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Simulation of energy recovery system of high speed transmission shaft dynamic torque test bench

Simulation of energy recovery system of high speed transmission shaft dynamic torque test bench

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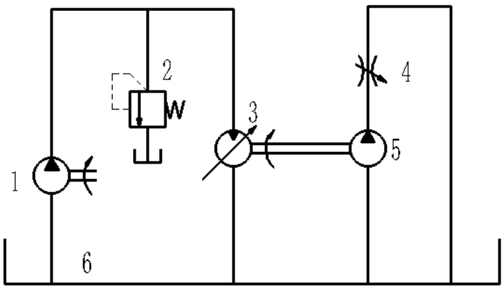
Abstract—A hydraulic energy recovery and control system is proposed to solve the energy loss problem of the current vehicle transmission shaft torque test bench. On the basis of the original system, power recovery is realized by parallel hydraulic compensation. PID module is used to control the displacement of piston variable pump and motor respectively to adjust the speed and torque to achieve the established working conditions. The feasibility of the hydraulic system and control scheme is verified by AMEsim- MATLAB software and experiments. The results show that the proposed system can not only adjust the set speed and torque dynamically, but also achieve high power recovery.

1. Introduction

As the main artery of energy transmission, the running condition of the drive shaft directly affects the safety of the vehicle. In the process of power transmission, the actual working conditions are complex and changeable, and the drive shaft must bear the adverse effects of sudden change of torque, overload operation, high amplitude vibration and impact. In the transportation industry, transport vehicles need to run for a long time under heavy load, so the safe and stable operation of the transmission shaft is very important [1]. Therefore, from production to put into use must go through strict quality inspection, including torque test. Torque reflects the acceleration capacity and load capacity, and the bearing capacity of equipment, work efficiency, service life and safety performance is closely related, it is a typical parameter to measure the overall performance of mechanical transmission system [2]. Different from the static torque test, the dynamic torque test simulates the actual working state of the rotating shaft of the vehicle, and measures the torque and speed parameters of these parts under the condition of loading and running, which can scientifically evaluate the transmission efficiency and power distribution of the whole power mechanical transmission system, and find the problems existing in the transmission shaft [3].



1—Hydraulic pump 2—The overflow valve 3—Hydraulic motor 4—Throttle valve 5—The load of pump
Fig.1 Conventional torque testing system

The existing dynamic torque test hydraulic system (as shown in Fig. 1) uses hydraulic pump 1 to supply oil to the motor to drive the drive shaft. The other end is throttled by a throttle valve to generate load torque. The system uses the drive shaft as an energy transfer device rather than an actuator, and with the exception of small friction losses and rotational kinetic energy, most of the energy is lost as heat as the oil passes through the throttle [4]. The hot oil drained into the tank needs to be cooled, resulting in a secondary waste of energy. If the drive shaft for a long time, high load, large quantities of dynamic torque test is bound to cause a lot of energy waste, does not meet the hydraulic machinery in the more and more important environmental protection and energy saving requirements. Aiming at this problem, the hydraulic energy recovery secondary regulation technology is applied to reform the torque testing system to achieve the goal of energy saving and emission reduction.

2. Composition and principle of the improved torque test and control system

2.1 Parallel hydraulic compensation power recovery system

The energy recovery torque test system is similar to the original system. The difference is that the oil outlet of the original loading pump is communicated with the hydraulic motor through a one-way valve, and the compensation pump supplies oil to the motor together. Accumulator can effectively attenuate oil pulsation, absorb the pressure shock protection system caused by the sudden change of motor and return pump displacement, and improve the stability of system work.

