(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 4 560 148 A1

(12) **EUROPEAN PATENT APPLICATION**
published in accordance with Art. 153(4) EPC

(43) Date of publication:
28.05.2025 Bulletin 2025/22

(21) Application number: **24814098.0**

(22) Date of filing: **30.04.2024**

(51) International Patent Classification (IPC):
F15B 21/14 *(2006.01)*

(86) International application number:
PCT/CN2024/091025

(87) International publication number:
WO 2024/244893 (05.12.2024 Gazette 2024/49)

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC ME MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA
Designated Validation States:
GE KH MA MD TN

(30) Priority: **30.05.2023 CN 202310627461**

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(54) ENERGY RECOVERY AND CASCADE UTILIZATION CONTROL METHOD FOR FORKLIFT HYDRAULIC SYSTEM, AND CONTROL SYSTEM

(57) The present invention discloses a control method for cascade utilization of recovered energy in a forklift hydraulic system. The control method comprises the following steps: Step 1: Detecting a state of a fork of the forklift and, if the fork is in a lifting state, proceeding to Step 2; Step 2: Releasing high-pressure oil from different accumulators in the cascade energy storage section to supply energy to a lifting cylinder 3 of the fork; Step 3: Determining an output flow of a variable oil pump based on a lifting speed of the fork; Step 4: Determining a lifting power demand based on a load of the fork; Step 5: Determining an output power of the energy storage section based on a real-time output pressure value of the energy storage section; Step 6: Determining a minimum output power of the engine based on the lifting power demand and an output power of the accumulators; Step 7: Controlling an engine speed and a displacement of the variable oil pump. The present invention proposes a cascade energy utilization method for accumulators, determining an energy supply method of the accumulators based on the pressure of a rodless chamber and a supply pressure of each cascade accumulator, ensuring maximum energy supply efficiency; and determining the engine power output based on the load power demand and the energy supply power of the energy storage section, greatly reducing the excess power loss of the engine and improving energy utilization efficiency.

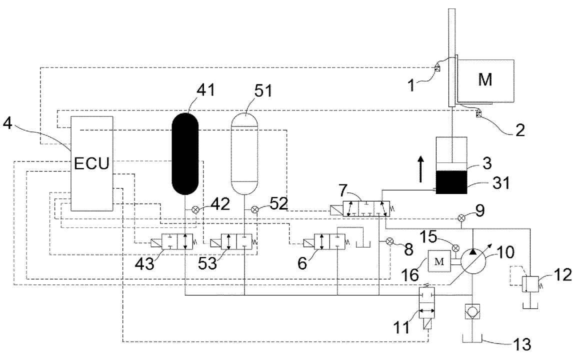

Fig. 1

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Description

Technical Field

[0001] The present invention relates to the field of forklift energy recovery and reuse technology, specifically a control method and control system for cascade utilization of recovered energy in a forklift hydraulic system.

Background

[0002] Counterbalanced forklifts, as commonly used cargo handling vehicles, are widely used in ports, freight yards, factory workshops, warehouses, and other locations, primarily for the distribution of goods. In recent years, influenced by national policies and the development of the logistics industry, forklift sales have surged, gradually becoming an indispensable part of industrial development. However, improving the energy utilization rate and work efficiency of forklifts has been a driving force for research. Currently, the main energy loss pathways in forklift lifting systems comprise: overflow valve overflow, directional valve throttling, pipeline bypass loss, and power mismatch loss. In the current era of energy conservation and emission reduction, any form of energy loss is considered wasteful.

[0003] During the lifting process of a forklift, throttling loss is mainly caused by the lack of power matching between the engine, hydraulic pump, and load, resulting in the engine and hydraulic pump outputting more energy than required by the lifting load. The engine often runs at low efficiency. Therefore, by controlling the speed of the hydraulic pump and reasonably outputting the energy required for lifting, the throttling loss at the directional valve ports can be effectively reduced.

Summary

[0004] The purpose of the present invention is to provide a control method and control system for cascade utilization of recovered energy in a forklift hydraulic system, aiming to solve the problems mentioned in the background.

[0005] To achieve the above purpose, the present invention provides the following technical solutions:

A control method for cascade utilization of recovered energy in a forklift hydraulic system, comprising an oil supply section and an energy storage section connected to a lifting cylinder, wherein the oil supply section comprises a variable oil pump connected to the lifting cylinder, and the energy storage section comprises at least one accumulator, wherein the control method comprises the following steps:

Step 1: Detecting a state of a fork of the forklift and, if the fork is in a lifting state, proceeding to Step 2;

Step 2: Releasing high-pressure oil from different accumulators in the cascade energy storage section to supply energy to the lifting cylinder of the fork;

Step 3: Determining an output flow of the variable oil pump based on a lifting speed of the fork;

Step 4: Determining a lifting power demand based on a load of the fork;

Step 5: Determining an output power of the energy storage section based on a real-time output pressure value of the energy storage section;

Step 6: Determining a minimum output power of the engine based on the lifting power demand and an output power of the accumulators;

Step 7: Controlling an engine speed and a displacement of the variable oil pump.

[0006] In a further solution of the present invention, the accumulators are connected to a controller, and the energy storage section comprises a first accumulator and a second accumulator. The control method for cascade release of high-pressure oil from all accumulators in the energy storage section in Step 2 comprises the following steps:

Step 2.1: Collecting an oil pressure of the first accumulator and the second accumulator, and an oil pressure at an outlet of the variable oil pump by the controller;

Step 2.2: Calculating a difference between an oil pressure of the first accumulator and the second accumulator and the oil pressure at the outlet of the variable oil pump;

Step 2.3: Selecting an accumulator with the smallest pressure difference that satisfies the condition of the pressure difference being greater than 0 for energy supply;

Step 2.4: Continuing to supply energy by the selected accumulator until SOC state is 0, then returning to Step 2.1.

[0007] In a further solution of the present invention, the output flow of the variable pump in Step 3 is:

$$Q_g = A_g \cdot v_g$$

[0008] Where, A_g is a cross-sectional area of the rodless chamber of the lifting cylinder, m^2 ; v_g is a speed of the lifting cylinder, m/s . The cross-sectional area A_g of the rodless chamber of the lifting cylinder is a fixed initial value of the system, and the speed v_g of the lifting cylinder is a preset value.

[0009] In a further solution of the present invention, the lifting power demand in Step 4 is:

$$P_l = F \cdot \frac{Q_g}{A_g} = p_g \cdot Q_g$$

[0010] Where, F is a force driving a load of the lifting cylinder, *N;* Q_g is a flow driving the lifting cylinder, ml/min; p_g is a pressure in the rodless chamber for driving the lifting cylinder, Mpa.

[0011] In a further solution of the present invention, the output power of the energy storage section in Step 5 is:

$$P_a = p_a \cdot Q_g$$

[0012] Where, p_a is the output pressure of the accumulator supplying oil, Mpa.

[0013] In a further solution of the present invention, the output power of the engine in Step 6 is:

$$P_e = P_l - P_a + P_s$$

[0014] Where, P_l is a load power demand, kW ; P_a is the output power of the energy storage section, kW ; P_s is a pipeline loss power, kW . The load power demand P_l and the output power P_a of the energy storage section are calculated as above, and P_s is an estimated value of the system.

[0015] In a further solution of the present invention, the engine speed control comprises the following steps:

Step 7.1.1: According to a target engine speed n_e based on engine universal characteristics and a real-time engine speed, the controller using an engine speed error e and an error change rate *ec* as inputs, and using Δk_p, Δk_i , Δk_d as outputs of a fuzzy controller, then adding these output values to initial values to obtain new parameters:

$$\begin{cases} k_p = k_{p0} + \Delta k_p \\ k_i = k_{i0} + \Delta k_i \\ k_d = k_{d0} + \Delta k_d \end{cases}$$

[0016] Where, k_{p0}, k_{i0}, k_{d0} are the initial values of the PID controller, k_p is a proportional gain of the controller, k_i is an integral gain of the controller, and k_d is a derivative gain of the controller.

