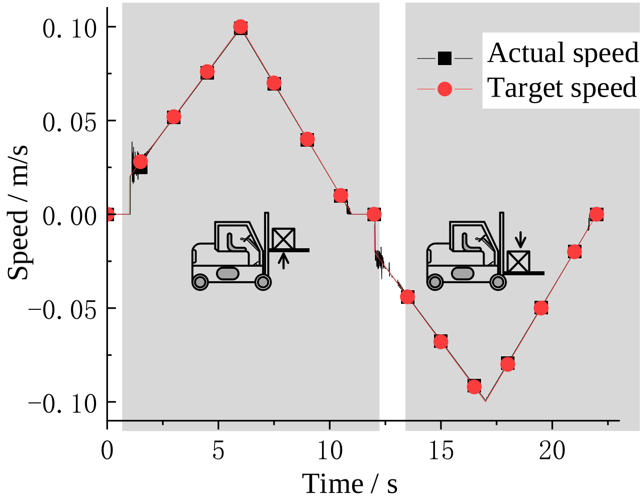


Figure 8. Cylinder speed curves under constant pressure-difference speed regulation.

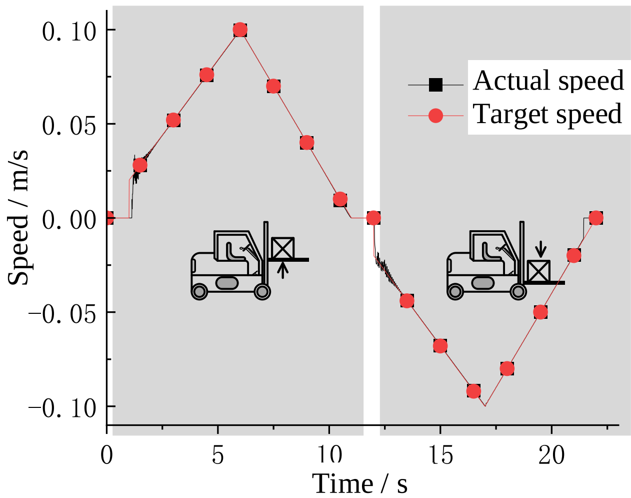


Figure 9. Cylinder speed curves under variable pressure-difference speed regulation.

By comparing Figures 8 and 9, it can be seen that no matter if the cylinder is lifting or descending, there is some extent of fluctuation and overshoot of the cylinder speed. But the overall trend is consistent with the target speed. The time to reach stability is basically the same. The stability for lifting is 0.6 s and that for descending is 1.2 s. However, under constant pressure-difference speed regulation, the cylinder speed fluctuation is higher.

Figures 10 and 11 show the curves of throttle valve opening and the pressure difference across the valve. Under constant pressure-difference speed regulation, the overall opening of the valve is small, so the pressure difference across the valve fluctuates greatly, while under variable pressure-difference speed regulation, the system's pressure-difference stability is higher. And because of the small pressure difference, the energy saving ability is also better. This indicates that the variable pressure-difference speed-regulation control strategy is feasible and can effectively improve the pump−valve compound control to optimize the system's energy saving ability and controllability.

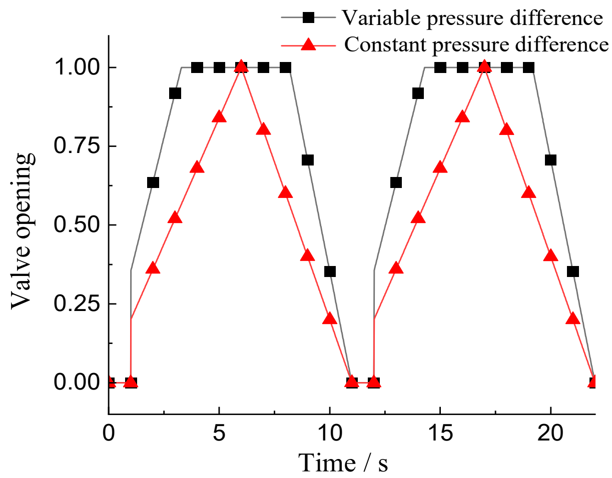


Figure 10. Throttle valve opening curves.

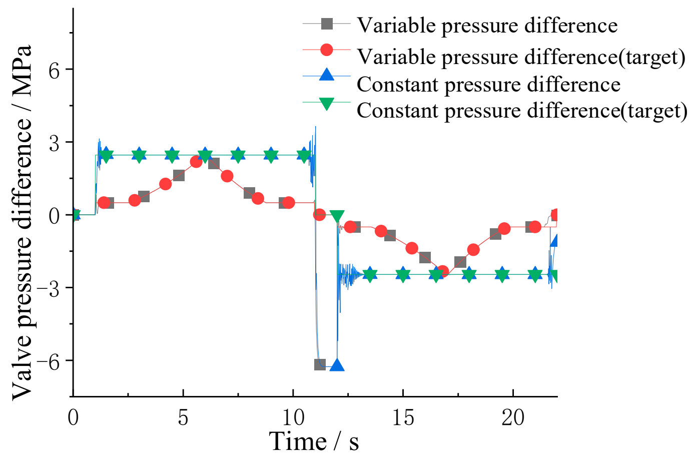
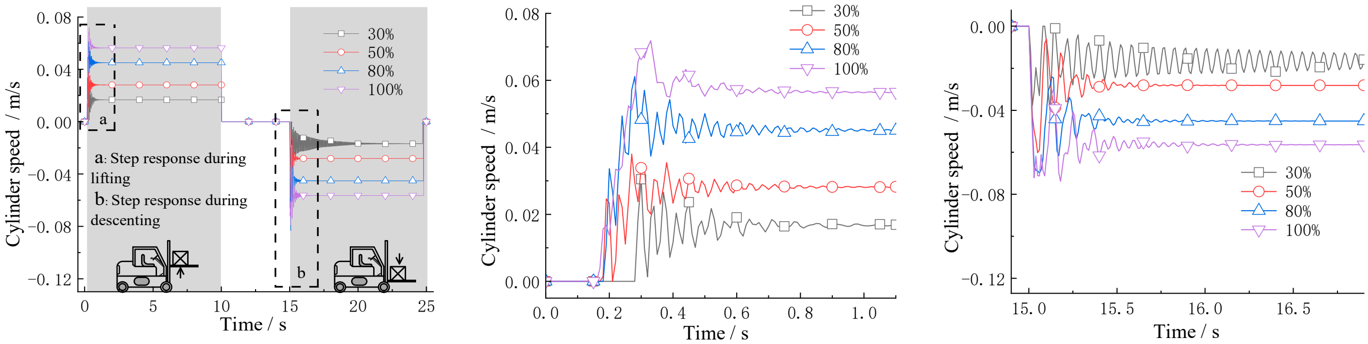