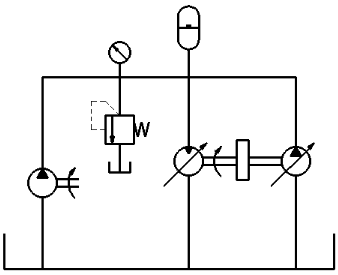


Fig.2 Original drawing of torque test energy recovery system

According to the torque balance formula:

$$\frac{P_P V_P}{2\pi\eta_{\mathrm{pm}}} = \frac{P_M V_M}{2\pi}\eta_{\mathrm{m}m}$$

In the formula, P_P means the pressure difference at both ends of return pump. P_M means the differential pressure between two ends of variable motor. V_P means Return pump displacement. V_M means Variable motor displacement. $\eta_{\mathrm{p}m}$ means Mechanical efficiency of return pump. $\eta_{\mathrm{m}m}$ means

Variable motor mechanical efficiency[5].

During normal operation, $P_P = P_m$, the displacement of hydraulic motor and return oil pump controlled by electro-hydraulic servo valve determines the speed and load torque respectively. The displacement difference determines the rotational acceleration. In order to stabilize the speed, it is necessary to respond to a fast and accurate control strategy to dynamically adjust the displacement difference.

2.2 Torque and speed control strategy

In order to meet the diversity of transmission shaft torque test, test torque and speed can be adjusted dynamically. Double closed-loop negative feedback control mode is adopted. The displacement of return oil pump and hydraulic motor determines the torque and speed of the transmission shaft respectively. The control principle is shown in the figure. When there is a deviation between the preset torque and the actual torque, the deviation signal will adjust the pump displacement through the controller in real time until the actual torque reaches the set value. The electric signal into the servo valve is the superposition of torque control signal and speed deviation control signal. There is a difference in speed can also be in the initial state of torque balance feedback adjustment.

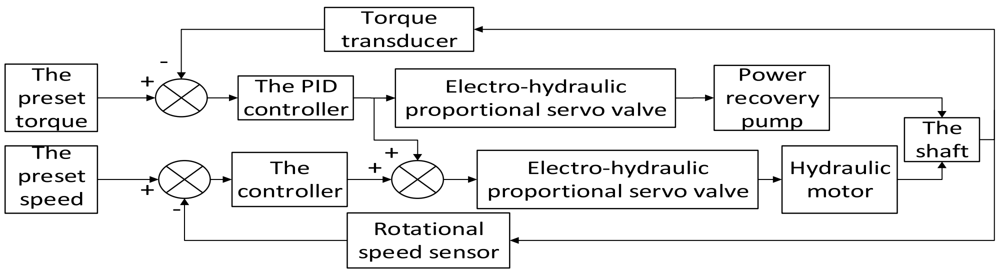


Fig.3 Control schematic diagram

2.3 Analysis of energy recovery efficiency of torque testing system

According to the flow balance formula:

$$Q_{po}\eta_{pv} + Q_{p'0}\eta_{p'v} = \frac{Q_{mo}}{\eta_{mv}}$$

Where $\eta_{p'v}$, η_{pv}, η_{mv} respectively represent the volumetric efficiency of parallel compensation pump, return pump and variable motor. $Q_{p'0}$, Q_{p0}, Q_{m0} respectively is the theoretical flow of parallel compensation pump, return pump and variable motor [6].

$$Q_{p'o} = \frac{Q_{mo}}{\eta_{mv}\eta_{p'v}} - \frac{Q_{po}\eta_{pv}}{\eta_{p'v}}$$

Considering the overflow loss of the overflow valve, take

$$Q_{p'o} = \frac{Q_{mo}}{\alpha\eta_{mv}\eta_{p'v}} - \frac{Q_{po}\eta_{pv}}{\eta_{p'v}}$$

Where α means overflow influence coefficient is 0.96-1.

The power recovery coefficient of the system is given by

$$\xi = \frac{N_m}{N_m + N_{P'}}$$

$$N_{p'} = \frac{pQ_{p'o}}{\eta_{p'm}} = \frac{p}{\eta_{p'm}}\left(\frac{Q_{mo}}{\alpha\eta_{mv}\eta_{p'v}} - \frac{Q_{po}\eta_{pv}}{\eta_{p'v}}\right)$$

$$N_m = \mathrm{p}Q_{mo}\eta_{mm}$$

Where N_m, N_p respectively is the output power of variable motor and compensation pump.