[0017] Step 7.1.2: Designing linguistic variables for the engine speed error e, the error change rate *ec,* and an output throttle opening *to* as fuzzy subsets, [*BS MS S M SB MB B*], representing [Big-Small, Medium-Small, Small, Zero, Small-Big, Medium-Big, Big], respectively. A fuzzy universe of discourse for the speed error e and the error change rate *ec* is [-3,3], and a fuzzy universe of discourse for the output Δk_p, Δk_i is determined as [-6,6], and a fuzzy universe of discourse for Δk_d is determined as [-1,5].

[0018] Step 7.1.3: Adjusting control system parameters by the system according to quantization factors, the quantization factors being determined as:

$$K_e = n / e$$

$$K_{ec} = m / ec$$

[0019] Where, K_e, K_{ec} are the quantization factors for the error and the error change rate, respectively; *n, m* are universe of discourse levels.

[0020] Step 7.1.4: Determining fuzzy rules as follows:

When the engine speed error e is large, to improve the speed response of the system, a large Δk_p, should be selected; to prevent e from increasing excessively, a small Δk_d should be selected; and to avoid overshoot at the same time, Δk_i =0 should be selected.

[0021] When e is moderate, a small Δk_p, should be selected for the system to have a small overshoot, and appropriate Δk_i , Δk_d should be selected.

[0022] When e is small, to maintain system stability and avoid oscillation, a large Δk_p, Δk_i should be selected, and at the same time the value of Δk_d should be inversely proportional to |*ec*|.

[0023] Step 7.1.5: Obtaining a fuzzy relation matrix from a two-input single-output fuzzy controller as:

$$\begin{cases} R_p = \bigcup_{i,j}\left(E_i \times EC_j \times \Delta kp_{ij}\right) \\ R_i = \bigcup_{i,j}\left(E_i \times EC_j \times \Delta ki_{ij}\right) \\ R_p = \bigcup_{i,j}\left(E_i \times EC_j \times \Delta kd_{ij}\right) \end{cases}$$

[0024] Where, E_i is the i-th fuzzy state of *e*; EC_j is the *j*-th fuzzy state of EC_j; Δkp_{ij}, Δki_{ij} , Δkd_{ij} are the output fuzzy states under the *i*-th *e* and *j*-th *ec* fuzzy states.

[0025] Step 7.1.6: Defuzzifying the outputs by using the maximum membership method to take the maximum value in a fuzzy set and obtaining a throttle opening output, thereby controlling the engine speed through the throttle opening.

[0026] In a further solution of the present invention, the displacement of the variable oil pump is determined as follows:

$$q = \frac{Q_g}{n_e}$$

[0027] Where, n_e is the target engine speed.

[0028] The variable oil pump displacement controller is constructed as follows:

Step 7.2.1: determining a transfer function of a displacement of a variable cylinder of the variable oil pump to an opening of a solenoid valve of the variable oil pump according to structural characteristics of a swashplate variable oil pump as follows:

$$G_F(s) = \frac{x_t(s)}{x_L(s)} = \frac{\dfrac{K_q}{A_t}\left(1 + \dfrac{V_t}{4\beta_e K_c}s\right)}{\dfrac{mV_t}{4A_t^2\beta_e}s^3 + \dfrac{mK_c}{A_t}s^2 + s} = \frac{\dfrac{K_q}{A_t}\left(1 + \dfrac{V_t}{4\beta_e K_c}s\right)}{s(\dfrac{s^2}{\omega^2} + \dfrac{2\zeta}{\omega}s + 1)}$$

[0029] Where, $\omega = \sqrt{\dfrac{4A^2\beta_e}{mV_0}}$ is a natural frequency of the hydraulic system; $\zeta = \dfrac{K_c}{A_t}\sqrt{\dfrac{\beta_e m}{V_0}}$ is a damping ratio of the hydraulic system; K_q is a flow gain coefficient of the proportional valve, $m^3 / s \cdot pa$; m_t is a mass of the piston and the swashplate of the variable pump, *kg;* V_t is a volume of the variable cylinder, m^3 ; A_t is a force area of the variable cylinder, m^2 ; K_c is a pipeline leakage coefficient, m^3 / s ; β_e is a bulk modulus of the hydraulic oil, *pa* ; *s* is a complex variable; $x_L(s)$ is the displacement of the variable cylinder; $x_t(s)$ is the opening of the solenoid valve.

[0030] Step 7.2.2: Describing the transfer function in Step 1 with a mathematical model as follows:

$$\frac{y(k)}{u(k)}=\frac{\frac{\omega^2 K_q}{A_t}}{s^3+2\zeta\omega s^2+\omega^2 s}$$

[0031] Where, *y(k)* is an output at time *k* ; *u(k)* is an input at time *k* .

[0032] Step 7.2.3: Writing a non-minimal realization form of the system transfer function based on the Diophantine equation. Since the controlled system order $n = 3$, $m = 0$, the relative order $n^* = 3$, it follows that:

$$y(k)=\frac{1}{D(s)}[b_0 u(k)+\frac{R(s)}{Q(s)}u(k)+\frac{H(s)}{Q(s)}y(k)]$$

[0033] Where, $Q(s) = s^2 + g_1 s + g_0$; $D(s)= s^3 + d_2 s^2 + d_1 s + d_0$; $R(s) = r_1 s + r_0$;

$$H(s)=h_2 s^2+h_1 s+h_0 .$$

[0034] Step 7.2.4: Simplifying the non-minimal realization form of the system as follows:

$$y(k)=\Theta^T(k)\zeta(k)$$

[0035] Where, $\zeta(k)=\frac{1}{s^3+d_2 s^2+d_1 s+g_0}\xi(k)$; $\Theta^T(k) = (b_0\ r_1\ r_0\ h_2\ h_1\ h_0)$ are the parameters to be adjusted; b_0, r_1, h_2, h_1, h_0 are functions of time *k* ;

$$\xi^T(k)=\Big[u(k)\quad \frac{s}{s^2+g_1 s+g_0}u(k)\quad \frac{1}{s^2+g_1 s+g_0}u(k)\quad \frac{s^2}{s^2+g_1 s+g_0}y(k)\quad \frac{s}{s^2+g_1 s+g_0}y(k)\quad \frac{1}{s^2+g_1 s+g_0}y(k)\Big].$$

[0036] Step 7.2.5: Determining a system output estimate:

$$\hat{y}(k)=\hat{\Theta}^T(k-1)\zeta(k)$$

[0037] Where, $\hat{\Theta}^T(k-1)$ is an adaptive adjustment parameter at time *k* -1; *y(k)* is the system output estimate.

[0038] Step 7.2.6: Obtaining an adaptive rate as:

$$\hat{\Theta}(k)=\hat{\Theta}(k-1)-\frac{\Gamma(k-1)\zeta(k)e(k)}{1+\zeta^T(k)\Gamma(k-1)\zeta(k)}$$

$$\Gamma(k)=\Gamma(k-1)-\frac{\Gamma(k-1)\zeta(k)\zeta^T(k)\Gamma(k-1)}{1+\zeta^T(k)\Gamma(k-1)\zeta(k)}$$

[0039] Where, Γ is a positive definite symmetric matrix, *e(k) is* an error between the output and the target at time *k* .

[0040] A control system for cascade utilization of recovered energy in a forklift hydraulic system, comprising a lifting cylinder, wherein a piston end of the lifting cylinder is operatively connected to a fork, and the fork is equipped with a height sensor and a load sensor. The lifting cylinder is connected to an oil supply section and an oil return section through a lifting solenoid valve.

[0041] The oil supply section comprises an oil supply pipeline and a variable oil pump arranged on the oil supply pipeline. The oil supply pipeline is connected with an overflow valve and an oil return pressure sensor, and an oil supply valve is arranged at an end of the oil supply pipeline close to the variable oil pump.

