

Article

Parameter Matching of Energy Regeneration System for Parallel Hydraulic Hybrid Loader

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Abstract: Oil shortages and environmental pollution are attracting worldwide attention incrementally. Hybrid falls within one of the effective techniques for those two problems. Taking the loader with high energy consumption and high emission as the target, combined with the hydraulic hybrid technology with high power density and strong energy storage capacity, the parallel hydraulic hybrid loader (PHHL) based on brake energy regeneration is proposed. Firstly, the dynamic models of the key components of the PHHL are established, and the parameters of the part which coincides with the ordinary loader are corrected based on the V-type duty cycle. Then, considering the energy recovery efficiency as well as the characteristics of the loader from the V-type duty cycle, the parameters for several major parts of the energy regeneration system (ERS) were calculated and matched. Then, based on the initial matching, the improved adaptive genetic algorithm (AGA) is employed to optimize the control variable of the control strategy and the design parameters of ERS to enhance the economic benefit and performance of the ERS. Furthermore, a simulation validation was conducted. Simulation results show that the ERS with optimized parameters could improve the fuel-saving effect by 25% compared to the ERS with initial parameters, which indicated the rationality of the optimized parameters. Finally, the fuel consumption test of the PHHL prototype under the V-type duty cycle is performed. The results show that the PHHL with the optimization scheme can achieve 9.12% fuel saving, which is on the brink of the potential of brake energy recovery and verifies the feasibility of applying hydraulic hybrid technology on the loader.

Keywords: parallel hydraulic hybrid loader; energy regeneration system; parameter matching; adaptive genetic algorithm; fuel saving



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1. Introduction

With the rapid development of industrial technology in the world, environmental problems and energy shortages have become increasingly serious which restrict sustainable development [1–3]. Therefore, countries and companies over the world demand the development and application of energy-saving technology to reduce pollutant emission and improve fuel economy [4]. Especially, with the development of renewable energy such as hydropower, the world gained a strong confidence in sustainable development [5–9]. However, owing to the new energy technologies such as pure electric and fuel cell are limited by the cost of the battery, mileage, and charging facilities, hybrid power as an effective intermediate scheme has attracted much attention [10–12].

According to the system configuration, the hybrid can be divided into series, parallel, and series-parallel. Furthermore, considering the energy type, the hybrid can be classified as hybrid electric and hydraulic hybrid. Undoubtedly, parameter matching of power systems is essential regardless of their type [13–15]. At present, many types of research on parameter matching of hybrid electric vehicles have been published. The basic method is to determine the required power of the powertrain through the technical indicators of

the vehicle such as accelerated start-up, uniform climbing, and maximum speed, and then calculate the parameters of the key components such as the engine [16–19], motor [16–23], transmission system [16–18,20], battery [16–21,23], and capacitor [22,24]. The basic method only considers the power demand of the extreme condition and lacks the optimization of the system configuration [25–27]. Hence, by taking into account the influence of driving cycles on the operating characteristics such as the working efficiency of the key components, researchers optimize parameters of the battery, motor, or engine. Zhu et al. [28] proposed a parameter matching method considering both the efficiency and the power demand to decide the rated speed and the rated torque, which can improve the overall efficiency of the motor. Smith et al. [29] constructed a commuter driving cycle according to the parameters defining the functionality of a light-duty vehicle and the database collected from 76 vehicles in Winnipeg. With this, the battery size for a commuter sedan car was optimized by an energy-based simulation. Chu et al. [30] developed a method to calculate the power and speed of the driving motor based on the driving cycle and probability statistics, which can optimize the basic parameters of the permanent magnet synchronous motor and improve the overall efficiency of the motor operations. Whereas due to the influences of the driving cycles on components efficiency are considered separately, the range of parameters for selection and the comprehensive optimization effect are limited. Therefore, there are studies to optimize the selection through the simulation and comparative analysis of multiple groups of parameters in the form of a parameter matrix designed by the basic parameter matching method [31–35]. Fu et al. [31] calculated the parameter range of different parts for the parallel hybrid electric vehicle by the basic method and designed the parameter matrix near the initial value, moreover, the multi-objective optimization problem consists of emissions and fuel consumption is converted into a single objective optimization problem by the weight coefficient method, then the selection of the engine, motor, battery and capacitor parameters from the parameter matrix is optimized. Wang et al. [32] optimized the hybrid energy storage system which is composed of the battery and the supercapacitor in a plug-in hybrid electric vehicle based on an enumeration method that considers cost and weight as the optimization objective. However, in the literature, it is hard to search for the whole option range. To solve this problem, various optimization methods are applied to optimize the parameters of hybrid electric vehicles [36–43]. Wang et al. [36] applied the quantum genetic algorithm to the dual-motor hybrid power system for solving the problem in optimization matching. Chen et al. [41] adopted a genetic algorithm to optimize the plug-in hybrid electric bus with a single-shaft series-parallel powertrain for enhancing power performance and economic efficiency. Borthakur et al. [42,43] applied the divided rectangles algorithm to series, improved series, and series-parallel hybrid electric vehicles, which not only met the vehicle performance requirements but also improved the fuel economy. These optimization algorithms are verified by hybrid vehicles.

The successful application of hybrid power in vehicles has guiding significance for construction machinery with large emissions and high fuel consumption. Relatively speaking, parameter matching researches on hybrid construction machinery are few which mainly focus on hybrid excavator [44,45] and hybrid-electric loader [46]. As a piece of typical construction machinery, the loader is widely used and has a large market share with the characteristics of high working mass with load and frequent start and stop, which leads to the great potential in kinetic energy recovery. It is suitable to build a hybrid loader with the goal of brake energy recovery. In addition, compared with the hybrid-electric which uses the battery or supercapacitor as an energy storage device, the hydraulic hybrid technology has the characteristics of strong energy release and storage capacity in a short time with high power density [47]. Therefore, it is worth studying the application of hydraulic hybrid in loader.

To overcome the shortcomings of previous researches and optimize parameter matching of the ERS for the PHHL which lacks theoretical study support, the optimal matching method for the ERS is proposed and a corresponding prototype is established to verify the fuel-saving effect in this paper. Firstly, the simulation model is developed by the AMESim

with a verification. Next, both the initial matching method for the parameters of ERS based on the brake strength of the V-type duty cycle and an energy management strategy, that is, a dynamic rule-based control strategy to propel the PHHL are mentioned. Then, an adaptive genetic algorithm was proposed to optimize the control variable of the control strategy and the design parameters of the ERS with the incorporation of AMESim and Python. At last, the effectiveness of the AGA was demonstrated by the simulation while the fuel-saving of the ERS after optimization was verified by both the simulation and experiment.

2. Structure and Models

2.1. Structure

The propulsion system of the PHHL considered in this paper is shown in Figure 1. Compared with the original wheel loader, an ERS is attached in the PHHL. The hydraulic system (including steering system and working system) of the PHHL is driven by the engine through gears. The motion of the PHHL is mainly driven by the engine and hydrodynamic-mechanical transmission system (HMTS), supplemented by the ERS. The ERS recovers the kinetic energy of the PHHL (named brake energy) during braking and reuses the energy while the PHHL starts to run. An accumulator is applied to store brake energy. Variable displacement pump/motor (VDPM) is applied for the conversion between mechanical energy and hydraulic energy. The power coupler regulates the power flow into the HTMS, to which an ERS is connected in parallel. The conventional version of the PHHL is a 5 t wheel loader (956T) of SINOMACH. Table 1 listed the key parameters of the conventional version of the PHHL. Besides, the ERS's parameters, including the VDPM displacement, the power coupler transmission ratio, accumulator volume, and the accumulator working pressures, need to be matched and optimized.

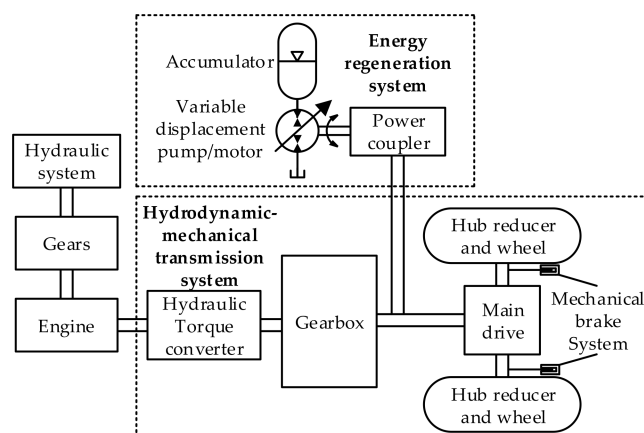


Figure 1. The architecture of the PHHL propulsion system.

Table 1. Key parameters of the 956T (conventional version of the PHHL).

Parameter Description	Symbol	Value	Unit
Operating weight	m_o	17,300	kg
Rated load	m_L	5000	kg
Engine rated power	P_e	162	kW
Engine rated speed	n_e	2000	r/min
torque ratio	K_{tc}	4 ± 0.2	—
Transmission ratio (first/second/reverse)	i_g	2.547/0.683/1.864	—
Transmission ratio of the main drive	i_{md}	4.625	—
Transmission ratio of hub reducer	i_{hr}	4.928	—
Tire power radius	r_w	0.76	m
Displacement of the steering pump	D_{sp}	83.6	mL/r
Displacement of working pump	D_{wp}	100	mL/r
Displacement of change pump	D_{cp}	60	mL/r

2.2. Engine Modeling

The main power source of the PHHL is the engine. Since the simulation of the PHHL focuses on the energy-saving performance and pays greater attention to the relationship between input and output, the engine is suitable for the hybrid modeling method supplemented by theoretical modeling and dominated by experimental modeling.

An engine with a power rating of $P_e = 162$ kW is adopted in this paper. The model of the engine needs the key parameters of the engine, the most important is universal characteristics which consist of speed characteristics and load characteristics as shown in Figure 2.