Figure 11. Curves of pressure difference across the valve.

4.2.2. Analysis of Valve Opening and Minimum Pressure Difference

By adjusting the simulation settings, the single-variable method was used to analyze the influence of the valve pressure difference and valve opening on system controllability. The control target pressure difference was the same. Under different step signals given to the throttle opening, the cylinder-speed response curves are shown in Figure 12. Within 0~10 s, the cylinder speed is positive, and the cylinder drives the load lifting. Within 10~15 s, the throttle valve is closed, the motor does not work, and the cylinder and load are stationary. Within 15~25 s, the load drives the cylinder downward, the motor reverses, and the cylinder speed is negative.

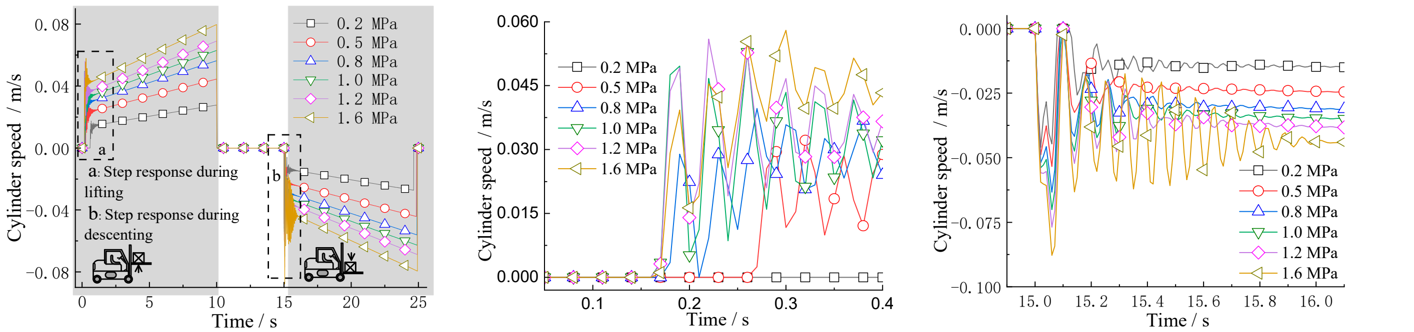
As can be seen in Figure 12b,c, when the valve opening is small, the cylinder speed fluctuates greatly and takes a long time to stabilize. With the valve opening increasing, the overshoot of the cylinder speed increases. Therefore, the appropriate increase in throttle opening is conducive to improving the system's controllability.



(**a**) (**b**) (**c**)

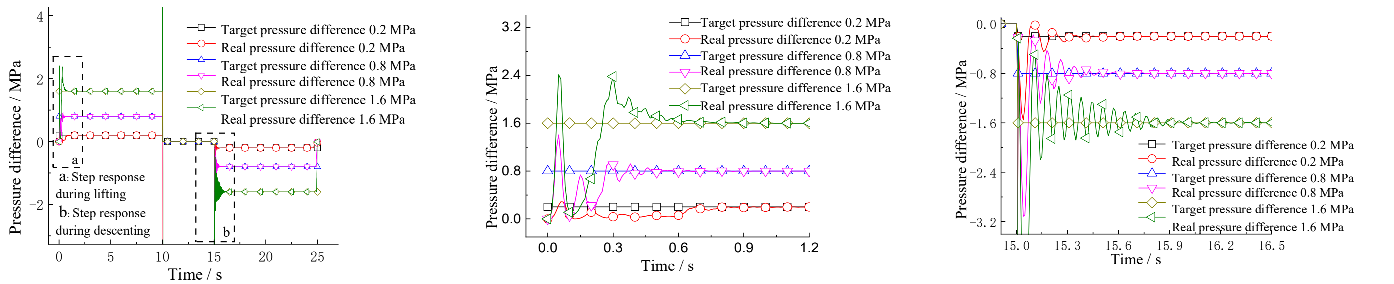
Figure 12. Cylinder speed curves under different valve openings. (**a**) Cylinder speed curves; (**b**) enlarged image of range a; (**c**) enlarged image of range b.

The valve opening was set to be the same, which was set to 50% of the valve's full opening at 0 s and 15 s and was set to increase to 100% of the valve's full opening in the following 10 s. By changing the target pressure difference across the valve, the curves of the cylinder-speed response and the pressure difference across the throttle valve are shown in Figures 13 and 14, respectively.



(**a**) (**b**) (**c**)

Figure 13. Cylinder speed curves under different target pressure differences. (**a**) Cylinder speed curves; (**b**) enlarged image of range a; (**c**) enlarged image of range b.