[0042] The oil return section comprises an oil return pipeline, an end of which is connected to an oil inlet of the variable oil pump. The oil return pipeline is connected with an energy storage section, which comprises at least one accumulator, which is connected to the oil return pipeline through an accumulator valve.

[0043] The oil inlet of the variable oil pump is connected to an oil tank, and a check valve is arranged between the variable oil pump and the oil tank. The connection between the oil return pipeline and the variable oil pump is located between the check valve and the variable oil pump. The oil supply pipeline is connected with an oil supply pressure sensor and a safety valve.

[0044] A rodless chamber of the lifting cylinder is connected with a lifting solenoid valve, which is connected with the energy storage section and the variable oil pump. The variable oil pump is connected to the oil tank, and an oil outlet of the variable oil pump is connected with an oil supply pressure sensor and a safety valve. The variable oil pump is operatively connected to the engine, which is equipped with a speed sensor. A swash plate of the variable oil pump is connected with a variable cylinder, which is connected with a solenoid valve of the variable oil pump. An end of the variable cylinder is connected with a displacement sensor, and an amplifier is arranged between the displacement sensor and the solenoid valve of the variable oil pump.

[0045] In a further solution of the present invention, the energy storage section comprises a first accumulator and a second accumulator. The first accumulator is a high-pressure accumulator, and the second accumulator is a low-pressure accumulator. The first accumulator is connected with a first oil pressure sensor, and the first accumulator is connected with the oil supply pipeline through a first accumulator valve. The second accumulator is connected with a second oil pressure sensor, and the second accumulator is connected with the oil supply pipeline through a second accumulator valve.

[0046] Compared with the prior art, the beneficial effects of the present invention are:

1. The present invention proposes a cascade energy utilization method for accumulators, determining the energy supply method of the accumulators based on the pressure of the rodless chamber and the supply pressure of each cascade accumulator, ensuring maximum energy supply efficiency. At the same time, a power matching method between the engine, variable pump, accumulators, and load is proposed, determining the engine power output based on the load power demand and the energy supply power of the energy storage section, greatly reducing the excess power loss of the engine and improving energy utilization efficiency.

2. The present invention proposes a method for determining the flow rate of the variable pump based on the lifting speed requirements of the fork, ensuring the operational safety of the forklift while meeting work efficiency requirements, and avoiding the risk of goods falling and injuring workers during operation.

3. The present invention proposes a control method for controlling the engine speed and the displacement of the variable pump, effectively reducing energy waste in the forklift lifting system, reducing valve throttling loss, and improving work efficiency by adjusting the engine speed and the displacement of the variable pump.

Biref Description of the Drawings

[0047]

Figure 1 is a schematic diagram of the energy recovery system in this embodiment;
Figure 2 is a flowchart of the cascade utilization of the accumulators;
Figure 3 is a block diagram of the power matching control system;
Figure 4 is a structural diagram of the variable oil pump displacement regulating mechanism;

Listing of Reference Signs

[0048]

1 height sensor

2 load sensor

3 lifting cylinder

31 rodless chamber of the lifting cylinder

41 first accumulator

42 first oil pressure sensor

43 first accumulator valve

51 second accumulator

52 second oil pressure sensor

53 second accumulator valve

6 overflow valve

7 lifting solenoid valve

8 oil return pressure sensor

9 oil supply pressure sensor

10 oil pump

11 oil supply valve

12 safety valve

13 oil tank

14 controller

15 speed sensor

16 engine

17 solenoid valve of the variable oil pump

18 swash plate of the variable oil pump

19 variable cylinder

20 displacement sensor

21 amplifier

Detailed Description

[0049] The technical solutions in the embodiments of the present invention will be clearly and completely described below in conjunction with the accompanying drawings in the embodiments of the present invention. Obviously, the described embodiments are only a part of the embodiments of the present invention, rather than all of them. Based on the embodiments of the present invention, all other embodiments obtained by those of ordinary skill in the art without creative work shall fall within the protection scope of the present invention.

[0050] Referring to Figures 1-4, a power matching and cascade energy utilization system for a forklift hydraulic system is provided, comprising a lifting cylinder 3, which is connected to an oil supply section and an oil return section through a lifting solenoid valve 7.

[0051] The oil supply section comprises an oil supply pipeline and a variable oil pump 10 arranged on the oil supply pipeline.

[0052] The oil return section comprises an oil return pipeline, an end of which is connected to an oil inlet of the variable oil pump 10. The oil return pipeline is connected with an energy storage section. The energy storage section comprises at least one accumulator, which is connected to the oil return pipeline through an accumulator valve.

[0053] A rodless chamber 31 of the lifting cylinder 3 is connected with the lifting solenoid valve 7, and a piston end of the lifting cylinder 3 is connected with a fork. The fork is equipped with a height sensor 1 and a load sensor 2. The height sensor is used to detect the height of the fork, and the load sensor is used to detect the load signal.

[0054] An oil inlet of the variable oil pump 10 is connected to an oil tank 13, and a check valve is arranged between the variable oil pump 10 and the oil tank 13. The connection between the oil return pipeline and the variable oil pump 10 is located between the check valve and the variable oil pump 10. The oil supply pipeline is connected with an oil supply pressure sensor 9 and a safety valve 12.

[0055] The energy storage section comprises a first accumulator 41 and a second accumulator 51. The first accumulator 41 is a high-pressure accumulator, and the second accumulator 51 is a low-pressure accumulator. The first accumulator 41 is connected with a first oil pressure sensor 42, and the first accumulator 41 is connected with the oil supply pipeline through a first accumulator valve 43. The second accumulator 51 is connected with a second oil pressure sensor 52, and the second accumulator 51 is connected with the oil supply pipeline through a second accumulator valve 53.

[0056] The oil supply pipeline is connected with an overflow valve 6 and an oil return pressure sensor 8, and an oil supply valve 11 is arranged at an end of the oil supply pipeline close to the variable oil pump 10.

[0057] The variable oil pump 10 is connected to the engine 16 through a transmission shaft, and the transmission shaft is connected with a speed sensor 15. The swash plate 18 of the variable oil pump is connected with a variable cylinder 19, which is connected with a solenoid valve of the variable oil pump 17. An end of the variable cylinder 19 is connected with a displacement sensor 20, and an amplifier 21 is arranged between the displacement sensor 20 and the solenoid valve of the variable oil pump 17.

[0058] A control method for cascade utilization of recovered energy in a forklift hydraulic system, comprising the following steps:

Step 1: Detecting a state of a fork of the forklift and, if the fork is in a lifting state, proceeding to Step 2;

Step 2: Releasing high-pressure oil from different accumulators in the cascade energy storage section to supply energy to the lifting cylinder 3 of the fork. During this process, a controller 14 selects an accumulator for energy supply based on the pressure of the rodless chamber of the lifting cylinder and the pressure state of each accumulator to supply energy to the oil inlet of the variable oil pump;

[0059] The control method for cascade release of high-pressure oil from all accumulators in the energy storage section comprises the following steps:

Step 2.1: Collecting the oil pressure of the first accumulator 41, the second accumulator 51, and an oil pressure at an outlet of the variable oil pump 10 by the controller 14. In this embodiment, the controller 14 collects signals from the first oil pressure sensor 42, the second oil pressure sensor 52, and the oil supply pressure sensor 9;

Step 2.2: Calculating a difference between an oil pressure of the first accumulator 41 and the second accumulator 51 and the oil pressure at the outlet of the variable oil pump 10;

Step 2.3: Selecting an accumulator with the smallest pressure difference that satisfies the condition of the pressure difference being greater than 0 for energy supply;

Step 2.4: Continuing to supply energy by the selected accumulator until SOC state is 0, then returning to Step 2.1;

Step 3: Determining an output flow of the variable oil pump based on a lifting speed of the fork;

[0060] The output flow of the variable pump is determined as follows:

$$Q_g = A_g \cdot v_g$$

[0061] Where, A_g is a cross-sectional area of the rodless chamber of the lifting cylinder, m^2; v_g is a speed of the lifting cylinder, *m/s*. The cross-sectional area A_g of the rodless chamber of the lifting cylinder is a fixed initial value of the system, and the speed v_g of the lifting cylinder is a preset value.