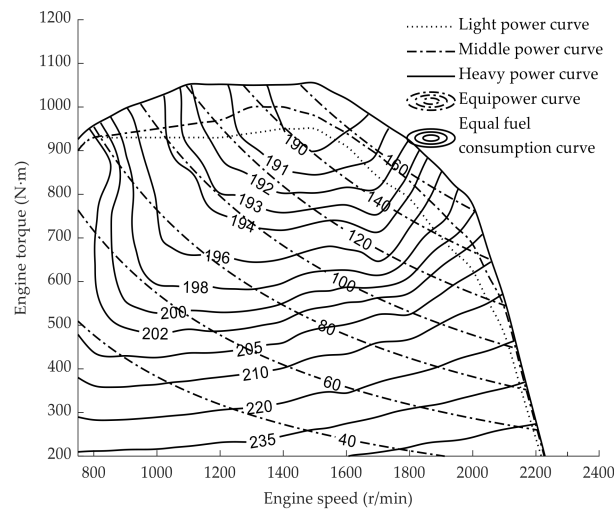


Figure 2. Universal characteristics of the engine.

Among them, the speed characteristics are the relationship between the engine speed n_e and the engine torque T_e , which is represented by below:

$$T_e = T(n_e). \quad (1)$$

The load characteristics are the relationship between the engine load (engine speed and torque) and the fuel consumption rate g_e , which is expressed as

$$g_e = L(T_e, n_e). \quad (2)$$

2.3. HMTS Modeling

HMTS includes a hydraulic torque converter, gearbox, main drive, and hub reducer. It is a typical gear transmission system, and its main parameters are the transmission ratio and efficiency. In this paper, i_g , i_{md} , and i_{hr} represent the transmission ratios of the gearbox, the main drive, and the hub reducer, respectively, and η_g , η_{md} , and η_{hr} represent the transmission efficiency of the gearbox, the main drive, and the hub reducer, respectively. The model of the hydraulic torque converter (HTC) is indicated by the mathematical relationship between hydraulic torque converter input (pump impeller) and output (turbine). The torque of pump impeller M_p is expressed by the following formula [48]:

$$M_p = \lambda_p \rho g n_p^2 D^5. \quad (3)$$

For a certain HTC, the density of working oil in HTC ρ , the effective diameter of the circular circle of HTC D , and the acceleration of gravity g can be considered as fixed values. It means M_p is decided by the torque factor of impeller λ_p and the speed of pump impeller

n_p . The λ_p is related to the transmission ratio of HTC i_{tc} , which belongs to the HTC original characteristics (M_{p1000} is the M_p while $n_p = 1000$ r/min and obtained by test) [48]:

$$\begin{aligned}\lambda_p &= f_\lambda(i_{tc}), \\ K_{tc} &= f_K(i_{tc}) = M_t/M_p, \\ M_{p1000} &= f_M(i_{tc}, n_p = 1000 \text{ r/min}), \\ i_{tc} &= n_t/n_p.\end{aligned}\quad (4)$$

Therefore, the speed of pump impeller and turbine (n_p and n_t) determine the torque of pump impeller and turbine (M_p and M_t), which can be concluded as

$$\begin{aligned}M_p &= M_{p1000} n_p^2 / 1000^2, \\ M_t &= K_{tc} M_p.\end{aligned}\quad (5)$$

2.4. ERS Modeling

The ERS consists of a power coupler, VDPM, accumulator, and hydraulic control system. The power coupler is responsible for the dynamic coupling of ERS and HMTS, which can be simplified as the clutch and transmission. According to the law of rotation, the kinetic equations of the clutch are [49]:

$$\begin{aligned}J_{cpm} (d\omega_{cpm}/dt) &= T_{cpm} - T_{cf}, \\ J_{cds} (d\omega_{cds}/dt) &= T_{cf} - T_{cds},\end{aligned}\quad (6)$$

where T_{cpm} is the VDPM torque, T_{cds} is the torque of driven shaft (far from VDPM) of the clutch, T_{cf} is the transmission torque of the clutch, ω_{cpm} is the speed of VDPM and driveshaft (near VDPM) of the clutch, ω_{cds} is the speed of the driven shaft of the clutch, J_{cpm} is the equivalent rotary inertia on the drive shaft of the clutch and J_{cds} is equivalent rotary inertia on the driven shaft of the clutch.

The maximum clutch transmission T_{cfmax} is related to the number of friction surfaces n_f , the friction coefficient u_f , the effective radius of friction force r_f , and the clamping force of friction plate F_f [50]:

$$T_{cfmax} = n_f u_f r_f F_f. \quad (7)$$

For the designed clutch, n_f , u_f , and r_f are constant, thereby the transmission torque of the clutch T_{cf} is constrained by

$$\begin{aligned}T_{cf} &\leq k_{cf} T_{cfmax}, \\ k_{cf} &= p_{cf} / p_{cfmax},\end{aligned}\quad (8)$$

where the k_{cf} is the ratio of shift working pressure p_{cf} to maximum shift pressure p_{cfmax} . In addition, the transmission of the power coupler is described by the transmission ratio i_c and the transmission efficiency η_c .

The accumulator is another power source for the PHHL, which stores the recovered brake energy. The accumulator with heat exchange in AMESim (HA0020) is considered as the model of the accumulator. The gas in the accumulator, nitrogen, is modeled as a semi-perfect which the specific capacity is following a 4th order polynomial in temperature T as described by the Janaf definition [51]. The state of the gas can be entirely determined by the volume V and the pressure P as state variables. The volume change is determined by the incoming oil, which is considered to be stiff and determined by the relation:

$$\frac{dV}{dt} = \left(\frac{\partial V}{\partial T} \right)_P \frac{dT}{dt} + \left(\frac{\partial V}{\partial P} \right)_T \frac{dP}{dt}. \quad (9)$$

VDPM can work in four quadrants (divided by speed and torque). The real flow rate of a hydraulic machine, either pump or motor, is defined as [52]

$$\begin{aligned}Q_{pm} &= e D_{pm} n_{pm} \eta_{pm}^i, \\ e &= D_{pmw} / D_{pm},\end{aligned}\quad (10)$$

where i values ± 1 according to the state of VDPM (motor: -1 , pump: 1), e is the working displacement ratio of VDPM, D_{pm} is the max displacement of VDPM, D_{pmw} is the working displacement of VDPM, n_{pm} is the VDPM speed and η_{pmv} is the volumetric efficiency of VDPM.

Similarly, the torque of a hydraulic machine, either pump or motor, is defined as [52]

$$T_{cpm} = e\Delta p_{pm} D_{pm} \eta_{pm}^{-i} / (2\pi), \quad (11)$$

where Δp_{pm} is differential pressure at the hydraulic ports of VDPM and η_{pmm} is the mechanical efficiency of VDPM.

The work efficiency is fitted according to empirical formula and test data [53]:

$$\begin{aligned} \eta_{pmv} &= \left(1 - i \frac{60C_s \Delta p_{pm}}{|e| n_{pm} u}\right)^i, \\ \eta_{pmm} &= \left(1 + i \frac{u C_v n_{pm}}{60 \Delta p_{pm} |e|} + i \frac{C_f}{|e|}\right)^i, \end{aligned} \quad (12)$$

where C_v is the laminar resistance coefficient, C_s is the laminar leakage coefficient, C_f is the mechanical resistance coefficient and u is the hydrodynamic viscosity.

2.5. Final Model of the PHHL

The engine and ERS supply the required energy fully during the PHHL operating. The power balance is described by [54]

$$P_e + P_{ERS} = P_u + P_{aux} + P_H, \quad (13)$$

where P_e is the engine power, P_{ERS} is the ERS power, P_u is the demand running power, P_{aux} is the power demand of vehicular auxiliary system and P_H is the power demand of the hydraulic system.

The longitudinal dynamics of the PHHL is [54]:

$$F_k = F_f + F_i + F_a + F_g + F_w + F_{br} = fmg \cos \theta + mg \sin \theta + C_D A_w (v_w - v_t)^2 / 21.15 + \delta ma + F_w + F_{br}, \quad (14)$$

where m is the mass of the PHHL, A_w is the front area of the loader, C_D is the air resistance coefficient, θ is the slope angle, f is the rolling resistance coefficient, v_w is the PHHL speed, v_t is the wind speed, δ is the conversion factor of the rotating mass, a is the acceleration, F_k is the propelling force, F_{br} is the mechanical braking force, F_f is the rolling resistance, F_i is the slope resistance, F_a is the air resistance, F_g is the acceleration resistance and F_w is the working resistance.

The final model is established through the Simcenter AMESim which integrates abundant libraries for the model of components and allows users to customize components at the same time. In addition, it provides a nice interface for scripting languages such as Python and enables detailed analyses relying on the strong numerical core. The model of the PHHL is shown in Figure 3.