(**a**) (**b**) (**c**)

Figure 14. Pressure difference curves under different target pressure differences. (**a**) Curves of pressure differences across valves; (**b**) enlarged image of range a; (**c**) enlarged image of range b.

As can be seen in Figures 13 and 14, when the target pressure difference of the valve is small, the speed response of the cylinder is slow, and the speed fluctuation is relatively small. With the increase in the target pressure difference, the response speed presents a parabolic trend. The greater the pressure difference is, the more violent the speed fluctuation during descending is. Therefore, the target pressure difference cannot be set too large or too small. The simulation results show that appropriate target pressure difference is 0.8 MPa.

5. Test Analysis

To further verify the feasibility of the proposed control strategy for the lifting system of electric forklifts, a test bench was built, as shown in Figure 15, which included a power battery, battery management system (BMS), motor control unit (MCU), vehicle control unit (VCU), electric handle, permanent magnet synchronous motor (motor/generator), internal gear pump (hydraulic pump/motor), hydraulic valve block, and forklift lifting mechanism. Plate-type standard weight was used for loading.

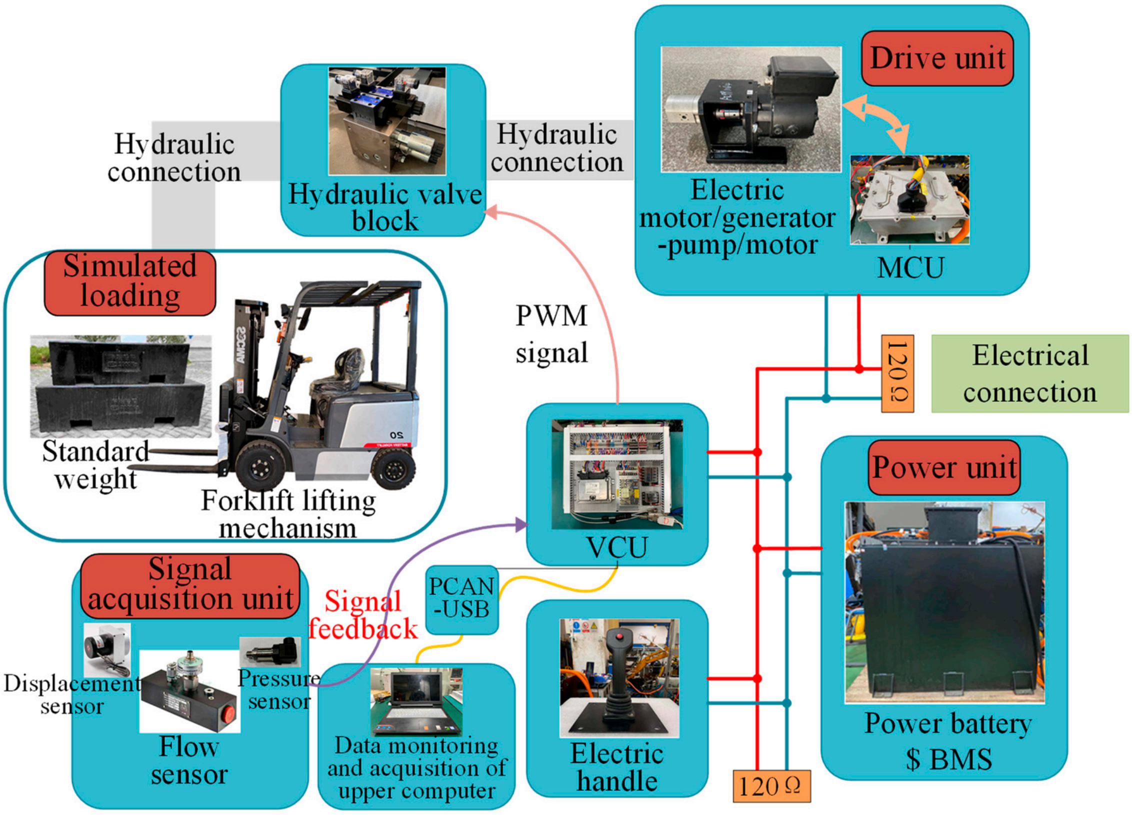


Figure 15. Electric forklift hydraulic system test bench.

After parameter matching, the key component parameters of the test bench are shown in Table 2. And the sampling acquisition frequency used was 0.01 s.

Table 2. Test bench key parameters.

Component	Parameter	Value
Hydraulic pump/motor	Rated pressure	20 MPa
	Rated displacement	25 mL/r
	Rated speed	100~2500 r/min
Electric motor/generator	Rated torque	44 Nm
	Rated speed	2000 r/min
	Rated power	9 kW
Lithium battery	Rated voltage	307 V
	Rated capacity	120 Ah

5.1. Controllability Test

Keep the target pressure difference the same, and set as 0.8 MPa. Give the throttle valve 50% and 100% step signals, respectively. Figure 16 shows the cylinder speed curves under different valve openings.