Step 4: Determining a lifting power demand based on a load of the fork;

[0062] The lifting power demand is determined as follows:

$$P_l = F \cdot \frac{Q_g}{A_g} = p_g \cdot Q_g$$

[0063] Where, F is a force driving a load of the lifting cylinder, *N* ; Q_g is a flow driving the lifting cylinder, ml/min ; p_g is a pressure in the rodless chamber for driving the lifting cylinder, Mpa. The load force F is collected by the load sensor (2), and the pressure p_g in the rodless chamber of the lifting cylinder is collected by the oil supply pressure sensor (9).

Step 5: Determining an output power of the energy storage section based on a real-time output pressure value of the energy storage section;

[0064] The output power of the energy storage section is determined as follows:

$$P_a = p_a \cdot Q_g$$

[0065] Where, p_a is the output pressure of the accumulator supplying oil, Mpa. The value is collected by the oil return pressure sensor (8).

Step 6: Determining a minimum output power of the engine based on the lifting power demand and an output power of the accumulators.

[0066] The output power of the engine is determined as follows:

$$P_e = P_l - P_a + P_s$$

[0067] Where, P_l is a load power demand, kW ; P_a is the output power of the energy storage section, kW ; P_s is a pipeline loss power, kW. The load power demand P and the output power P_a of the energy storage section are calculated as above, and P_s is an estimated value of the system.

[0068] Step 7: Controlling an engine speed and a displacement of the variable oil pump. Specifically, the speed sensor 15 collects the real-time engine speed signal and transmits it to the controller 14. An adaptive fuzzy PID control is used to construct an engine speed controller, and a model reference adaptive control is used to construct a variable oil pump displacement controller.

[0069] The engine speed controller is constructed as follows:

Step 7.1.1: According to a target engine speed based on engine universal characteristics and a real-time speed signal from the speed sensor, the controller using an engine speed error e and an error change rate *ec* as inputs, and using Δk_p, Δk_i, Δk_d as outputs of a fuzzy controller, then adding these output values to initial values to obtain new parameters:

$$\begin{cases} k_p = k_{p0} + \Delta k_p \\ k_i = k_{i0} + \Delta k_i \\ k_d = k_{d0} + \Delta k_d \end{cases}$$

[0070] Where, k_{p0}, k_{i0}, k_{d0} are the initial values of the PID controller, k_p is a proportional gain of the controller, k_i is an integral gain of the controller, and k_d is a derivative gain of the controller.

[0071] Step 7.1.2: Designing linguistic variables for the engine speed error e, the error change rate *ec,* and an output throttle opening *to* as fuzzy subsets, [*BS MS S M SB MB B*], representing [Big-Small, Medium-Small, Small, Zero, Small-Big, Medium-Big, Big], respectively. A fuzzy universe of discourse for the speed error e and the error change rate *ec* is [-3,3], and a fuzzy universe of discourse for the output Δk_p, Δk_i is determined as [-6,6], and a fuzzy universe of discourse for Δk_d is determined as [-1,5].

[0072] Step 7.1.3: Adjusting control system parameters by the system according to quantization factors, the quantization factors being determined as:

$$K_e = n / e$$
$$K_{ec} = m / ec$$

[0073] Where, K_e, K_{ec} are the quantization factors for the error and the error change rate, respectively; *n, m* are universe of discourse levels.

[0074] Step 7.1.4: Determining fuzzy rules as follows:
When the engine speed error e is large, to improve the speed response of the system, a large Δk_p, should be selected; to prevent e from increasing excessively, a small Δk_d should be selected; and to avoid overshoot at the same time, Δk_i =0 should be selected.

[0075] When e is moderate, a small Δk_p, should be selected for the system to have a small overshoot, and appropriate Δk_i , Δk_d should be selected.

[0076] When e is small, to maintain system stability and avoid oscillation, a large Δk_p, Δk_i should be selected, and at the same time the value of Δk_d should be inversely proportional to |*ec*|.

[0077] Step 7.1.5: Obtaining a fuzzy relation matrix from a two-input single-output fuzzy controller as:

$$\begin{cases} R_p = \bigcup_{i,j} \left(E_i \times EC_j \times \Delta kp_{ij} \right) \\ R_i = \bigcup_{i,j} \left(E_i \times EC_j \times \Delta ki_{ij} \right) \\ R_p = \bigcup_{i,j} \left(E_i \times EC_j \times \Delta kd_{ij} \right) \end{cases}$$

[0078] Where, E_i is the i-th fuzzy state of *e;* EC_j is the *j*-th fuzzy state of *ec;* Δkp_{ij}, Δki_{ij}, Δkd_{ij} are the output fuzzy states under the *i*-th *e* and *j*-th *ec* fuzzy states.

[0079] Step 7.1.6: Defuzzifying the outputs by using the maximum membership method to take the maximum value in a fuzzy set and obtaining a throttle opening output.

[0080] The displacement of the variable oil pump is determined as follows:

$$q = \frac{Q_g}{n_e}$$

[0081] Where, n_e is the target engine speed.

[0082] The variable oil pump displacement controller is constructed as follows:
Step 7.2.1: Determining a transfer function of a displacement of a variable cylinder 19 of the variable oil pump to an opening of a solenoid valve 17 of the variable oil pump according to structural characteristics of a swashplate variable oil pump as follows:

$$G_F(s) = \frac{x_t(s)}{x_L(s)} = \frac{\frac{K_q}{A_t}\left(1 + \frac{V_t}{4\beta_e K_c} s\right)}{\frac{mV_t}{4A_t^2\beta_e} s^3 + \frac{mK_c}{A_t} s^2 + s} = \frac{\frac{K_q}{A_t}\left(1 + \frac{V_t}{4\beta_e K_c} s\right)}{s(\frac{s^2}{\omega^2} + \frac{2\zeta}{\omega} s + 1)}$$

[0083] Where, $\omega = \sqrt{\frac{4A^2\beta_e}{mV_0}}$ is a natural frequency of the hydraulic system; $\zeta = \frac{K_c}{A_t}\sqrt{\frac{\beta_e m}{V_0}}$ is a damping ratio of the hydraulic system; K_q is a flow gain coefficient of the proportional valve, $m^3 / s \cdot pa$; m_t is a mass of the piston and the swashplate of the variable pump, *kg;* V_t is a volume of the variable cylinder, m^3; A_t is a force area of the variable cylinder, m^2 ; K_c is a pipeline leakage coefficient, m^3 / s; β_e is a bulk modulus of the hydraulic oil, *pa*; *s* is a complex variable; $x_L(s)$ is the displacement of the variable cylinder; $x_t(s)$ is the opening of the solenoid valve of the variable oil pump.