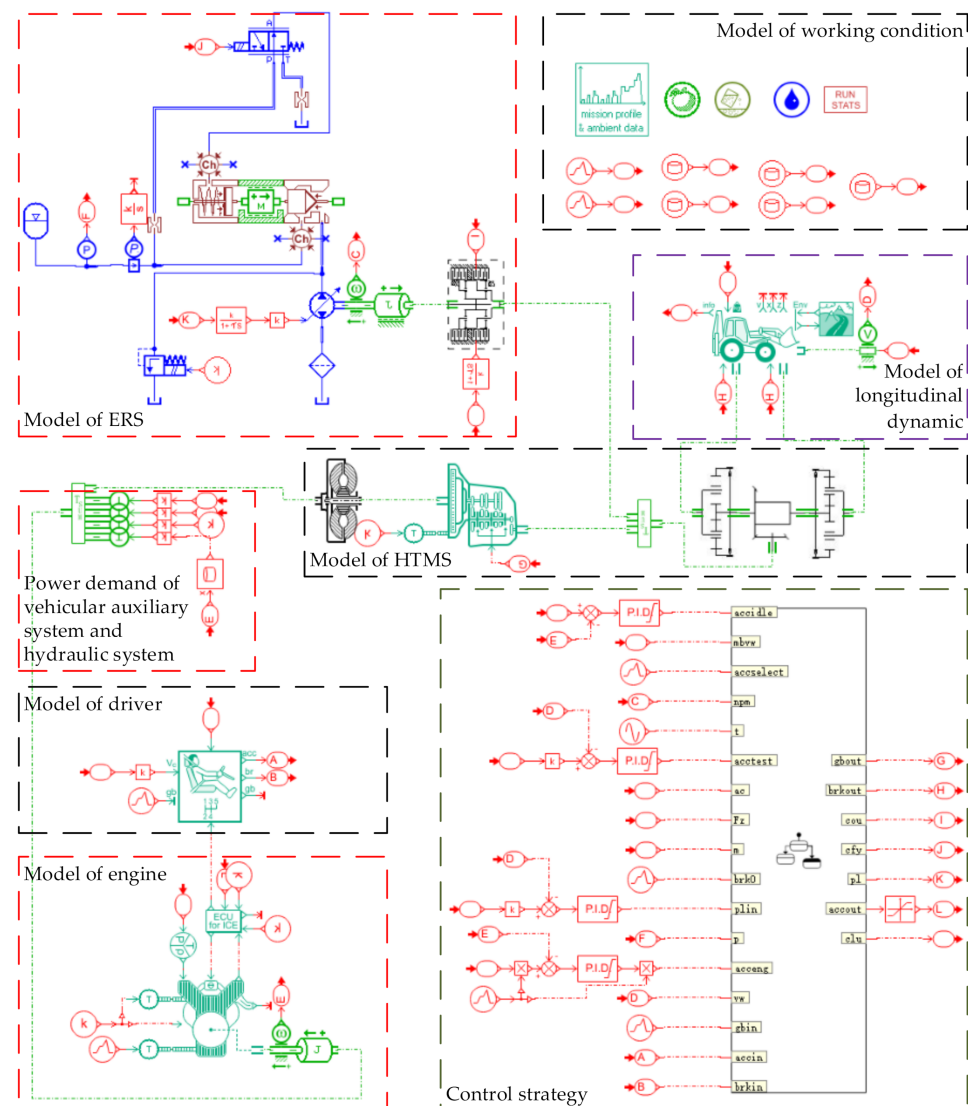


Figure 3. Final model of the PHHL.

3. Duty Cycle

3.1. Energy Saving Potential

For the PHHL, the brake energy directly affects the energy-saving effect. Therefore, it is necessary to analyze the braking condition of the duty cycle. The typical duty cycles include I-type, V-type, L-type, and T-type. Although the operating routes of those duty cycles are different, the variation of velocity and working device are similar. Among them, the V-type is widely adopted owing to its small site limitation and short duty cycle time. Figure 4 shows the V-type duty cycle adopted in this paper, which is divided into six sections. During the cycle, the operating angle is 60° . The variation of throttle and velocity is shown in Figure 5.

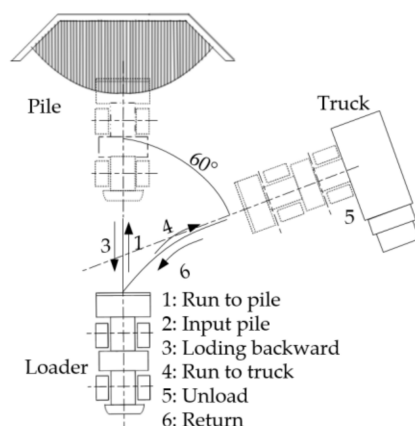


Figure 4. V-type duty cycle.

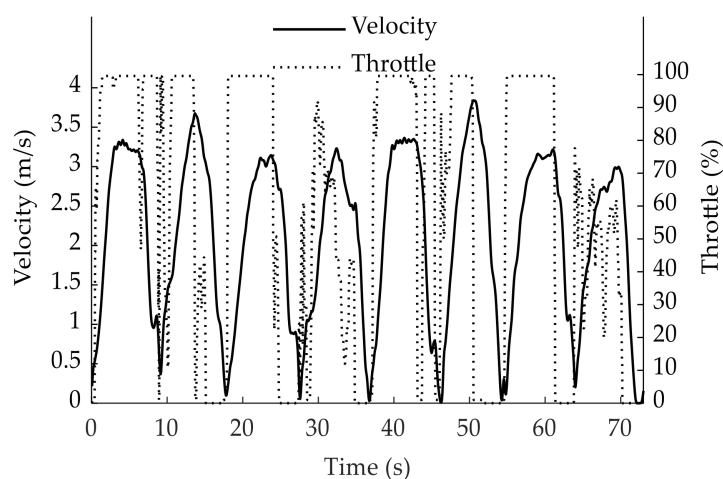


Figure 5. Velocity and throttle.

Each duty cycle includes four acceleration phases and four deceleration phases. However, according to the throttle, the first deceleration phase is caused by inserting the pile and is not suitable for recovering brake energy. In addition, according to the duty cycle, the loader is no load during the first and last deceleration, while fully loaded in the second and third deceleration. Combining velocity (initial velocity of the i th deceleration denoted v_{di}) and mass (rated load m_L is 5000 kg and operating weight m_O is 17,300 kg), the recoverable kinetic energy E_k of a duty cycle can be calculated by

$$E_k = 0.5(m_O + m_L)v_{d2}^2 + 0.5(m_O + m_L)v_{d3}^2 + 0.5m_Ov_{d4}^2, \quad (15)$$

In fact, the potential of recoverable brake energy can be estimated by the recoverable kinetic energy E_k of a duty cycle and the diesel calorific value q_c . The equivalent fuel consumption of the recoverable brake energy V_{ck} is computed by the ratio between the E_k and the effective heat energy released per unit diesel E_c which is calculated by diesel calorific value q_c , engine effective thermal efficiency η_c , and fuel consumption of a duty cycle V_c :

$$\begin{aligned} V_{ck} &= E_k / E_c, \\ E_c &= q_c \eta_c V_c. \end{aligned} \quad (16)$$

By the way, for the loader, the potential energy of the working device E_p is another direction of the energy-saving, which is calculated by the gravitational potential energy of

the bucket from the highest position to the lowest position during unloading. Similar to Equation (16), the equivalent fuel consumption V_{cp} can be estimated by

$$\begin{aligned} E_p &= m_w g H_{hl}, \\ V_{cp} &= E_p / E_c. \end{aligned} \quad (17)$$

Substitute parameters from the selected loader and its test data, the V_{ck} and V_{cp} could be obtained. The potential of recoverable brake energy is around 10%, which is far greater than that of the working device (1.6%).

3.2. Analysis of Brake Strength

Obviously, the ERS needs to provide sufficient brake strength to complete the effective recovery for brake energy. To achieve this goal, this paper analyzes the brake strength of the V-type duty cycle. In fact, the test data of velocity is fluctuant which means the acceleration fluctuating. For convenience in the analysis, the velocity is simplified, that is, data are replaced by a straight line, as shown in Figure 6. The slope of the simplified velocity is acceleration, and the acceleration of the deceleration phases is the brake strength.

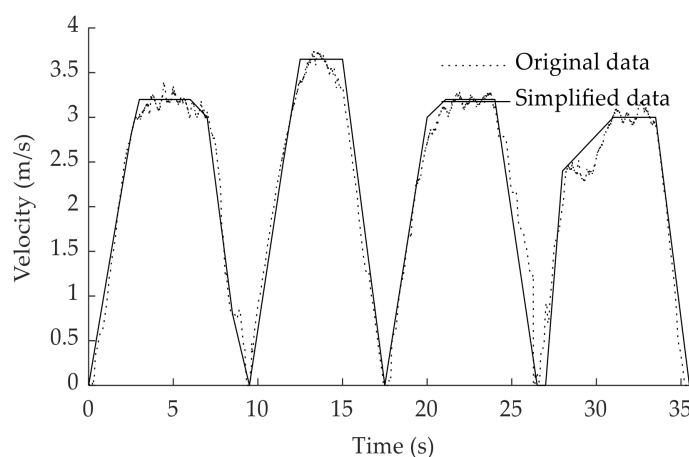


Figure 6. Velocity simplified.

As shown in Figure 7, max brake strength for recyclable deceleration of 10 duty cycles ranges from 1.2 m/s^2 to 1.9 m/s^2 . Since the first deceleration phase of each cycle is not suitable for recovering brake energy, there are three recoverable conditions: full load backward, full load forward, and unload backward (operating conditions at the second, third, and fourth deceleration of each duty cycle). The brake strength of the full load backward is the highest which means larger brake energy as shown in Figure 8. Furthermore, the greater the brake strength provided by the ERS, the less energy consumed by the mechanical brake system, which means the higher the proportion of the final recovered energy. Therefore, in the subsequent parameter matching, it is necessary to satisfy the brake strength of the full load backward to recover more brake energy.

3.3. Hydraulic Load Spectrum

The hydraulic system of the loader includes a steering system and working system, which adopts the principle of double pump confluence and steering priority. Instead of establishing a complete dynamic model, this paper chooses to simulate the consumption of the hydraulic system through the pumps and their load spectrum as shown in Figure 9. The pumps include the steering pump, working pump, and gear shift pump, and the load spectrum is obtained through experiments.

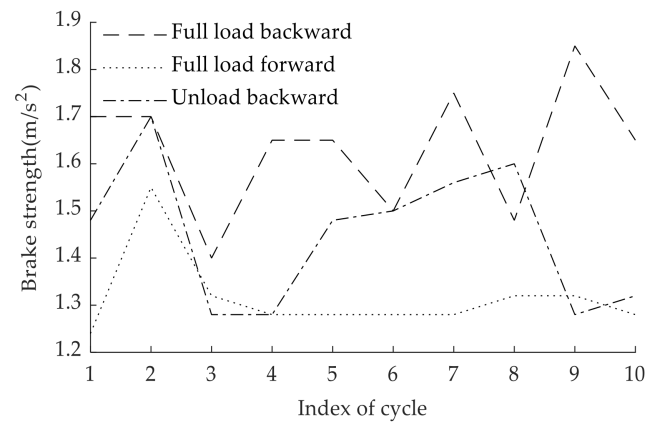


Figure 7. Max brake strength of recoverable deceleration.