The flow rate of return pump is given by

$$Q_{\mathrm{po}} = Q_{mo}\eta_{Pm}\eta_{Mm},$$

$$\xi = \frac{\alpha\eta_{\mathrm{p'}}\eta_m}{1+\alpha\eta_{p'}\eta_m - \alpha\eta_p\eta_m}$$

In the formulation $\eta_{\mathrm{p'}}, \eta_{\mathrm{m}}, \eta_{\mathrm{p}}$ respectively represent Total efficiency of refill pump, variable motor and return pump. The mechanical efficiency and volume efficiency of compensation pump, return pump and hydraulic motor are 0.95 and 0.85 respectively. Take $\alpha = 0.96$. The calculated recovered power can reach 62.6% [7].

3. AMEsim-MATLAB Torque test system modeling

Co-simulation technology is used for modeling, which gives full play to the powerful hydraulic system modeling function of AMEsim and significant advantages of MATLAB control system modeling.

3.1 Hydraulic system model of AMEsim.

A simplified model of torque testing system is established in AMEsim as shown in the figure, including constant pressure variable pump, motor, relief valve, sensor and other components.

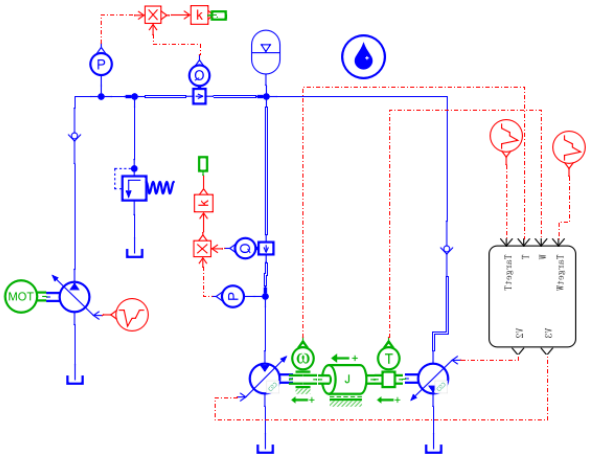


Fig.4 Hydraulic system simulation model

Where Motor speed is $1000\,\mathrm{rmp}\cdot\mathrm{min}^{-1}$; relief valve opening pressure is 15MPa; accumulator capacity is 5L; hydraulic pump, hydraulic motor, compensation pump displacement is 180cc/rev. The oil density is $850\,\mathrm{kg}/m^3$. The static friction torque is 20N/M. The viscous friction coefficient is 0.01 $N\mathrm{m}/(rev/\mathrm{min})$.

3.2 Control system model of simlink

The general form of the conventional PID control algorithm is:

$$\mathrm{u}(t) = k_p e(t) + k_i\int_0^t e(t)dt + k_d\frac{de(t)}{d(t)}$$

Where, k_p is the proportional coefficient, $\mathrm{k_i}$ is the integral coefficient and $\mathrm{k_d}$ is the differential coefficient. $e(t)$ is the real-time deviation of the control quantity. The controller is built according to the control strategy as shown in the figure 6. The torque control adopts traditional PID control considering the rapidity of the response of the system itself, and the PID control parameters are set as $k_p = 0.01, k_i = 0.1, k_d = 0$.