[0084] Step 7.2.2: Describing the transfer function in Step 1 with a mathematical model as follows:

$$\frac{y(k)}{u(k)}=\frac{\frac{\omega^2 K_q}{A_t}}{s^3+2\zeta\omega s^2+\omega^2 s}$$

[0085] Where, *y(k)* is an output at time *k* ; *u*(*k*) is an input at time *k.*

[0086] Step 7.2.3: Writing a non-minimal realization form of the system transfer function based on the Diophantine equation. Since the controlled system order *n* = 3, *m* = 0, the relative order *n** = 3, it follows that:

$$y(k)=\frac{1}{D(s)}[b_0u(k)+\frac{R(s)}{Q(s)}u(k)+\frac{H(s)}{Q(s)}y(k)]$$

[0087] Where, $Q(s) = s^2 + g_1s + g_0$; $D(s) = s^3 + d_2s^2 + d_1s + d_0$; $R(s) = r_1s + r_0$;

$$H(s)=h_2s^2+h_1s+h_0 .$$

[0088] Step 7.2.4: Simplifying the non-minimal realization form of the system as follows:

$$y(k)=\Theta^T(k)\zeta(k)$$

[0089] Where, $\zeta(k)=\frac{1}{s^3+d_2s^2+d_1s+g_0}\xi(k)$; $\Theta^T(k) = (b_0\ r_1\ r_0\ h_2\ h_1\ h_0)$ are the parameters to be adjusted; b_0, r_1, h_2, h_1, h_0 are functions of time *k ;*

$$\xi^T(k)=\left[u(k)\frac{s}{s^2+g_1s+g_0}u(k)\frac{1}{s^2+g_1s+g_0}u(k)\right.$$
$$\left.\frac{s^2}{s^2+g_1s+g_0}y(k)\frac{s}{s^2+g_1s+g_0}y(k)\frac{1}{s^2+g_1s+g_0}y(k)\right]$$

[0090] Step 7.2.5: Determining a system output estimate:

$$\hat{y}(k)=\hat{\Theta}^T(k-1)\zeta(k)$$

[0091] Where, $\hat{\Theta}^T(k\text{-}1)$ is an adaptive adjustment parameter at time *k* -1; *y(k)* is the system output estimate.

[0092] Step 7.2.6: Obtaining an adaptive rate as:

$$\hat{\Theta}(k)=\hat{\Theta}(k-1)-\frac{\Gamma(k-1)\zeta(k)e(k)}{1+\zeta^T(k)\Gamma(k-1)\zeta(k)}$$

$$\Gamma(k)=\Gamma(k-1)-\frac{\Gamma(k-1)\zeta(k)\zeta^T(k)\Gamma(k-1)}{1+\zeta^T(k)\Gamma(k-1)\zeta(k)}$$

[0093] Where, Γ is a positive definite symmetric matrix, *e*(*k*) is an error between the output and the target at time *k.*

[0094] Through the aforementioned control method, the error between the output value and the target value can be obtained. Based on the requirements of the actual target value, the output of the variable pump can be controlled, thereby enabling precise control of the variable pump and reducing its energy losses.

Embodiment

[0095] Taking a standard 3-ton forklift as an example:
When applying the energy cascade utilization system of this embodiment to a standard 3-ton forklift, the measured parameters of the forklift are as follows: the forklift has a rated load capacity of 3 tons, a maximum lifting height of 3 meters, a maximum stroke of the lifting cylinder of 1.5 meters, a cross-sectional area of the rodless chamber of the lifting cylinder of 0.0024 m^2, and a hydraulic transmission system. The selected parameters for the first accumulator in the cascade recovery are 8 L and 16 MPa, and for the second accumulator, 10 L and 8 MPa.

[0096] Taking a fork load of 2.5-ton as an example, when the fork is lifted, the pressure in the rodless chamber of the lifting cylinder is measured to be 24 Mpa, and the pressures of the high-pressure and low-pressure bladder accumulators are 20 Mpa and 15 Mpa, respectively. According to the high-pressure and low-pressure sequence, the high-pressure accumulator is first selected for hydraulic energy recovery. The required speed of the lifting cylinder is 0.25 m/s, and the output flow of the variable pump is determined to be 60 L/min. Under the pressure supply of the accumulator, the initial pressure difference between the inlet and outlet of the variable pump is 4 Mpa, the load power demand is 25 kW, the output power of the accumulators is 20 kW, and the power loss is 1 kW. Therefore, the minimum output power of the engine is 6 kW, reducing the engine output power by 20 kW.

[0097] During the lifting process, the output pressure of the high-pressure accumulator continuously decreases, and the pressure difference between the inlet and outlet of the variable oil pump continuously increases until the pressure of the high-pressure accumulator drops to 16 Mpa, at which point the system switches to the low-pressure accumulator for energy supply. When the pressure of the low-pressure accumulator drops to 8 Mpa, the oil supply valve is closed, and the variable oil pump directly draws oil from the oil tank.

[0098] For those skilled in the art, it is clear that the present invention is not limited to the details of the exemplary embodiments described above, and the invention can be implemented in other specific forms without departing from its spirit or essential characteristics. Therefore, the embodiments should be regarded as illustrative and non-restrictive in all aspects, and the scope of the invention is defined by the appended claims rather than the foregoing description. It is intended to encompass all modifications that fall within the meaning and scope of the equivalent elements of the claims. No reference numerals in the claims should be construed as limiting the scope of the claims.

[0099] Furthermore, it should be understood that although this specification is described in terms of implementations, each implementation does not necessarily contain only a single independent technical solution. The narrative style of the specification is solely for the sake of clarity. Skilled artisans should consider the specification as a whole, and the technical solutions in the various embodiments can also be appropriately combined to form other implementations that are understandable to those skilled in the art.

Claims

1. A control method for cascade utilization of recovered energy in a forklift hydraulic system, **characterized by** comprising an oil supply section and an energy storage section connected to a lifting cylinder (3), wherein the oil supply section comprises a variable oil pump (10) connected to the lifting cylinder (3), and the energy storage section comprises at least one accumulator, wherein the control method comprises the following steps:

Step 1: detecting a state of a fork of the forklift and, if the fork is in a lifting state, proceeding to Step 2;
Step 2: releasing high-pressure oil from different accumulators in the cascade energy storage section to supply energy to the lifting cylinder (3) of the fork;
Step 3: determining an output flow of the variable oil pump based on a lifting speed of the fork;
Step 4: determining a lifting power demand based on a load of the fork;
Step 5: determining an output power of the energy storage section based on a real-time output pressure value of the energy storage section;
Step 6: determining a minimum output power of the engine based on the lifting power demand and an output power of the accumulators;
Step 7: controlling an engine speed and a displacement of the variable oil pump.

2. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 1, **characterized in that** the accumulators are connected to a controller (14), and the energy storage section comprises a first accumulator (41) and a second accumulator (51), wherein the control method for cascade release of high-pressure oil from all accumulators in the energy storage section in Step 2 comprises the following steps:

Step 2.1: collecting an oil pressure of the first accumulator (41), the second accumulator (51), and an oil pressure

at an outlet of the variable oil pump (10) by the controller (14);
Step 2.2: calculating a difference between an oil pressure of the first accumulator (41) and the second accumulator (51) and the oil pressure at the outlet of the variable oil pump (10);
Step 2.3: selecting an accumulator with the smallest pressure difference that satisfies the condition of the pressure difference being greater than 0 for energy supply;
Step 2.4: continuing to supply energy by the selected accumulator until SOC state is 0, then returning to Step 2.1.

3. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 1, **characterized in that** the output flow of the variable pump in Step 3 is:

$$Q_g = A_g \cdot v_g$$

where, A_g is a cross-sectional area of the rodless chamber of the lifting cylinder, m^2; v_g is a speed of the lifting cylinder, m/s, wherein the cross-sectional area A_g of the rodless chamber of the lifting cylinder is a fixed initial value of the system, and the speed v_g of the lifting cylinder is a preset value.

4. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 3, **characterized in that** the lifting power demand in Step 4 is:

$$P_l = F \cdot \frac{Q_g}{A_g} = p_g \cdot Q_g$$

where, F is a force driving a load of the lifting cylinder, N; Q_g is a flow driving the lifting cylinder, ml/min; p_g is a pressure in the rodless chamber for driving the lifting cylinder, Mpa.

5. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 4, **characterized in that** the output power of the energy storage section in Step 5 is:

$$P_a = p_a \cdot Q_g$$

where, p_a is the output pressure of the accumulator supplying oil, Mpa.

6. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 5, **characterized in that** the output power of the engine in Step 6 is:

$$P_e = P_l - P_a + P_s$$

where, P_l is a load power demand, kW; P_a is the output power of the energy storage section, kW; P_s is a pipeline loss power, kW, wherein the load power demand P_l and the output power P_a of the energy storage section are calculated as above, and P_s is an estimated value of the system.

7. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 1, **characterized in that** the engine speed control comprises the following steps:

Step 7.1.1: according to a target engine speed n_e based on engine universal characteristics and a real-time engine speed, the controller using an engine speed error e and an error change rate ec as inputs, and using Δk_p, Δk_i, Δk_d as outputs of a fuzzy controller, then adding these output values to initial values to obtain new parameters:

$$\begin{cases} k_p = k_{p0} + \Delta k_p \\ k_i = k_{i0} + \Delta k_i \\ k_d = k_{d0} + \Delta k_d \end{cases}$$

where, k_{p0}, k_{i0}, k_{d0} are the initial values of the PID controller, k_p is a proportional gain of the controller, k_i is an integral gain of the controller, and k_d is a derivative gain of the controller;
Step 7.1.2: designing linguistic variables for the engine speed error e, the error change rate *ec,* and an output throttle opening *to* as fuzzy subsets, [*BS MS S M SB MB B*], representing [Big-Small, Medium-Small, Small, Zero, Small-Big, Medium-Big, Big], respectively, wherein a fuzzy universe of discourse for the speed error e and the error change rate *ec* is [-3,3], and a fuzzy universe of discourse for the output Δk_p, Δk_i is determined as [-6,6], and a fuzzy universe of discourse for Δk_d is determined as [-1,5];
Step 7.1.3: adjusting control system parameters by the system according to quantization factors,
the quantization factors being determined as:

$$K_e = n / e$$

$$K_{ec} = m / ec$$

where, K_e, K_{ec} are the quantization factors for the error and the error change rate, respectively; *n, m* are universe of discourse levels;
Step 7.1.4: determining fuzzy rules as follows:

when the engine speed error e is large, to improve the speed response of the system, a large Δk_p, should be selected; to prevent e from increasing excessively, a small Δk_d should be selected; and to avoid overshoot at the same time, Δk_i =0 should be selected;
when e is moderate, a small Δk_p, should be selected for the system to have a small overshoot, and appropriate Δk_i , Δk_d should be selected;
when e is small, to maintain system stability and avoid oscillation, a large Δk_p, Δk_i should be selected, and at the same time the value of Δk_d should be inversely proportional to |*ec*|;

Step 7.1.5: obtaining a fuzzy relation matrix from a two-input single-output fuzzy controller as:

$$\begin{cases} R_p = \bigcup_{i,j} \left(E_i \times EC_j \times \Delta kp_{ij} \right) \\ R_i = \bigcup_{i,j} \left(E_i \times EC_j \times \Delta ki_{ij} \right) \\ R_p = \bigcup_{i,j} \left(E_i \times EC_j \times \Delta kd_{ij} \right) \end{cases}$$

where, E_i is the i-th fuzzy state *of e; EC_j* is the *j*-th fuzzy state of *ec;* Δkp_{ij}, Δki_{ij}, Δkd_{ij} are the output fuzzy states under the *i*-th *e* and *j*-th *ec* fuzzy states;
Step 7.1.6: defuzzifying the outputs by using the maximum membership method to take the maximum value in a fuzzy set and obtaining a throttle opening output, thereby controlling the engine speed through the throttle opening.

8. The control method for cascade utilization of recovered energy in a forklift hydraulic system according to claim 1, **characterized in that** the displacement of the variable oil pump is determined as follows:

$$q = \frac{Q_g}{n_e}$$

where, n_e is the target engine speed;
the variable oil pump displacement controller is constructed as follows:

Step 7.2.1: determining a transfer function of a displacement of a variable cylinder (19) of the variable oil pump to an opening of a solenoid valve (17) of the variable oil pump according to structural characteristics of a swashplate variable oil pump as follows:

$$G_F(s)=\frac{x_t(s)}{x_L(s)}=\frac{\frac{K_q}{A_t}\left(1+\frac{V_t}{4\beta_e K_c}s\right)}{\frac{mV_t}{4A_t^2\beta_e}s^3+\frac{mK_c}{A_t}s^2+s}=\frac{\frac{K_q}{A_t}\left(1+\frac{V_t}{4\beta_e K_c}s\right)}{s(\frac{s^2}{\omega^2}+\frac{2\zeta}{\omega}s+1)}$$

where, $\omega=\sqrt{\frac{4A^2\beta_e}{mV_0}}$ is a natural frequency of the hydraulic system; $\zeta=\frac{K_c}{A_t}\sqrt{\frac{\beta_e m}{V_0}}$ is a damping ratio of the hydraulic system; K_q is a flow gain coefficient of the proportional valve, $m^3 / s \cdot pa$; m_t is a mass of the piston and the swashplate of the variable pump, kg; V_t is a volume of the variable cylinder, m^3; A_t is a force area of the variable cylinder, m^2 ; K_c is a pipeline leakage coefficient, m^3 / s; β_e is a bulk modulus of the hydraulic oil, pa ; s is a complex variable; $x_L(s)$ is the displacement of the variable cylinder; $x_t(s)$ is the opening of the solenoid valve;

Step 7.2.2: describing the transfer function in Step 1 with a mathematical model as follows:

$$\frac{y(k)}{u(k)}=\frac{\frac{\omega^2 K_q}{A_t}}{s^3+2\zeta\omega s^2+\omega^2 s}$$

where, $y(k)$ is an output at time k ; $u(k)$ is an input at time k ;

Step 7.2.3: writing a non-minimal realization form of the system transfer function based on the diophantine equation, wherein since the controlled system order n = 3, m = 0, the relative order n^* = 3, it follows that:

$$y(k)=\frac{1}{D(s)}[b_0u(k)+\frac{R(s)}{Q(s)}u(k)+\frac{H(s)}{Q(s)}y(k)]$$

where, $Q(s) = s^2 + g_1s + g_0$; $D(s) = s^3 + d_2s^2 + d_1s + d_0$; $R(s) = r_1s + r_0$;

$$H(s)=h_2s^2+h_1s+h_0\,;$$

Step 7.2.4: simplifying the non-minimal realization form of the system as follows:

$$y(k)=\Theta^T(k)\zeta(k)$$

where, $\zeta(k)=\frac{1}{s^3+d_2s^2+d_1s+g_0}\xi(k)$; $\Theta^T(k) = (b_0, r_1\ r_0\ h_2\ h_1\ h_0)$ are the parameters to be adjusted; b_0, r_1, h_2, h_1, h_0 are functions of time k ;

$$\xi^T(k)=\left[u(k)\quad \frac{s}{s^2+g_1s+g_0}u(k)\quad \frac{1}{s^2+g_1s+g_0}u(k)\quad \frac{s^2}{s^2+g_1s+g_0}y(k)\quad \frac{s}{s^2+g_1s+g_0}y(k)\quad \frac{1}{s^2+g_1s+g_0}y(k)\right];$$

Step 7.2.5: determining a system output estimate:

$$\hat{y}(k) = \hat{\Theta}^T (k-1)\zeta(k)$$

where, $\hat{\Theta}^T(k-1)$ is an adaptive adjustment parameter at time k -1; $y(k)$ is the system output estimate;
Step 7.2.6: obtaining an adaptive rate as:

$$\hat{\Theta}(k) = \hat{\Theta}(k-1) - \frac{\Gamma(k-1)\zeta(k)e(k)}{1+\zeta^T(k)\Gamma(k-1)\zeta(k)}$$

$$\Gamma(k) = \Gamma(k-1) - \frac{\Gamma(k-1)\zeta(k)\zeta^T(k)\Gamma(k-1)}{1+\zeta^T(k)\Gamma(k-1)\zeta(k)}$$

where, Γ is a positive definite symmetric matrix, $e(k)$ is an error between the output and the target at time k .