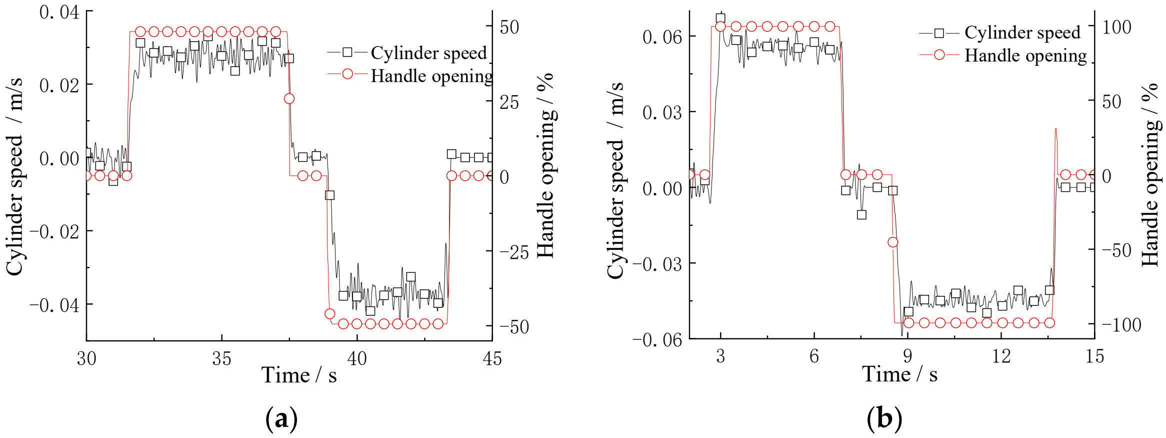


Figure 16. Cylinder speed test curves under different valve openings. (**a**) 50%; (**b**) 100%.

As shown in Figure 16, under step signal, there is a delay in speed response. With 50% of valve full opening, the rising time is about 1 s and the speed fluctuations are about 0.005 m/s, while for 100% of valve full opening, the rising time is about 0.6 s and the speed fluctuations are about 0.005 m/s too.

Figure 17 shows the valve pressure-difference test curves under different valve openings. As shown, the smaller the throttle valve opening is, the more drastic of the pressure difference fluctuation is.

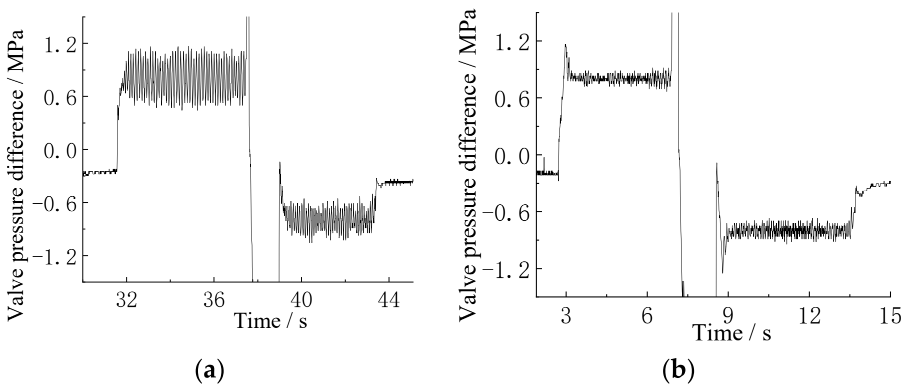


Figure 17. Throttle pressure-difference test curves under different valve openings. (**a**) 50%; (**b**) 100%.

The target pressure difference was adjusted to 0.1 MPa, 0.8 MPa, and 1.6 MPa, respectively. Under a 1 t load, give the ramp signal to the valve opening. Figure 18 shows the cylinder speed curves under different target pressure differences. One can see that, when the target pressure difference is too small (0.1 MPa), the cylinder speed fluctuates greatly and reaches ±0.01 m/s, while when the pressure difference is 0.8 MPa and 1.6 MPa, the speed fluctuation is about ±0.005 m/s. This is because when the target pressure difference is too small, if the flow rate has a little fluctuation, it will cause the pressure difference to fluctuate and it cannot be stabilized at the target value. As a result, the speed of the motor is difficult to stabilize, and the system flow fluctuation is aggravated. Therefore, the target pressure difference cannot be set too small.

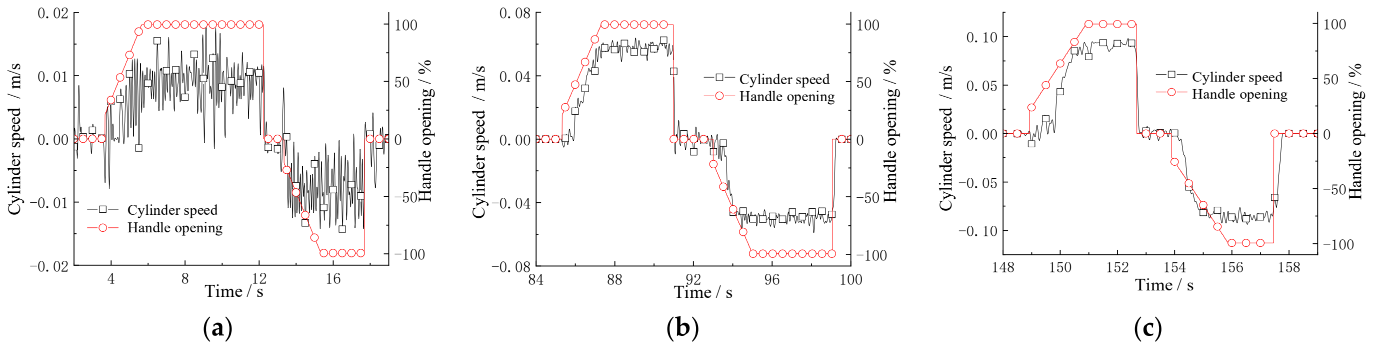


Figure 18. Cylinder speed test curves under different target pressure differences. (**a**) 0.1 MPa; (**b**) 0.8 MPa; (**c**) 1.6 MPa.