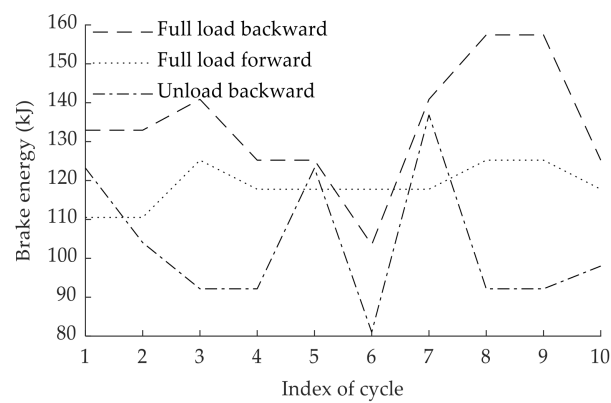


Figure 8. Max brake energy of recoverable deceleration.

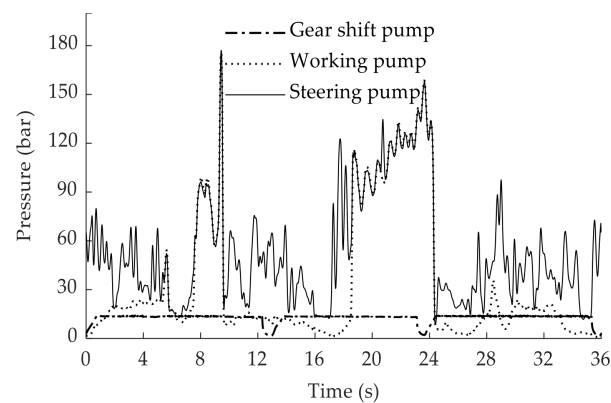


Figure 9. Pressure of pumps in a duty cycle.

Relatively, it is hard to get the data of insertion resistance (working resistance of loader) directly, and an empirical formula is used here [55]:

$$F_w = 9.8K_1K_2L_C^{1.25}B_KK_3K_4, \quad (18)$$

where K_1 is the material block coefficient, K_2 is the material properties, K_3 is the material pile height coefficient, K_4 is the bucket form coefficient, L_C is the insertion depth, B_K is the width of the bucket.

Substitute the parameters and combined them with the displacement data of bucket cylinder and boom cylinder, the insertion resistance of loader is shown in Figure 10.

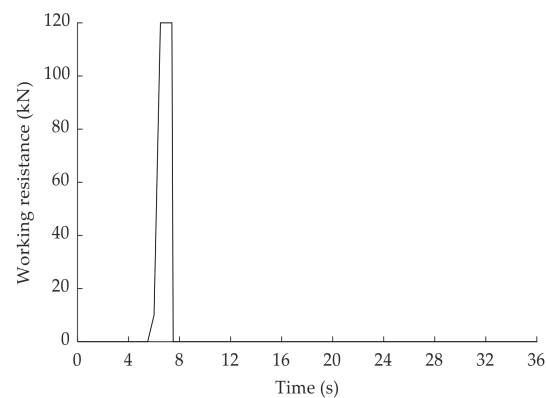


Figure 10. The working resistance of the loader.

3.4. Model Calibration

To fit the actual operation better, the control parameters of the model are corrected, mainly composed of the PID control parameters of the throttle and the PID control parameters of the mechanical brake system. As shown in Figure 11, the simulation results are close to the experimental data.

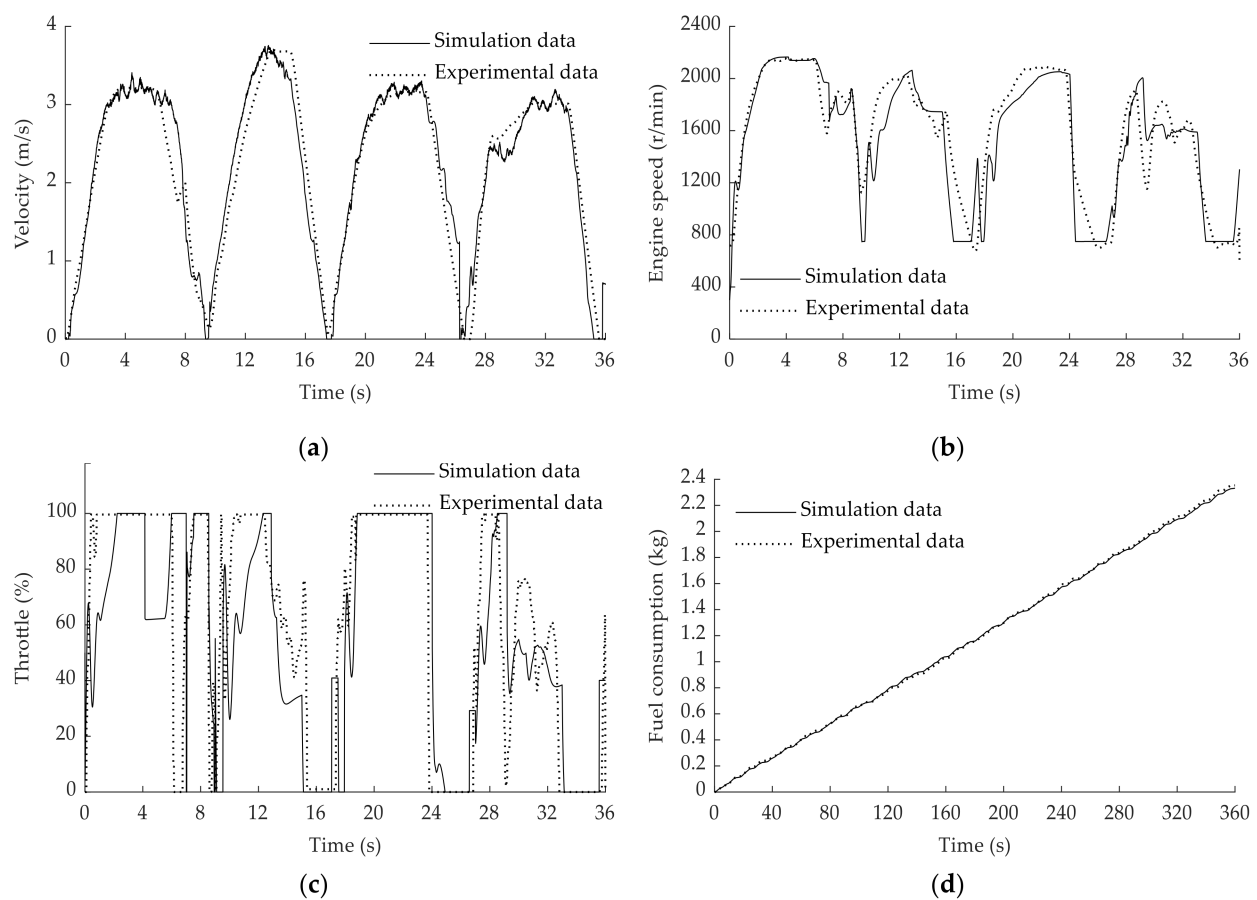


Figure 11. Simulation results. (a) velocity; (b) engine speed; (c) throttle; (d) fuel consumption.

4. Initial Parameter Matching

For the PHHL, the objective of initial parameter matching is to provide appropriate torque to recover brake energy efficiently.

4.1. Accumulator

As an energy storage element, the accumulator is responsible for rapidly absorbing brake energy and providing driving power under suitable conditions. This paper adopts the bladder accumulator with low airbag inertia, sensitive reaction, and low cost to achieve this goal. The inflation pressure p_0 , minimum working pressure p_1 , maximum working pressure p_2 , and volume V_0 determine whether the accumulator meets the need for energy storage and recovery. In fact, higher inflation pressure is beneficial to reduce the displacement of VDPM. And the maximum inflation pressure is 12 MPa while filling gas by nitrogen bottle. In this paper, 12 MPa is adopted as the inflation pressure and the maximum working pressure is not more than 31.5 MPa. Considering the service life of the accumulator, the relationship between working pressures are as follows [56]:

$$\begin{aligned} 0.25p_2 &\leq p_0 \leq 0.9p_1, \\ p_2 &\leq 3p_1. \end{aligned} \quad (19)$$

Depending on the above equation, minimum working pressure should better be higher than 13.5 MPa. On the one hand, the larger p_1 is, the smaller the VDPM displacement is, which is beneficial to the installation and application of the ERS. On the other hand, higher minimum working pressure means larger accumulator volume, which is detrimental to the installation and application of the ERS. Therefore, 18 MPa is taken here and optimized later. The volume is estimated by the storage capacity whether meets the requirements of single brake energy recovery. Figure 8 shows the maximum value of a single brake energy is 160 kJ under a V-type duty cycle. Since the accumulator absorbs oil instantaneously and the gas expands quite fast while recovering the brake energy. The change of gas can be thought of as an adiabatic process. With this, the accumulator energy storage capacity E_a is calculated from the following formula while the gas in the accumulator is assumed to obey a polytropic gas law [56]:

$$E_a = \left[1 - \left(\frac{p_1}{p_2} \right)^{\frac{1-\beta}{\beta}} \right] \frac{p_1 V_0}{\left(\frac{p_1}{p_0} \right)^{\frac{1}{\beta}} (1-\beta)}, \quad (20)$$

$$PV^\beta = \text{Constant}.$$

where β is the polytropic coefficient of gas and the constant is defined by the inflation pressure p_0 and the accumulator volume V_0 .

According to the above equation, $V_0 = 27.4$ L is obtained. Considering a certain margin, 30 L is taken by V_0 .

4.2. Coupler

The most important design parameter of the coupler is the transmission ratio which is constrained by

$$IC \leq \pi r_w n_{PM\max} / (30 v_{w\max} i_{ga} i_{md} i_{hr}), \quad (21)$$

The transmission ratio is decided by the allowable speed of VDPM $n_{PM\max}$ and maximum recovery velocity $v_{w\max}$, where $v_{w\max}$ is related to the driving cycle, $n_{PM\max}$ is related to the series, and displacement of VDPM. The bigger the transmission ratio is, the smaller the VDPM displacement is of the same torque demand, which is expected by the designer. Therefore, the matching of transmission ratio has to be combined with VDPM and cannot be calculated separately.

4.3. VDPM

The increase of VDPM displacement leads to the rise of the ERS brake capacity. However, the VDPM displacement unable to take the largest one while taking the cost, installation space, and the certain range of the brake strength for loader into account. Especially the VDPM displacement beyond a certain limit will cause the ERS incompetent to be accessed to the loader due to the space constraints. Therefore, the layout of the conventional loader is analyzed to determine the upper limit of the VDPM, then the VDPM with the smaller displacement is selected on the premise of meeting the brake strength.