9. A control system using the control method for cascade utilization of recovered energy in a forklift hydraulic system according to any one of claims 1-8, **characterized by** comprising a lifting cylinder (3), a piston end of the lifting cylinder (3) is operatively connected to a fork, and the fork is equipped with a height sensor (1) and a load sensor (2), wherein the lifting cylinder (3) is connected to an oil supply section and an oil return section through a lifting solenoid valve (7);

the oil supply section comprises an oil supply pipeline and a variable oil pump (10) installed in the oil supply pipeline, wherein the oil supply pipeline is connected to an overflow valve (6) and an oil return pressure sensor (8), wherein an end of the oil supply pipeline close to the variable oil pump (10) is equipped with an oil supply valve (11);
the oil return section comprises an oil return pipeline, an end of the oil return pipeline is connected to an inlet of the variable oil pump (10), and the oil return pipeline is connected to an energy storage section, wherein the energy storage section comprises at least one accumulator, which is connected to the oil return pipeline through an accumulator valve;
the inlet of the variable oil pump (10) is connected to an oil tank (13), and a check valve is installed between the variable oil pump (10) and the oil tank (13), wherein the connection between the oil return pipeline and the variable oil pump (10) is located between the check valve and the variable oil pump (10), wherein the oil supply pipeline is connected to an oil supply pressure sensor (9) and a safety valve (12);
a rodless chamber (31) of the lifting cylinder (3) is connected to a lifting solenoid valve (7), and the lifting solenoid valve (7) is connected to the energy storage section and the variable oil pump (10), wherein the variable oil pump (10) is connected to the oil tank (13), and an outlet of the variable oil pump (10) is connected to the oil supply pressure sensor (9) and the safety valve (12), wherein the variable oil pump (10) is operatively connected to the engine (16), and the engine (16) is equipped with a speed sensor (15), wherein the swashplate (18) of the variable oil pump (10) is connected to a variable cylinder (19), and the variable cylinder (19) is connected to a solenoid valve of the variable oil pump (17), wherein an end of the variable cylinder (19) is connected to a displacement sensor (20), and an amplifier (21) is installed between the displacement sensor (20) and the solenoid valve of the variable oil pump (17).

10. The control system for cascade utilization of recovered energy in a forklift hydraulic system according to claim 9, **characterized in that** the energy storage section comprises a first accumulator (41) and a second accumulator (51), wherein the first accumulator (41) is a high-pressure accumulator, and the second accumulator (51) is a low-pressure accumulator, wherein the first accumulator (41) is connected to a first oil pressure sensor (42), and the first accumulator (41) is connected to the oil supply pipeline through a first accumulator valve (43), wherein the second accumulator (51) is connected to a second oil pressure sensor (52), and the second accumulator (51) is connected to the oil supply pipeline through a second accumulator valve (53).

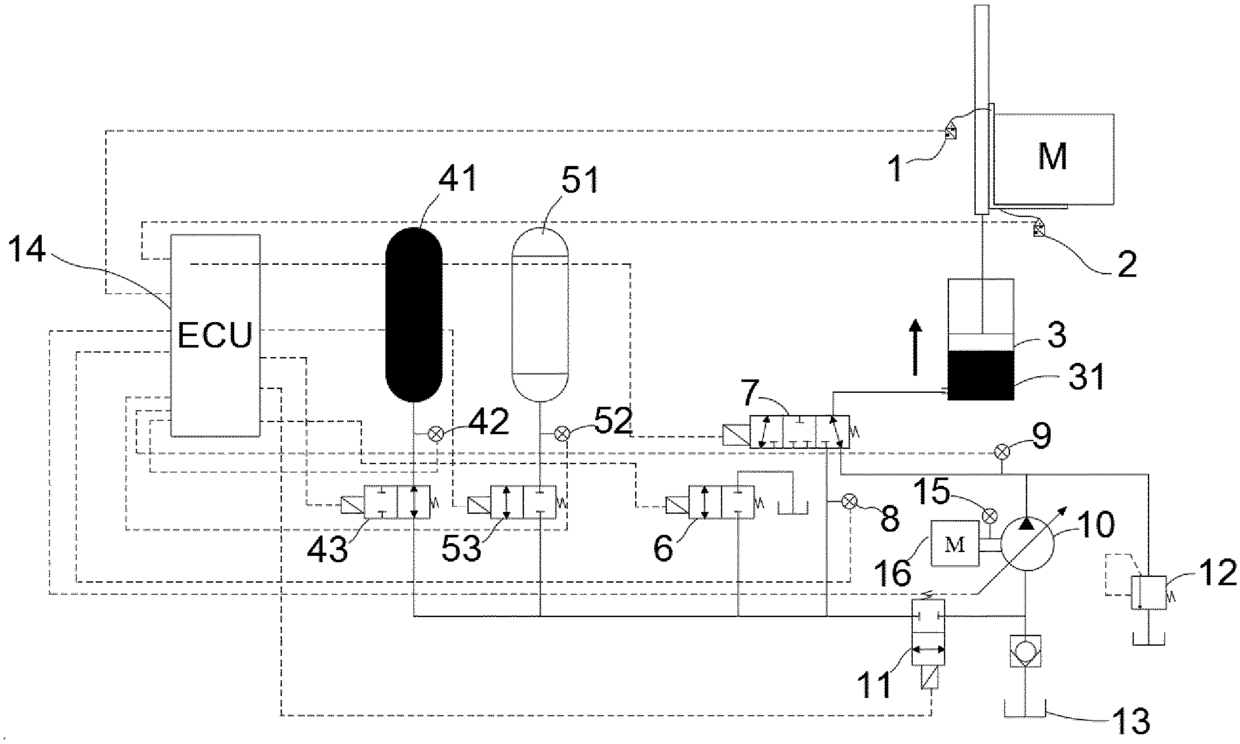

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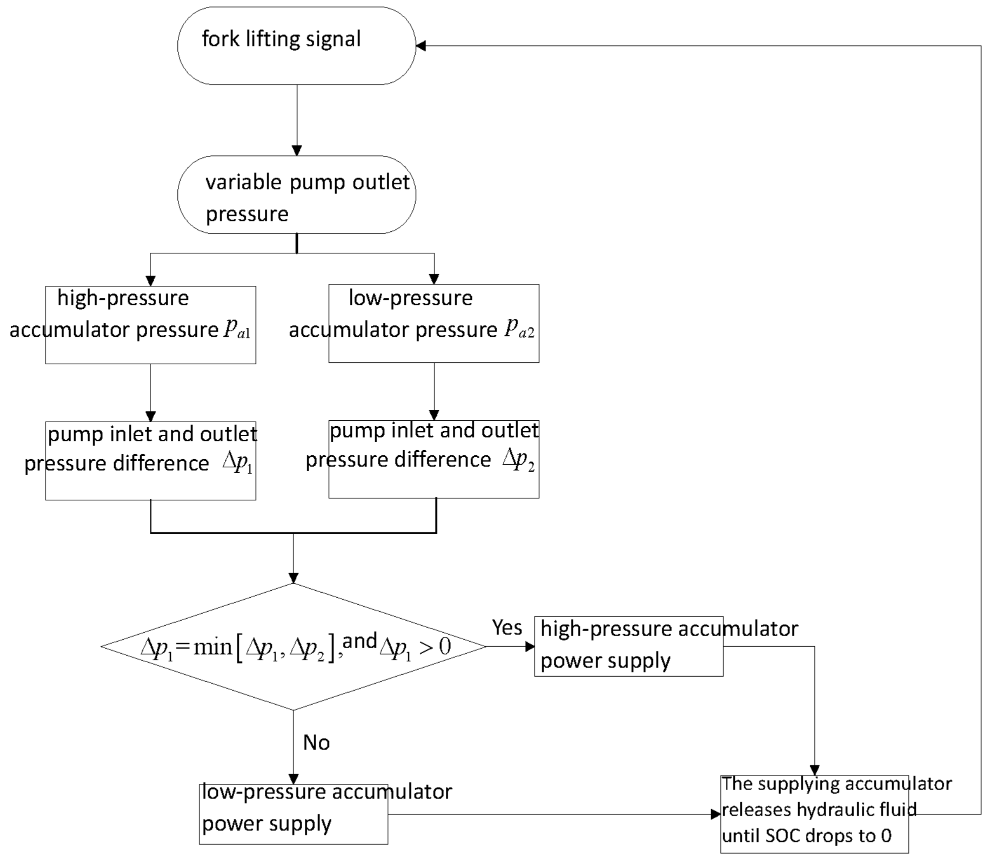
Fig. 1



fork lifting signal
variable pump outlet pressure
high-pressure accumulator pressure P_{a1}
low-pressure accumulator pressure P_{a2}
pump inlet and outlet pressure difference Δp_1
pump inlet and outlet pressure difference Δp_2
$\Delta p_1 = \min[\Delta p_1, \Delta p_2]$, and $\Delta p_1 > 0$
Yes
high-pressure accumulator power supply
No
low-pressure accumulator power supply
The supplying accumulator releases hydraulic fluid until SOC drops to 0