5.2. Energy-Saving Test

The energy-saving test was mainly divided into two parts; one was the system's total efficiency when lifting, and the other was the system's energy-recovery efficiency when descending. The battery energy consumed (recovered) E_{bat} can be described as

$$E_{bat} = \int_0^t U_{bat} \cdot I_{bat} dt \tag{1}$$

where U_{bat} is the battery bus voltage and I_{bat} is the battery bus current.

The load's potential energy consumed (released) E_{weight} can be calculated as

$$E_{weight} = M_{weight} g \Delta h \tag{2}$$

where Δh is the load's actual height and M_{weight} is the load's mass.

The total efficiency η_o when lifting can be given as

$$\eta_o = \frac{E_{weight}}{E_{bat}} \tag{3}$$

The energy-recovery efficiency η_i when descending can be given as

$$\eta_i = \frac{E_{bat}}{E_{weight}} \tag{4}$$

The system's energy-saving efficiency η can be deduced as

$$\eta = \eta_o \eta_i \tag{5}$$

During the test, the load mass and the target pressure difference are changed, respectively, and the step signal is given to the throttle valve. According to the system control principle, as long as the target pressure difference is the same, the cylinder speed will be the same. The pump pressure depends on the load. And the pump speed increases with the increase in target pressure difference, which is a positive correlation.

Figure 19 shows the system's total efficiency curves under different pump pressures and pump speeds during the lifting stage. Table 3 shows the system's total efficiency under different pressure differences and load masses during the lifting stage.

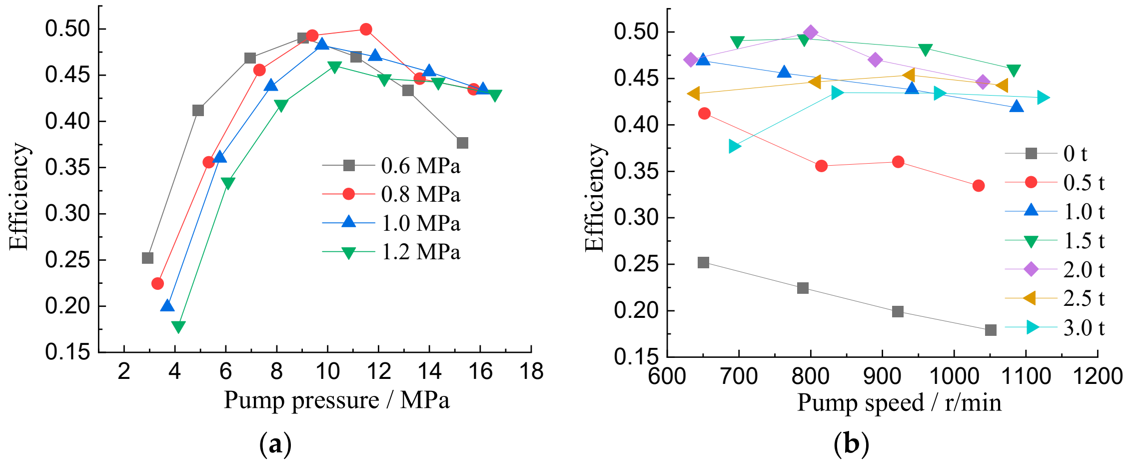


Figure 19. Total efficiency test curves under different pressures and pump speeds during lifting stage. (**a**) Efficiency vs. pump pressure; (**b**) efficiency vs. pump speed.

Table 3. System total efficiency during lifting stage.

Load Mass (t) \ Pressure Difference (MPa)	0.6	0.8	1.0	1.2
0	25.22	22.45	19.91	17.91
0.5	41.23	35.58	36.02	33.45
1.0	46.89	45.56	43.79	41.86
1.5	49.06	49.29	48.23	46.02
2.0	47.01	49.96	47.01	44.63
2.5	43.36	44.63	45.36	44.24
3.0	37.71	43.47	43.41	42.95

As can be seen in Table 3 and Figure 19, when the load mass is within the range of 0~1.0 t, the pressure difference across the valve is 0.6 MPa, and the lifting efficiency is higher. The lifting efficiency gradually decreases with the increase in the target pressure difference. This is because of the following: (1) When the load is unchanged, the increase in the target pressure difference will lead to an increase in pump pressure, which makes the pump leakage increase. (2) The increase in the target pressure difference also means that the speed is accelerated, and the pump leakage is increased. (3) The increase in target pressure difference leads to an increase in system throttling loss. The above impacts lead to a decrease in the system efficiency.

When the load mass is in the range of 1.5~2.5 t, the pump speed increases with the increase in the target pressure difference. And when the load is medium (4~16 MPa), the pump efficiency increases with the increase in the speed. Although the throttling loss is also increasing, under the comprehensive influence, the system's lifting efficiency is higher when the target pressure difference is 0.8 MPa than when it is 0.6 MPa. Similarly, when the load mass is 2.5 t, the system's lifting efficiency is higher when the target pressure difference is 1.0 MPa than when it is 0.8 MPa.

When the load mass is greater than 2.5 t, the volumetric efficiency of the pump is dominant, which makes the lifting efficiency decrease with the increase in the target pressure difference.

When the pump pressure is lower than 8 MPa, the lower the target pressure difference is, the higher the system efficiency is. When the pump pressure is greater than 8 MPa, the system efficiency is the highest when the target pressure difference is 0.8 MPa. When the load is greater than 1 t, the system efficiency decreases slightly with the increase in motor speed, but the overall efficiency is higher.

Figure 20 shows the system's recovery efficiency under different motor pressures and motor speeds during the descending stage.