Taking the conventional version of the PHHL adopted in this paper as an example, the gearbox has two forward shifts and one reverse shift. The best result is that the brake energy at maximum velocities of the three shifts can be recovered. Furthermore, the driving cycle of high braking frequency has the highest priority while the best result cannot be achieved. For the loader, the typical duty cycle with braking frequency mainly driving in first gear and reverse gear. Therefore, the basic requirement is ensuring the brake energy can be recovered under those two shifts.

Table 2 shows the brake strength for the loader which is divided into two kinds: calculated by the safety brake distance B_{s1} and obtained from the V-type duty cycle test B_{s2} .

The series of the VDPM selected in this paper is A4VSO, which is a rare VDPM with four working quadrants. Due to space constraints, the upper limit of the VDPM displacement is 250 mL/r. Table 3 shows the brake strengths provided by the VDPM with the largest transmission ratio of the power coupler. As the brake strengths provided by VDPM with displacements less than 250 mL/r (include 250 mL/r) are far from the objective brake strength in the second shift, the parameter matching of the second shift is abandoned. The most suitable VDPM displacement is 180 mL/r while considered the reverse and first shift. The next is 125 mL/r. Besides, the coupler has three shifts: neutral gear, positive gear (corresponding to the maximum velocity of the first shift), and negative gear (corresponding to the maximum velocity of the reverse shift). Additionally, the adopted gearbox is a planetary gearbox as shown in Figure 12 which is decided by the layout of the conventional version of the PHHL. Obviously, the positive gear transmission ratio i_p is always 1 higher than that of negative gear i_n . Table 4 indicates the maximum transmission ratio of the coupler when the loader drives at different shifts.

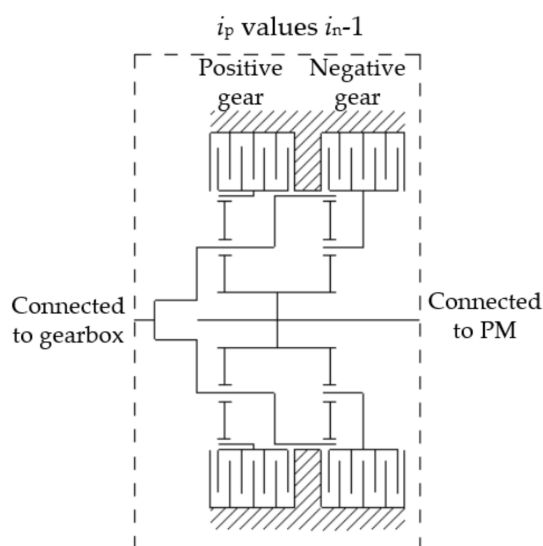


Figure 12. Transmission diagram coupler.

Table 2. Brake strength for the loader.

Working Condition	Maximum Velocity (km/h)	B_{s1} (m/s ²)	B_{s2} (m/s ²)
No load 1st shift	11.84	1.55	—
Full load 1st shift	11.79	1.23	1.55
No load 2nd shift	41.02	3.69	—
Full load 2nd shift	38.96	1.84	—
No load reverse shift	16.10	2.07	1.7
Full load reverse shift	15.92	1.56	1.8

Table 3. Brake strength for VDPM.

Working Condition	Barke Strength (Unit: m/s ²) for VDPM with Different Displacements (Unit: mL/r)				
	40	71	125	180	250
No load 1st shift	1.05	1.45	1.98	2.64	3.09
Full load 1st shift	0.87	1.18	1.59	2.10	2.46
No load 2nd shift	0.5	0.62	0.79	1	1.14
Full load 2nd shift	0.44	0.54	0.67	0.83	0.94
No load reverse shift	0.92	1.27	1.71	2.27	2.65
Full load reverse shift	0.77	1.04	1.39	1.82	2.12

Table 4. Maximum transmission ratio of the coupler for different VDPM.

Shift of Gearbox	Maximum Transmission Ratio of Coupler for Different VDPM Displacements (Unit: mL/r)				
	40	71	125	180	250
First	4.9597	4.1847	3.4098	3.2548	2.7898
Second	1.5737	1.3278	1.0819	1.0328	0.8852
Reverse	4.1966	3.5409	2.8852	2.7540	2.3606

5. Optimization

5.1. Control Strategy

Optimization-based control strategy and rule-based control strategy are two primary types of control strategies while the latter is convenient to transplant to the actual control system. This paper takes a dynamic rule-based control strategy based on the SOC (state of charge) for the accumulator. The SOC is calculated by working pressure of the accumulator p_{ac} and maximum pressure of the accumulator p_{max} (this paper takes 32 MPa which is slightly larger than 31.5 MPa):

$$SOC = p_{ac}/p_{max} \quad (22)$$

As shown in Figure 13, the control strategy is illustrated. Firstly, according to the demand torque T_{req} of driving, the state of PHHL is divided into braking ($T_{req} < 0$) or propelling ($T_{req} > 0$):

- (1) Braking: If the initial velocity v_0 is bigger than the maximum speed v_p which is corresponding to the positive gear of the power coupler, the power coupler shifts negative gear (transmission ratio is i_n), otherwise the coupler shifts positive gear (transmission ratio is i_p). Besides, the brake torque is provided by ERS prior while the insufficient part is made up by mechanical braking (T_{bm} is the mechanical brake torque).
- (2) Propelling: Firstly, if the SOC_0 (initial SOC of the current propel phase) is less than the start limit SOC_{start} or the SOC is less than lower limit SOC_{min} (accumulator pressure is p_1), the ERS stop working and the T_{req} supplied by the engine (T_{ed} is the part of the torque driven by the engine, which is used to propel the PHHL). Secondly, if the initial demand torque of driving T_{req0} is less than the drive capability of ERS at the

negative gear of the coupler which is calculated by SOC and $K(i_c)$ (a coefficient related to i_c), the coupler shifts negative gear, otherwise the coupler shifts positive gear. The propelling torque is provided by ERS prior while the insufficient part is made up by the engine.

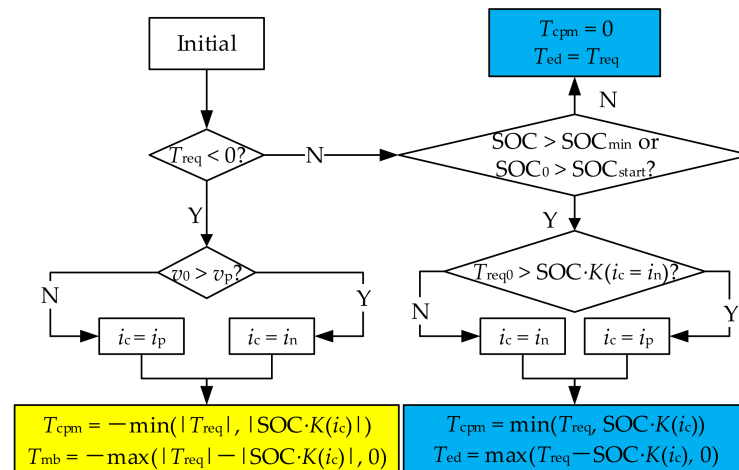


Figure 13. Control strategy.

5.2. Optimization Method

The objective function F , which is of special significance to the PHHL owner, is formulated to maximize the fuel economy. However, considering the difference between the initial SOC and the final SOC, the ratio of the recovered energy to the reused energy is taken into account as the correction factor:

$$\left\{ \begin{array}{l} F = \frac{\sum_{x=1}^{x=n} R_x}{\sum_{x=1}^{x=m} U_x} (f_{ol} - f_{PHHL}) \\ f_{EPHHL} = f_{ol} - F \end{array} \right. , \quad (23)$$

where f_{ol} is the total fuel consumption of 10 duty cycles for the original loader, f_{PHHL} is the fuel consumption of 10 duty cycles for the PHHL, f_{EPHHL} is the equivalent fuel consumption of 10 duty cycles for the PHHL, R_x is energy recovered in the x^{th} time, U_x is energy reused in the x^{th} time, m is the maximum times of energy reused, n is the maximum times of energy recovered.

The optimization variables include the design parameters D_{PM} , V_0 , p_1 , i_p , i_n , and the control variable SOC_{start} . According to the initial parameter matching, D_{PM} values 125 mL/r and 180 mL/r, i_p values $i_n - 1$. Therefore, the number of the optimization variables can be reduced to four: V_0 , i_p , p_1 , and SOC_{start} while the optimization process is carried out in two cases where the D_{PM} values 125 mL/r and 180 mL/r, respectively.

Based on the initial parameter matching for the accumulator, V_0 is around 30 L and more inclined to be greater than this value, this paper takes 20–60 L.

According to the initial parameter matching of accumulator and control strategy, the SOC_{min} and SOC_{start} satisfy:

$$p_1/p_{max} = SOC_{min} \leq SOC_{start} \leq p_2/p_{max}, \quad (24)$$

The minimum working pressure p_1 is between 13.5 and 31.5 MPa and takes 14–28 MPa after considering the energy storage capacity. Similarly, after a certain margin is reserved, SOC_{start} ranges from SOC_{min} to 0.9375 according to Equation (24).

The coupler transmission ratio i_p is related to D_{PM} which ranges from 2.89 to 3.89 for VDPM with 125 mL/r while this value ranges from 2.76 to 3.76 for VDPM with 180 mL/r according to Table 4. Besides, due to the D_{PM} only have two options, the optimization process is carried out with different VDPM displacements, respectively.