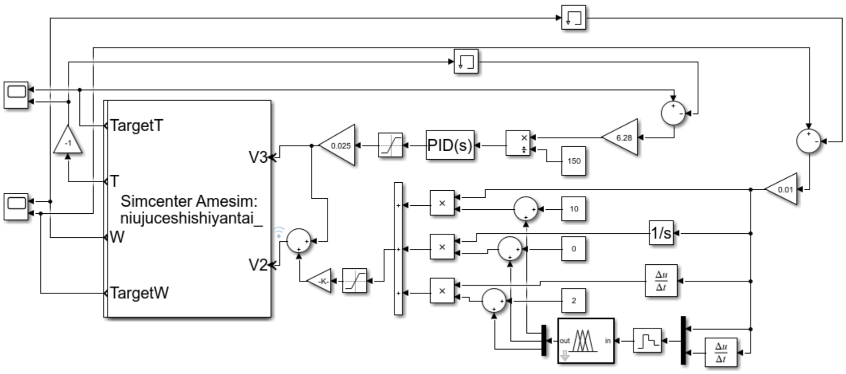
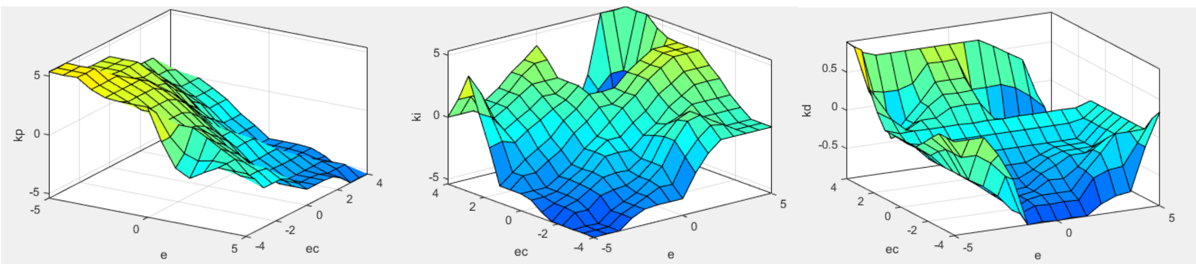


Fig.5 Control system simulation model

In order to improve the response characteristics of speed control. Fuzzy PID control is used to dynamically adjust the parameters of proportion, differential and integral links. The controller takes the speed error E and the rate of change of the speed error EC as the input and quantifies them together with the incremental adjustment parameters (Δk_p, Δk_i, Δk_d) of PID, and into seven grades, which are NB, NM, NS, ZO, PS, PM and PB respectively. The triangle membership function is selected.

When the deviation is large, in order to speed up the system response, Δk_p should be taken as large; in order to avoid differential saturation beyond the control range, Δk_d should be taken as small; in order to avoid large overshoot, the integral effect can be removed. When the deviation is small, in order to make the system have better steady-state performance, large Δk_p and Δk_d should be taken, and in order to avoid shock near the equilibrium point, appropriate Δk_d should be taken. In summary, the fuzzy adjustment rules are shown in the figure above.



(a) proportional coefficient (b) integral coefficient (c) differential coefficient

Fig.6 Fuzzy PID adjustment principle

4. Dynamic torque test simulation analysis

In order to verify the torque testing system under the action of the controller can be stable at the expected torque and expected torque, and can achieve the theoretical power recovery efficiency. Step expected speed signal of 500r/min was selected, simulation time was 100s, sampling time was 0.01s.

In order to shorten the simulation time, speed up the rotational speed and improve the rotational acceleration, the expected torque was set to 0 in the first 20 seconds of the simulation, and the expected torque was increased by 100N/M every 20 seconds thereafter.