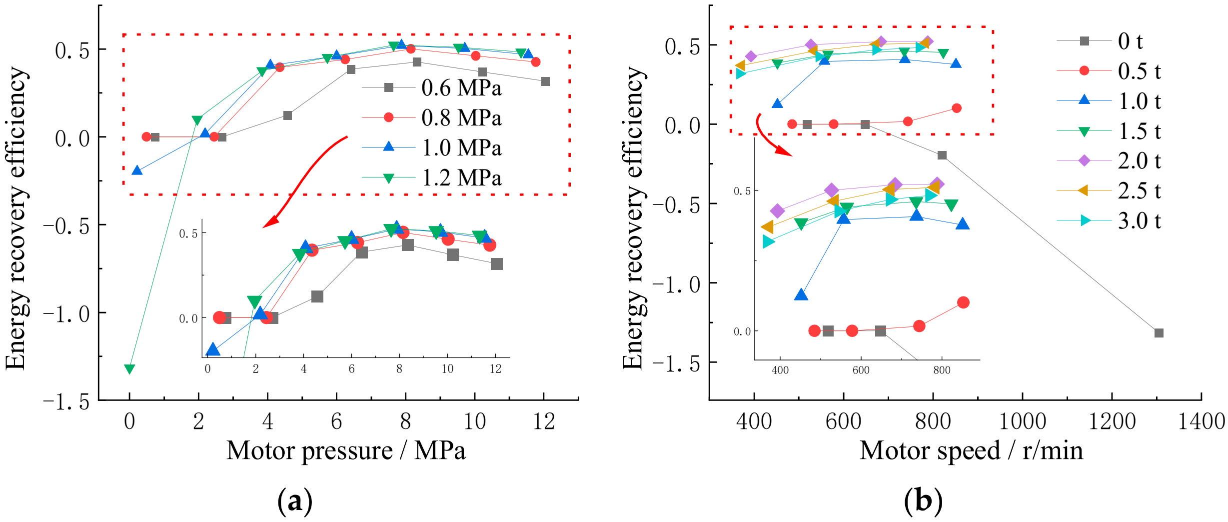


Figure 20. Energy-recovery efficiency test curves under different pressures and speeds during descending stage. (**a**) Energy-recovery efficiency vs. motor pressure; (**b**) energy-recovery efficiency vs. motor speed.

Different from the lifting stage, when descending, if the load mass is small (0~0.5 t), increasing the target pressure difference across the valve not only cannot recover the potential energy, but there will also be energy consumption. Therefore, when the load mass is too small, it should be switched to the traditional valve-controlled speed-regulation mode for descending and give up the potential energy recovery.

When the load mass increases between 0.5 and 1.5 t and the target pressure difference is 1.0 MPa, the recovery efficiency is the highest. In this interval, reducing the target pressure difference will lead to the reduction of pump speed and the pump's efficiency, while increasing the target pressure difference will reduce the inlet pressure of the hydraulic motor, which leads to the decrease in the mechanical efficiency of the hydraulic motor.

However, when the load mass is greater than 1.5 t, the larger target pressure difference across the valve can reduce the inlet pressure of the hydraulic motor, which can reduce the mechanical efficiency of the hydraulic motor but can improve its volumetric efficiency. In addition, due to the high speed, the volumetric efficiency has a great impact on the hydraulic motor efficiency. Therefore, the recovery efficiency is higher when the target pressure difference is 1.2 MPa.

When the inlet pressure of the hydraulic motor is greater than 2 MPa, the system's recovery efficiency is significantly improved. When the inlet pressure is greater than 8 MPa, the recovery efficiency begins to decrease, but the efficiency remains high. When the load mass is less than 1 t, the energy-recovery efficiency is low. The system efficiency decreases rapidly with the increase in motor speed when there is no load. When the load mass is greater than 1 t, the system's energy-recovery efficiency increases first and then decreases with the increase in the motor speed, but the efficiency is high.

Based on the above test data and combined with the system requirements for controllability and energy saving, the minimum pressure difference Δp_{min} switching table was developed and is shown in Table 4.

Table 4. Minimum pressure difference switching table.

Stage	Load Range (t)	Minimum Pressure Difference Δp_{min} (MPa)
Lifting	0~1.0	0.6
	1.0~3.0	0.8
Descending	0~0.5	0.6
	0.5~1.5	1.0
	1.5~3.0	1.2

5.3. Variable Pressure-Difference Pump—Valve Compound Speed-Control Test

The test was carried out under a 1 t load and the target speed signal was slowly given. The test results during lifting and descending with variable pressure difference pump—valve compound speed control are shown in Figures 21 and 22, respectively.

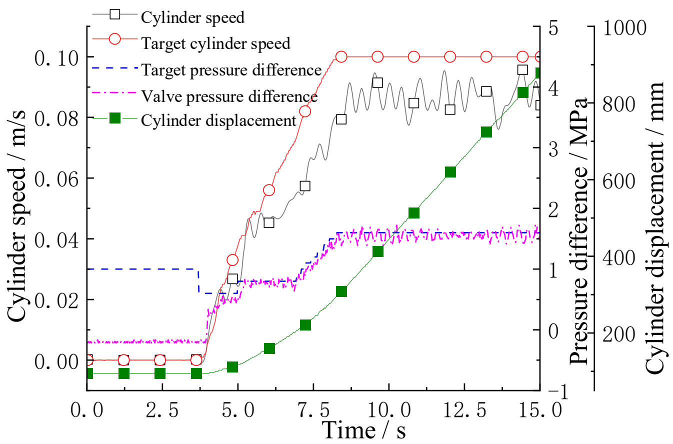


Figure 21. Variable pressure difference pump—valve compound speed control following ramp lifting.