The flowchart of the adaptive genetic algorithm (AGA) proposed to solve the optimization problem is presented in Figure 14 which is programmed in Python:

- (1) Initial parameters: Determine the optimization parameters and their ranges;
- (2) Code parameters: The parameters to be optimized are digitized by binary coding method;
- (3) Initial population and $i_{\text{ter}} = 0$, $\text{Flag}_{\text{con}} = 0$: A population is initialized randomly with the population size of 20. The differences between individuals in the population are the design parameters and the control parameter (i_p , p_1 , V_0 , and $\text{SOC}_{\text{start}}$) after coding. In addition, the number of iterations i_{ter} and the number of convergence Flag_{con} are set to 0 in this step;
- (4) Exit criteria: If the number of convergence Flag_{con} is equal to 4 or the i_{ter} is more than 500, the program goes to step (12), otherwise, the program goes to step (5);
- (5) Parameters decode and simulation by AMESim: The decoded individuals in the population are input into the model for simulating by AMESim;
- (6) Population fitness and $i_{\text{ter}} = i_{\text{ter}} + 1$: The fitness of the individuals in the population are calculated according to the Equation (23) and the number of iterations i_{ter} is increased by 1;
- (7) Convergence: The criterion for the first 3 times of convergence is that the best individual difference in 6 consecutive iterations does not exceed 0.5%, and the criterion for the last convergence is the difference in 10 consecutive iterations. If the fitness converges, the program goes to step (8), otherwise, the program goes to step (9);
- (8) $\text{Flag}_{\text{con}} = \text{Flag}_{\text{con}} + 1$: The number of convergence Flag_{con} is increased by 1;
- (9) Calculate crossover and mutation variation rates: Determine the probability of crossover and mutation variation of individuals in the population;
- (10) Parameters code, select, cross and mutate: The sire individuals are selected according to the fitness of individuals in the current population, and then the coded parameters are crossed and mutated according to the cross rate and mutation rate, thus the filial individuals are generated;
- (11) Regenerate population: Summarize the filial individuals generated in step (10) to generate a new generation of the population, then the program goes to step (4);
- (12) Decode parameters: Decode the optimized parameters;
- (13) Output parameters: Output the optimized parameters.

Furthermore, to reduce the local convergence, this paper has made the following improvements compared to the general genetic algorithm:

- (1) Exit criteria: The convergence occurs four times or the iterations is more than 500;
- (2) The rates of crossover and mutation change with convergence.

The crossover and mutation variation rates (r_{cross} and r_{muate}) are related to the number of convergence Flag_{con} :

$$\begin{aligned} r_{\text{cross}} &= \begin{cases} r_{\text{cross}0} \cdot k_{\text{cross}}^{\text{Flag}_{\text{con}}} & \text{Non-convergence} \\ 0.8 & \text{Convergence} \end{cases} \\ r_{\text{muate}} &= \begin{cases} r_{\text{muate}0} \cdot k_{\text{muate}}^{\text{Flag}_{\text{con}}} & \text{Non-convergence} \\ 0.8 & \text{Convergence} \end{cases} \end{aligned} \quad (25)$$

where the initial values of the r_{cross} and r_{muate} ($r_{\text{cross}0}$ and $r_{\text{muate}0}$) in this paper are taken as 0.5 and 0.07, respectively, while the growth factors of the r_{cross} and r_{muate} (k_{cross} and k_{muate}) value 1.15 and 1.3, respectively, in this paper.

Besides, as the PHHL model is closer to the actual loader, the cost of simulation time is correspondingly larger. Hence, the range and number of the optimization parameters in the optimization method proposed in this paper should be simplified according to the design objectives before optimization, otherwise, the simulation time will become a disaster.

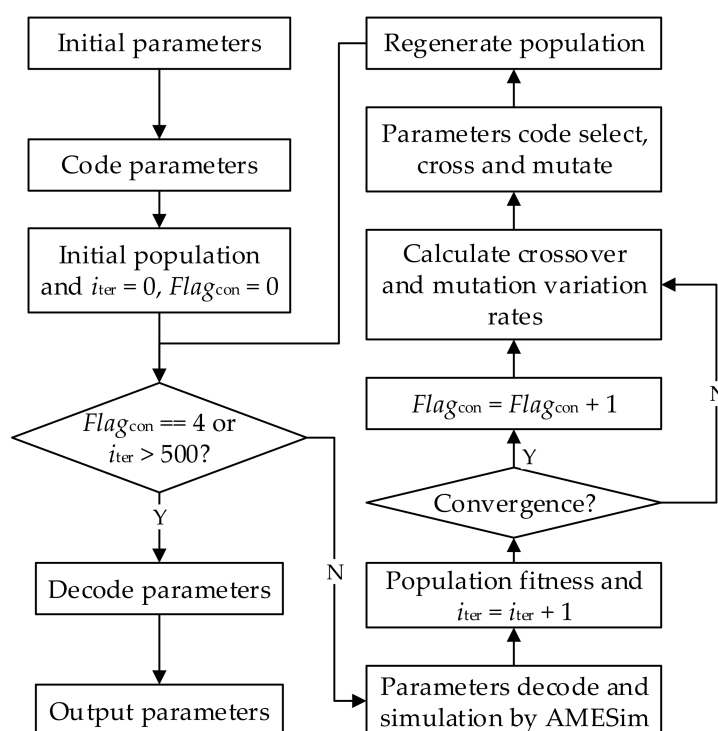


Figure 14. Flowchart of the optimization method.

6. Results & Discussions

6.1. Optimization Results

The optimization values of the objective function (equivalent fuel consumption) of the best individuals in each generation are shown in Figure 15. According to the variation of the optimization objective, the best individual in the whole optimization process is decided. Obviously, the VDPM with 180 mL/r has better fuel economy. In fact, Figure 15 shows the objective value fluctuates increases after convergence, which is in line with the proposed optimization method AGA, that is, the mutation rate and cross rate of the next-generation population after convergence are set to be larger. Besides, the optimization method proposed in this paper performs within 120 iterations in each optimization group distinguished by the VDPM displacement. By applying the optimization method AGA, the first convergence of the target achieves around the 30th iteration and all the convergences occur within 35 generations, which means the convergence speed is rapid.

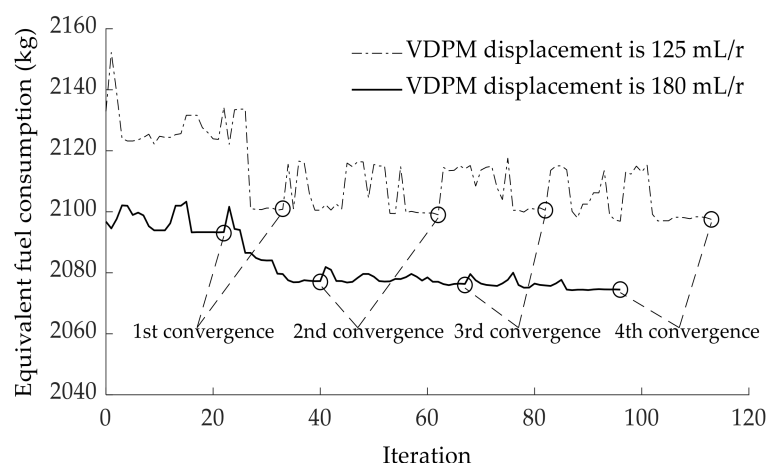


Figure 15. Variation of equivalent fuel consumption.

The search scopes of the control value and design parameters are given in Figures 16 and 17, respectively. To describe the distribution relationship of variables within its design range, the control value and design parameters are drawn on a standardized scale of [0, 1] according to the constraints of the parameters. Among them, Figure 16 illustrates the quality control chart of the design variables of different VDPM displacements. Obviously, the design variables are almost full of the search space, which means the global search capability of AGA is good, especially for the group with a better fuel-saving effect (VDPM displacement is 180 mL/r). Furthermore, as shown in Figure 17, the search deviation of the control variable is relatively large and the fluctuation is frequent, which can reduce local convergence and contribute to achieving global optimization. Combining Figures 15–17, the improved algorithm AGA can be considered effective.

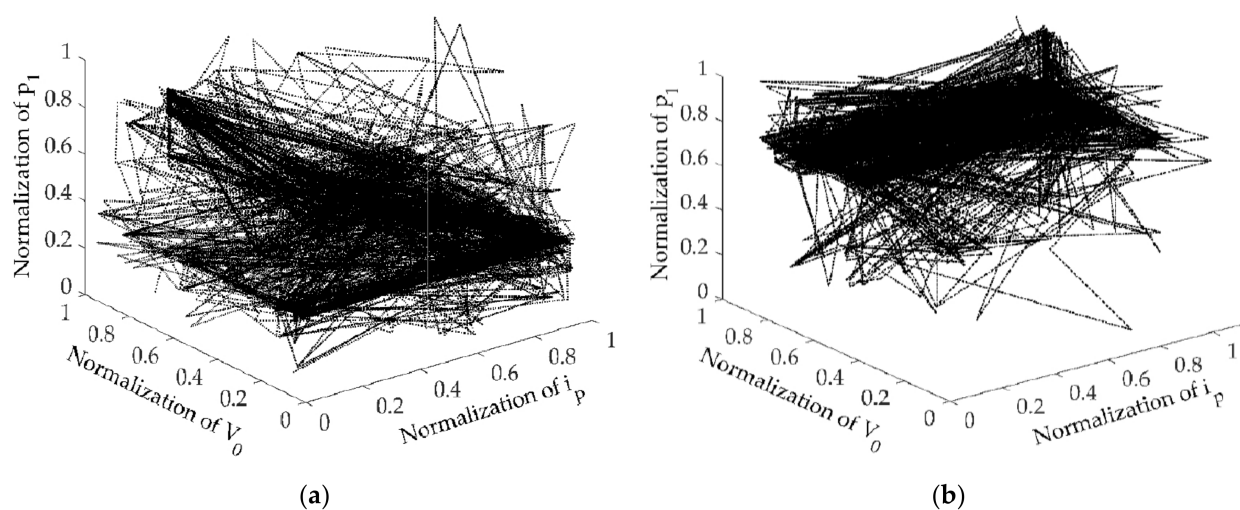


Figure 16. Variation of design variables. (a) VDPM displacement is 180 mL/r; (b) VDPM displacement is 125 mL/r.