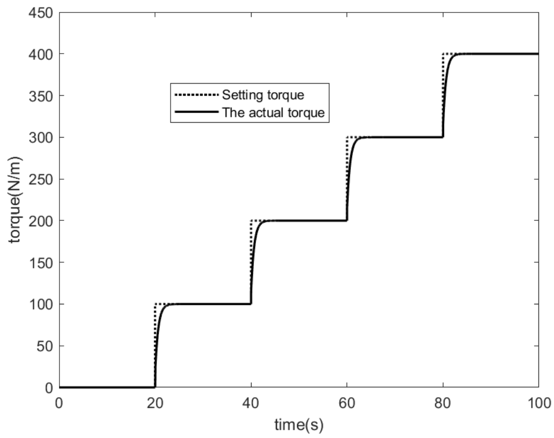


Fig.7 Transmission shaft torque response

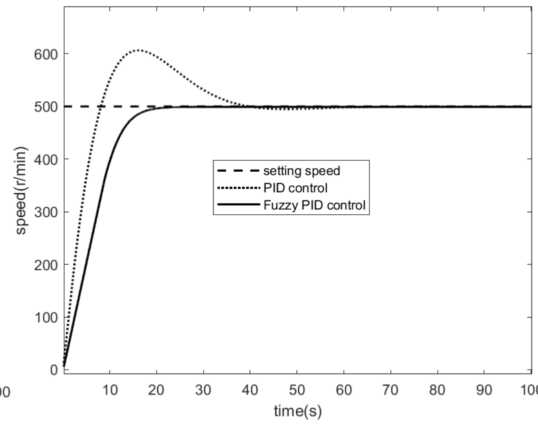


Fig.8 Transmission shaft speed response

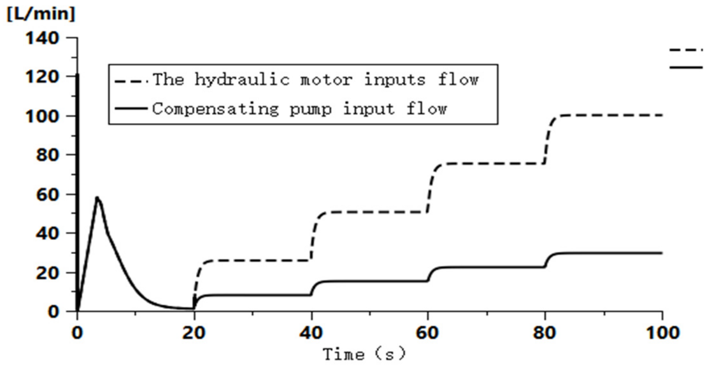


Fig.9 Flow curve

4.1 Torque control analysis
Torque simulation response is shown in the figure 8, and the rapidity and accuracy of torque response can be satisfied by applying PID control.

4.2 Speed control analysis
For the desired target speed, the simulation results of the two control systems are shown in the figure 9. Because traditional PID can not adjust parameters, it has a higher overshoot, and the adjustment time is longer. Fuzzy PID solves this problem and reduces the system error and adjustment time, and torque loading will not affect the speed change after the speed is stabilized.

4.3 Analysis of power recovery effect
The output flow of the compensation pump and the hydraulic motor under the above working conditions are shown in the figure 10. When the torque is set to 0 for the first 20 seconds, the hydraulic motor is supplied entirely by the compensation pump, so the two flows are equal. During rotation, the inlet and outlet pressures of the compensation pump and hydraulic motor were kept at 150bar. Using the simulated outlet flow, the power recovery efficiency of the system could be calculated by the equation, and the power recovery rates of different load torques in 20-40s, 40-60s, 60-80s, and 80-100s, were shown in the following table.

Tab.1 Simulate input and output flow

component	1	2	3	4
hydraulic motor	27.7L/min	54.0L/min	80.4L/min	106.8L/min
Compensation pump	10.9L/min	20.7L/min	30.5L/min	40.3L/min
Power recovery factor	60.6%	61.6%	62.1%	62.3%

5. Conclusion

According to the characteristics of high power consumption of the original drive shaft dynamic torque testing hydraulic system, the energy recovery is realized by improving the control strategy. Simulation experiments based on AMEsim and MATLAB verify the energy-saving control characteristics of the new system.

(1) The control system can stably and reliably control the torque and speed of the transmission shaft. Compared with traditional PID control, fuzzy PID control has lower overshoot and shorter adjustment time.

(2) The new test system combined with control technology can effectively save energy under the guarantee of test requirements.

This paper is limited to the simulation and theoretical research, the simulation speed did not improve the actual system pressure fluctuation can control scheme to do more improvement and optimization and experimental verification and promotion. The torque test system of energy recovery meets the background of energy saving and emission reduction and saves energy effectively.

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