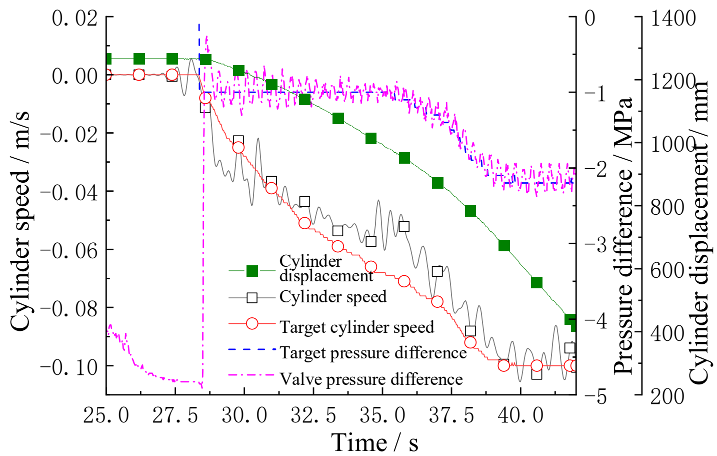


Figure 22. Variable pressure difference pump—valve compound speed control following ramp descending.

As can be seen in Figure 21, at 3.7 s, the system receives the target speed signal sent by the handle, and then starts running. Since the load has not completely left the ground, the system pressure has not reached the switching point, so the system's minimum pressure difference is set to 0.6 MPa. At 4.9 s, the load completely leaves the ground, the minimum pressure difference jumps to 0.8 MPa, and then remains unchanged. At 5.8 s, the cylinder speed no longer coincides with the target speed, but the trend is consistent. This is due to the reduction in the volumetric efficiency of the pump/motor. At 7.0 s, the throttle valve has been fully opened, and the speed increase is achieved by increasing the target pressure difference. At 8.3 s, the target speed reaches the maximum value, the valve pressure difference is stable at about 1.6 MPa with an error of ±0.15 MPa, and the actual cylinder speed is about 0.09 m/s with an error of ±0.005 m/s.

As can be seen in Figure 22, at 28.3 s, the handle sends the descending signal, and the target pressure difference is stable at −1.0 MPa until 35.6 s. At this time, the throttle valve is fully open, and the pressure difference needs to be increased to match the target speed demand. Therefore, the pressure difference gradually increases until the throttle pressure difference finally stabilizes at 2.19 MPa with an error of ±0.22 MPa at 39.8 s, and the cylinder speed is 0.098 m/s with an error of ±0.008 m/s.

Figure 23 shows the energy consumption curves in an actual working condition with the proposed variable pressure difference pump−valve compound speed regulation. The load mass is 1 t and the load lifting height is about 1.5 m. According to Equations (3)–(5), the system efficiency is calculated to be 42% when lifting, and the energy-recovery efficiency is 51% when descending. According to the standard test conditions of forklifts, the actual energy consumption during lifting is about 12.45 kWh and the recovered energy during descending is 2.67 kWh under the proposed variable pressure difference pump−valve compound speed regulation. Therefore, the energy-saving efficiency of the proposed system is 21.45%. The system's energy-recovery efficiency increases first and then decreases with the load increasing. Therefore, when the system load is more than 1 t, the energy-saving efficiency is more than 21.45%, while that in the literature [15] is 19%.

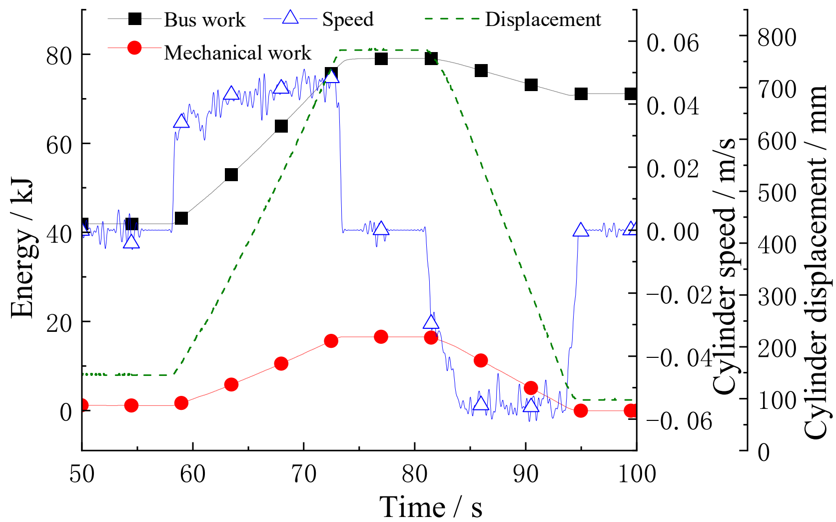


Figure 23. Energy-consumption curves of variable pressure-difference pump−valve compound speed-regulating system.

6. Conclusions

In view of the large amount of throttling losses and potential energy losses in the hydraulic lifting systems of traditional electric forklifts, an integrated energy-saving system of lifting drive and recovery based on an electric motor/generator−pump/motor with valve pressure-difference control is proposed which can reduce throttling loss, realize the load's potential energy-recovery when the cylinder is descending, and improve the efficiency and endurance of the vehicle.

A variable pressure-difference pump−valve compound speed-regulation control strategy based on the balance of control and energy saving is proposed. The simulation and experiment results show that the proposed speed regulation can improve the system's controllability and energy-saving efficiency. The test results show that under a 1 t load, the lifting efficiency is 42%, the recovery efficiency is 51%, and the energy-saving efficiency can reach 21.45%.

In future research, the minimum pressure difference switching point could be further refined to further clarify the zero point of recovery efficiency and optimize the system controllability during the small-load descending stage.