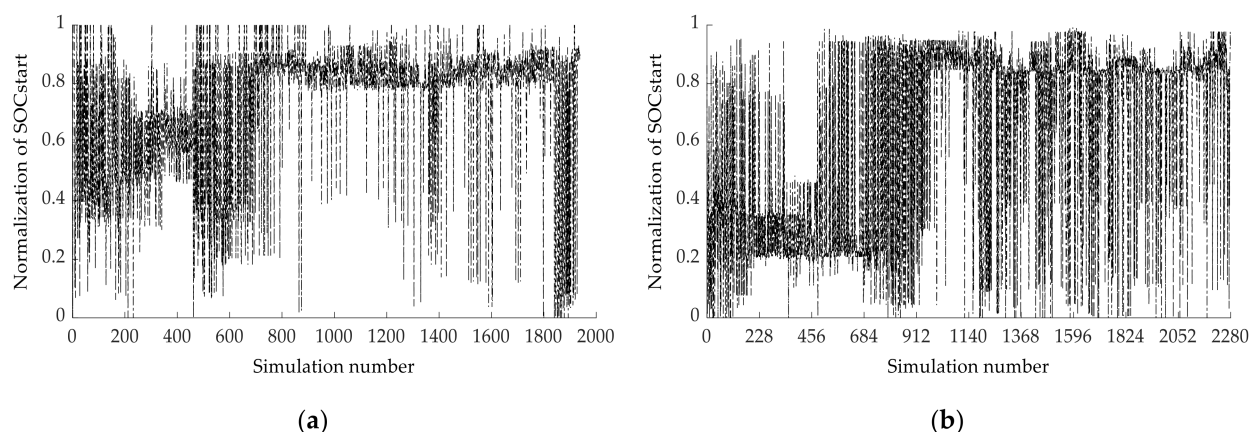


Figure 17. Variation of control variable SOC_{start} . (a) VDPM displacement is 180 mL/r; (b) VDPM displacement is 125 mL/r.

The strategy of selecting ‘integer-level’ parameters was utilized to filter results, where the coupler tooth number can be matched and accumulator volume can be selected from the sample book. The optimized parameters and minimum energy consumption of different VDPM displacements are given in Table 5. The minimum fuel consumption of 4240 groups after ‘integer-level’ filtering is 2.085 kg while the parameters are (180, 2.73, 18, 40, 0.86). Therefore, optimized parameters are $D_{PM} = 180$ mL/r, $i_p = 2.73$, $p_1 = 18$ MPa, $V_0 = 40$ L, and $SOC_{start} = 0.86$.

Table 5. Minimum energy consumption of different VDPM displacements.

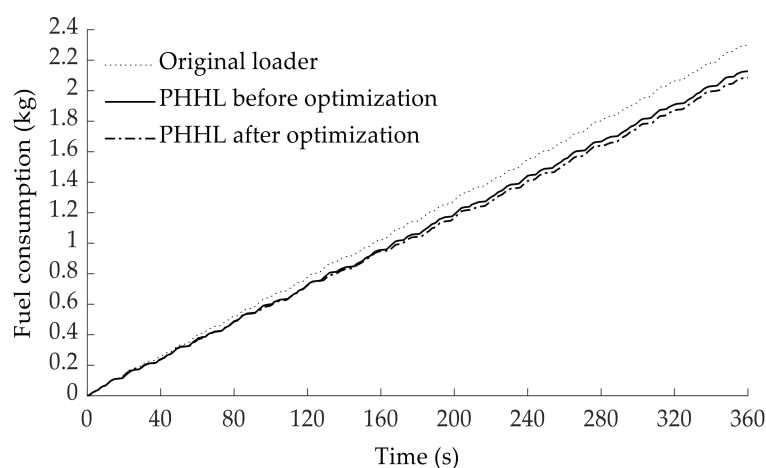
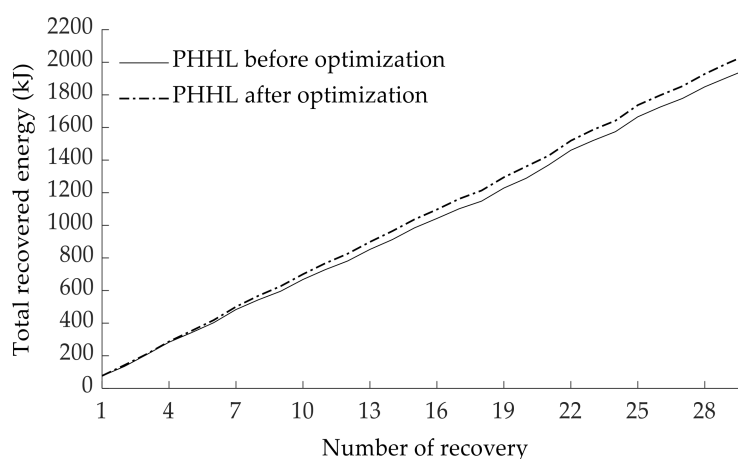
D_{PM} (mL/r)	i_p	p_1 (MPa)	V_0 (L)	SOC_{start}	Minimum Fuel Consumption (kg)
125	2.89	23	45	0.91	2.107
180	2.73	18	40	0.86	2.085

6.2. Comparison of Simulation Results before and after Optimization

Both the parameters acquired from the initial parameter matching and global optimization are simulated and verified on the platform of AMESim to confirm the optimization results of the ERS's parameters. The simulations are conducted under the V-type duty cycle including 10 cycles. Table 6 and Figures 18–20 exhibit the comparison consequences of the PHHL's fuel economy behavior before and after optimization, verifying the advantages of the PHHL after optimization.

Table 6. Simulation results.

Item	Fuel Consumption (kg)	Fuel Saving (%)	D_{PM} (mL/r)	i_p	V_0 (L)
Original loader	2.299	—	—	—	—
PHHL before optimization	2.128	7.44%	180	2.76	30
PHHL after optimization	2.085	9.31%	180	2.73	40

**Figure 18.** Fuel consumption of the original loader and PHHL.**Figure 19.** Total recovered energy.

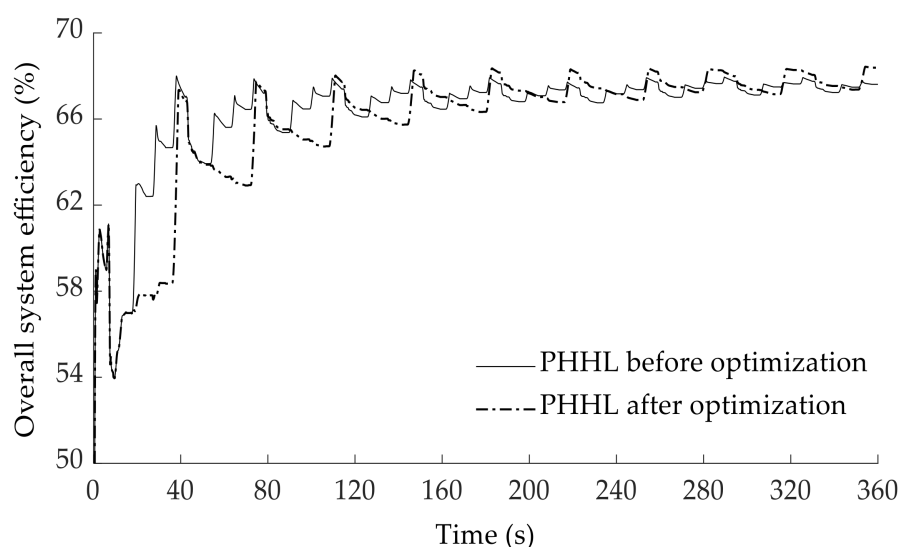


Figure 20. Overall transmission system efficiency.

The ERS with optimized parameters can reduce the fuel consumption by 25% at the end of 10 duty cycles in relation to that with initial parameters, of which the initial value of SOC_{start} is 0.5625 corresponding to the lowest working pressure. At the same time, compared with the conventional loader, the ERS of the PHHL after optimization led to a decrease of 9.31% in fuel consumption. As the parameters of the key components are relatively close, especially the parameter of the most expensive component VDPM. Therefore the difference in the cost of the ERS can be ignored, which means the fuel-saving determines the economic benefit. Obviously, the ERS optimized by AGA leads to a better economic benefit. Figure 18 shows the fuel consumption of the conventional loader, the PHHL before and after optimization. It can be observed that the fuel economy has improved significantly after optimization. Figures 19 and 20 show the performance improvement of the ERS after optimization. Among them, Figure 19 shows the total recovered energy calculated from the variation of SOC within 10 cycles. On the premise of the same initial state of the accumulator, the total recovered energy of the optimized ERS is higher than that of the ERS without optimization. In other words, the recovered brake energy is greater after optimization and gives rise to a better fuel economy. Figure 20 shows the difference between the ERS before and after optimization from the perspective of overall efficiency. To sum up, the overall efficiency is improved after optimization. For the engine and transmission system, the adoption of the optimized ERS increases the proportion of working time in high-efficiency range, which effectively reduces the fuel consumption.

Additionally, Figure 21 shows the SOC under the V-type duty cycle before and after optimization. The results indicate that the energy in the accumulator tends to be reused after multiple recoveries. This is related to the energy change during the accumulator discharge. As shown in Figure 22, the accumulator SOC continues to decrease for a period of time after receiving the stop command. This phenomenon is caused by the fact that the displacement of VDPM cannot be returned to zero immediately, and the degree of SOC reduction will increase with the increase of the VDPM response time.

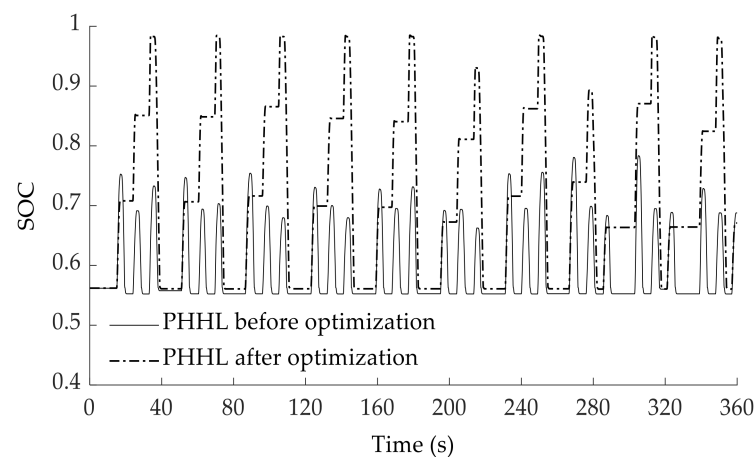


Figure 21. Accumulator SOC of the PHHL before and after optimization.