Fig. 2

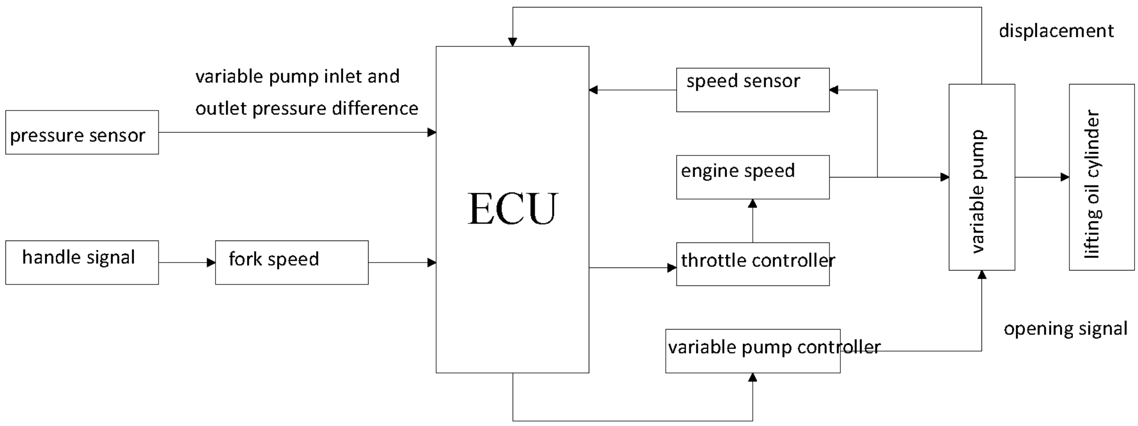

displacement
variable pump inlet and
outlet pressure difference
pressure sensor
ECU
speed sensor
engine speed
throttle controller
variable pump
lifting oil cylinder
handle signal
fork speed
variable pump controller
opening signal

Fig. 3

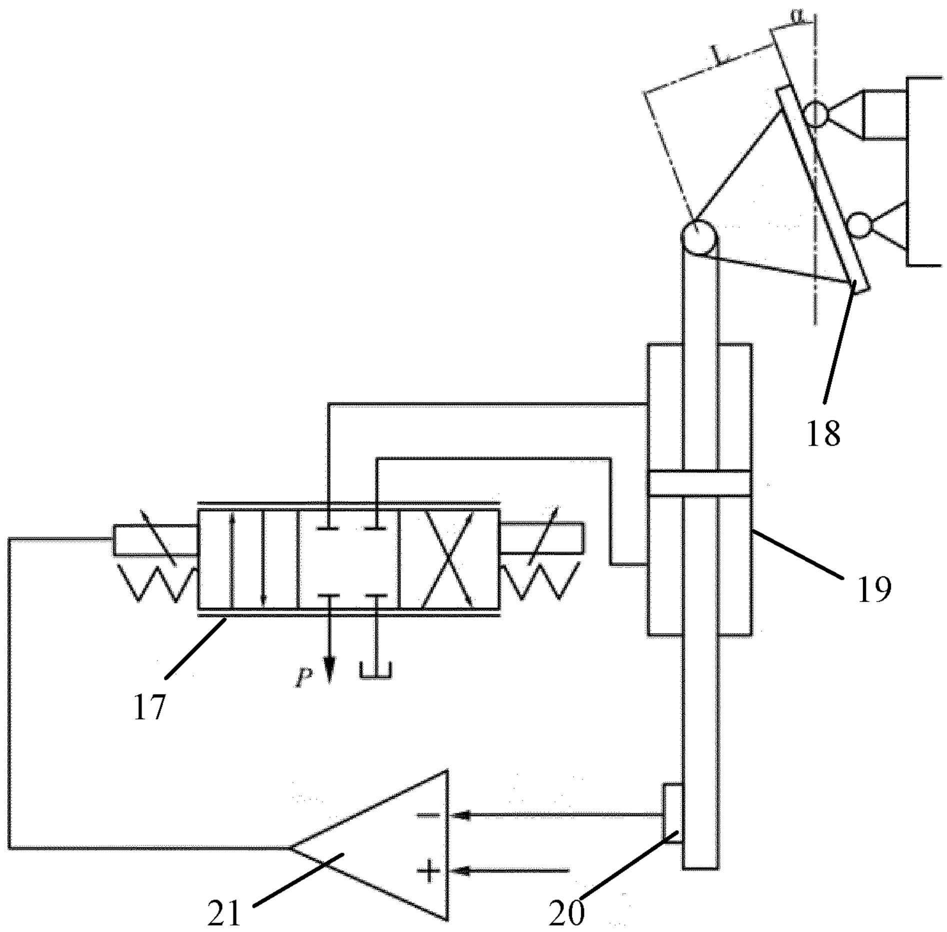

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Fig. 4

INTERNATIONAL SEARCH REPORT

International application No.
PCT/CN2024/091025

A. CLASSIFICATION OF SUBJECT MATTER

F15B21/14(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC:F15B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

CNABS, CJFD, CNTXT, ENTXTC, CNKI: 储能, 蓄能, 电机, 发动机, 缸, 功率, 回收, 再利用, 货叉, 举升, 压力, 液压; DWPI, ENTXT, VEN, WOTXT: accumulate, storage, engine, motor, cylinder, power, reuse, recycle, fork, pressure, hydraulic.

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
PX	CN 116658495 A (ANHUI HELI CO., LTD.) 29 August 2023 (2023-08-29) entire document	1-10
A	CN 114940467 A (HUAQIAO UNIVERSITY) 26 August 2022 (2022-08-26) description, paragraphs 43-118, and figures 1-3	1-10
A	US 2002189250 A1 (BRUUN ECOMATE AKTIEBOLAG) 19 December 2002 (2002-12-19) description, paragraphs 16-48, and figures 1-3	1-10
A	CN 102241379 A (JINAN JINHENG ENERGY-SAVING TECHNOLOGY CO., LTD.) 16 November 2011 (2011-11-16) entire document	1-10
A	CN 104047933 A (BAOSTEEL INDUSTRIAL FURNACE ENGINEERING TECHNOLOGY CO., LTD.) 17 September 2014 (2014-09-17) entire document	1-10
A	CN 111350710 A (HEBEI KAICHANG TECHNOLOGY CO., LTD.) 30 June 2020 (2020-06-30) entire document	1-10

☐ Further documents are listed in the continuation of Box C. ☑ See patent family annex.

* Special categories of cited documents:
- "A" document defining the general state of the art which is not considered to be of particular relevance
- "D" document cited by the applicant in the international application
- "E" earlier application or patent but published on or after the international filing date
- "L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
- "O" document referring to an oral disclosure, use, exhibition or other means
- "P" document published prior to the international filing date but later than the priority date claimed
- "T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
- "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
- "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
- "&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
22 July 2024	**03 August 2024**

Name and mailing address of the ISA/CN	Authorized officer
China National Intellectual Property Administration (ISA/CN) **China No. 6, Xitucheng Road, Jimenqiao, Haidian District, Beijing 100088**	Telephone No.

Form PCT/ISA/210 (second sheet) (July 2022)

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/CN2024/091025

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