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References

1. Edamura, M.; Ishida, S.; Imura, S.; Izumi, S. Adoption of electrification and hybrid drive for more energy-efficient construction machinery. *Hitachi Rev.* **2013**, *62*, 118.
2. Yu, Y.X.; Ahn, K.K. Application of hydraulic transformer on energy saving for boom system of hybrid hydraulic excavator. *Appl. Mech. Mater.* **2017**, *868*, 118–123. [CrossRef]
3. Zhang, J.; Yang, H.; Lu, Y. Research on speed control system of VVVF hydraulic elevator with energy feedback. *Chin. J. Mech. Eng.* **2000**, *36*, 61–65. [CrossRef]
4. Lin, T.; Lin, Y.; Ren, H.; Chen, H.; Li, Z.; Chen, Q. A double variable control load sensing system for electric hydraulic excavator. *Energy* **2021**, *223*, 119999. [CrossRef]
5. Fu, S.; Wang, L.; Lin, T. Control of electric drive powertrain based on variable speed control in construction machinery. *Autom. Constr.* **2020**, *119*, 103281. [CrossRef]
6. Fu, S.; Li, Z.; Lin, T.; Chen, Q.; Ren, H. A positive flow control system for electric excavators based on variable speed control. *Appl. Sci.* **2020**, *10*, 4826. [CrossRef]
7. Xu, B.; Ding, R.; Zhang, J.; Cheng, M.; Sun, T. Pump/valves coordinate control of the independent metering system for mobile machinery. *Autom. Constr.* **2015**, *57*, 98–111. [CrossRef]
8. Minav, T.A.; Laurila, L.I.; Pyrhönen, J.J. Analysis of electro-hydraulic lifting system's energy efficiency with direct electric drive pump control. *Autom. Constr.* **2013**, *30*, 144–150. [CrossRef]
9. Minav, T.; Murashko, K.; Laurila, L.; Pyrhönen, J. Forklift with a lithium-titanate battery during a lifting/lowering cycle: Analysis of the recuperation capability. *Autom. Constr.* **2013**, *35*, 275–284. [CrossRef]
10. Minav, T.; Immonen, P.; Pyrhönen, J.; Laurila, L. Effect of PMSM sizing on the energy efficiency of an electro-hydraulic forklift. In Proceedings of the XIX International Conference on Electrical Machines—ICEM 2010, Rome, Italy, 6–8 September 2010; pp. 1–6.
11. Minav, T.; Hänninen, H.; Sinkkonen, A.; Laurila, L.; Pyrhönen, J. Electric or hydraulic energy recovery systems in a reach truck–a comparison. *Stroj. Vestn.-J. Mech. Eng.* **2014**, *60*, 232–240. [CrossRef]
12. Fu, S.; Chen, H.; Ren, H.; Lin, T.; Miao, C.; Chen, Q. Potential energy recovery system for electric heavy forklift based on double hydraulic motor-generators. *Appl. Sci.* **2020**, *10*, 3996. [CrossRef]
13. Wang, T.; Wang, Q.; Lin, T. Improvement of boom control performance for hybrid hydraulic excavator with potential energy recovery. *Autom. Constr.* **2013**, *30*, 161–169. [CrossRef]
14. Xia, L.; Quan, L.; Yang, J.; Zhao, B. Research on the method of hydraulic-gas energy storage balancing hydraulic excavator boom weight. *J. Mech. Eng.* **2018**, *54*, 197–205. [CrossRef]
15. Yu, Y.-X.; Ahn, K.K. Energy Saving of an Electric Forklift with Hydraulic Accumulator. In Proceedings of the 2019 19th International Conference on Control, Automation and Systems (ICCAS), Jeju, Republic of Korea, 15–18 October 2019; pp. 408–411.
16. Yu, Y.; Zhou, J.; Wang, K. Novel Hybrid Energy Regeneration System for Electric Construction Machinery. In Proceedings of the 2022 25th International Conference on Mechatronics Technology (ICMT), Kaohsiung, Taiwan, 18–21 November 2022; pp. 1–4.
17. Zhou, J.; Zhang, Y.; Zou, H. Energy Saving of Electric Forklift with Novel Hybrid Energy Regeneration System. *Int. Core J. Eng.* **2023**, *9*, 96–111.
18. Wang, L.; Zhao, D.; Wang, Y.; Wang, L.; Li, Y.; Du, M.; Chen, H. Energy management strategy development of a forklift with electric lifting device. *Energy* **2017**, *128*, 435–446. [CrossRef]
19. Qian, Y.; Wang, H.; Jiang, Y. Energy Recovery Research on Electric Forklift Lifting System. *Mach. Tool Hydraul.* **2018**, *46*, 61–64. [CrossRef]
20. Wang, N. Study on Energy Efficiency and Energy Saving Design of Electric Fork Lift System. Master's Thesis, Zhejiang University of Technology, Hangzhou, China, 2019.
21. Zhang, J.; Jiang, J.-H.; Li, Y.-J. Flow characteristics of different cone valves. *Jilin Daxue Xuebao (Gongxueban)/J. Jilin Univ. (Eng. Technol. Ed.)* **2016**, *46*, 1900–1905.

22. Pugi, L.; Conti, R.; Nocciolini, D.; Galardi, E.; Rindi, A.; Rossin, S. A tool for the simulation ofturbo-machine auxiliary lubrication plants. *Int. J. Fluid Power* **2014**, *15*, 87–100. [CrossRef]
23. Pugi, L.; Alfatti, F.; Berzi, L.; Favilli, T.; Pierini, M.; Forrier, B.; D'hondt, T.; Sarrazin, M. Fast modelling and identification of hydraulic brake plants for automotive applications. *Int. J. Fluid Power* **2020**, *21*, 169–210. [CrossRef]