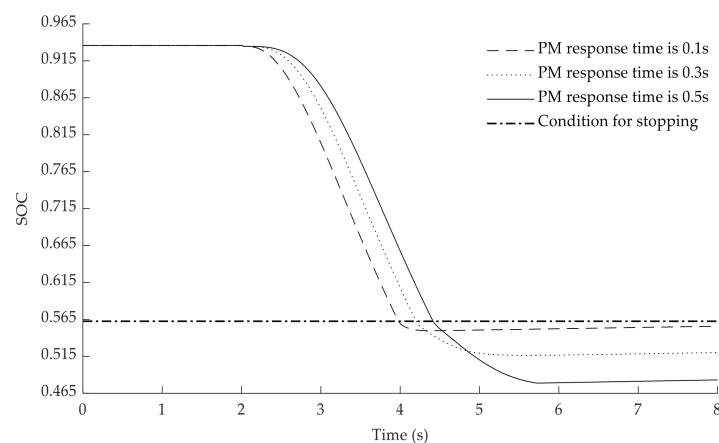


Figure 22. Variation of SOC with different VDPM response times.

6.3. Experiment

In check to see the effectiveness of the ERS, the energy consumption experiment was carried out. The block diagram of the experimental setup is shown in Figure 23 which is a control system based on CAN (Controller Area Network) bus network, the data of sensors are processed by the vehicle controller and sent to the human-machine interface through the CAN bus. Furthermore, the human-machine interface receives and records the experimental data from the vehicle controller and the engine ECU (Electronic Control Unit). It should be noted that the fuel consumption is calculated by the fuel consumption rate sent by the engine ECU and its sampling frequency is 100 Hz. A test cycle includes 10 V-type duty cycles that are continuously completed by the loader or the PHHL. Three test cycles are regarded as an effective experiment. To reduce the impact of manual operation and ensure the test accuracy, the experiment needs to ensure the fuel consumption difference of the loader or the PHHL between each two test cycles is less than 5%, otherwise, the number of the test will be increased. Figure 24 is the PHHL and test site. Due to the additional ERS, the operating weight of the PHHL is 17,800 kg. The parameters of ERS in the PHHL prototype are shown in Table 7.

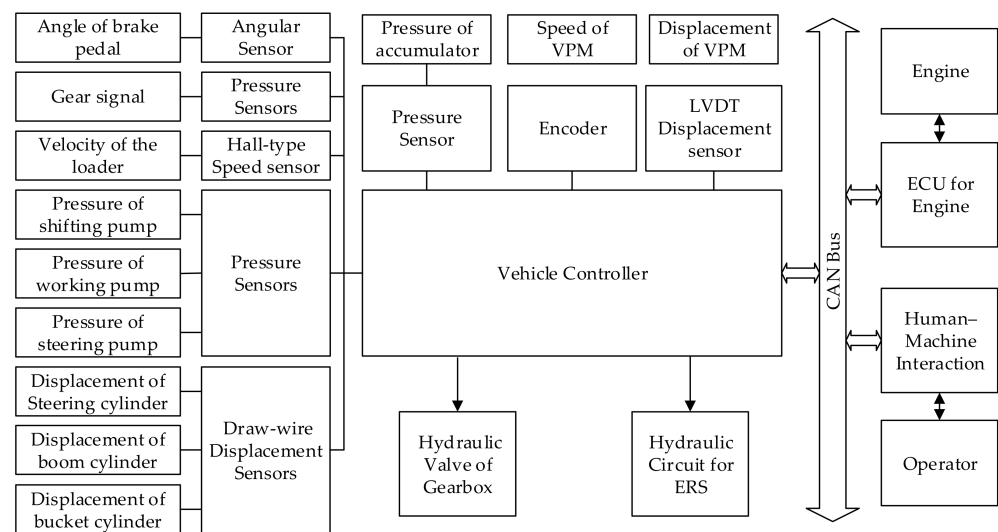


Figure 23. The block diagram of the experimental setup.



Figure 24. The test site and PHHL.

Table 7. The parameters of ERS in the PHHL prototype.

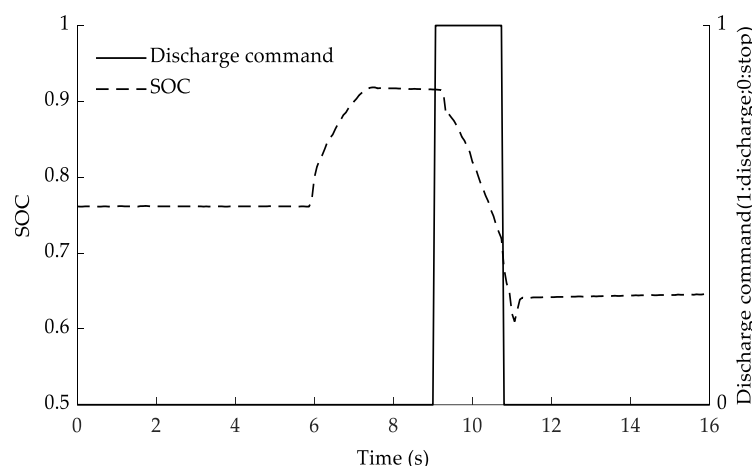
Parameter Description	Symbol	Value	Unit
VDPM displacement	D_{PM}	180	mL/r
Accumulator volume	V_0	40	L
Accumulator inflation pressure	p_0	12	MPa
Accumulator minimum working pressure	p_1	18	MPa
Coupler transmission ratio (positive/ negative)	i_c	3.73/2.73	—

Table 8 shows the fuel consumption experiment results of the original loader and the PHHL. The fuel consumption is close to the simulation, and the simulation error is within 2.5%. Compared with the hybrid-electric loader, the energy-saving effect is slightly weak [57–59]. However, considering the further improvement of the hybrid system and the PHHL has no limitation of supporting facilities such as charging piles, the energy-saving effect achieved by the regeneration of the brake energy is in accordance with the expectation. In addition, the combination of the electric hybrid and hydraulic hybrid is worth discussing.

Table 8. Fuel consumption experiment results.

Item	Fuel Consumption	Average Fuel Consumption	Fuel Saving	Simulation Err
Original loader	2.309 kg	2.3 kg	—	—
	2.254 kg			
	2.337 kg			
	2.048 kg			
PHHL	2.098 kg	2.091 kg	9.12%	2.04%
	2.126 kg			

Figure 25 indicates the change of SOC during the accumulator discharge. Significantly, during the regeneration process, the SOC still decreases after the stop command is issued, which is consistent with the simulation results. Therefore, the ERS reused the recovered energy after 2–3 times' accumulates. After numerous tests, the SOC_{start} is 0.88, which is close to the simulation result of 0.86. The existence of this limit SOC_{start} can appropriately reduce the frequency of energy regeneration, thereby reducing the frequency of energy waste caused by the end of discharge and preventing the overflow of the accumulator from causing the waste of brake energy during the recovery process at the same time, which can obtain a better fuel-saving effect.

**Figure 25.** Variation of SOC during discharging.

7. Conclusions

This paper developed an ERS for the PHHL and discussed the parameter matching method for optimal component sizing and energy management of ERS to get better fuel economy. The proposed configuration utilizes VDPM that operates both as a variable pump to transform brake energy into hydraulic to charge the accumulator and as a variable hydraulic motor to power the PHHL, thus assisting the engine during propelling. Then the methodical development of the ERS is recommended, which is divided into two steps: initial matching and optimization matching. The initial parameter matching determines the ranges of the design parameters and control parameters according to the vehicle dynamics and brake strength, then the subsequent optimal matching based on the AGA and the combined simulation platform of Python and AMESim can further improve the fuel economy. Besides, a dynamic rule-based control strategy was put forward to drive the powertrain properly. Both the conventional loader and the PHHL before and after optimization were comparatively analyzed under the V-type duty cycle by simulation. Finally, the fuel-saving of the optimized ERS was verified by an experiment between the original loader and a PHHL prototype. The key findings are summarized below:

- (1) The reduction of fuel consumption of the developed PHHL as compared to the original loader was acquired during the initial parameter matching of the ERS. Comparative

analysis indicated that the fuel consumption of the developed PHHL structure was decreased by 7.44% in relation to the original loader.

- (2) Based on the V-type driving cycle, the optimal VDPM displacement is 180 mL/r, accumulator volume is 40 L, minimum working pressure p_1 is 18 MPa, coupler transmission ratio i_p is 2.73, and the optimal control variable SOC_{start} is 0.86. Optimization results of the PHHL demonstrated that development in fuel economy by 25% is obtained compared to that of the PHHL before optimization, which proved the effectiveness of the improved optimization algorithm.
- (3) The fuel consumption of the PHHL prototype and original loader was investigated by repeating the V-type duty test three times, respectively, to survey the performances of the accumulator SOC and engine operation. Control variable SOC_{start} is 0.88 and fuel consumption are 2.091 kg (PHHL) and 2.30 kg (original loader), which are close to simulation results. The fuel-saving is 9.12% which is close to the energy-saving potential of brake energy (around 10%). In other words, the energy-saving effect of the proposed PHHL with optimized ERS approaches the upper limit of the energy-saving effect of the PHHL based on the brake energy regeneration without additional energy-saving technologies.

Author Contributions: The author's contributions are as follows: J.Y. and Y.B. were in charge of the whole trial; J.Y. wrote the manuscript; A.L., M.Y., and J.S. assisted with sampling and laboratory analyses. All authors have read and agreed to the published version of the manuscript.

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Data Availability Statement: The data presented in this study are available on request from the corresponding author. The data are not publicly available due to the supporting project has a confidentiality agreement.

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Conflicts of Interest: The authors declare no competing financial interests.

Acronyms

PHHL	Parallel hydraulic hybrid loader
HMTS	Hydrodynamic-mechanical transmission system
VDPM	Variable displacement pump/motor
PID	Proportional integral derivative
CAN	Controller area network
ERS	Energy regeneration system
AGA	Adaptive genetic algorithm
HTC	Hydraulic torque converter
SOC	State of charge
ECU	Electronic control unit

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