Application of fuzzy control in power recovery testing of hydraulic pump and motor

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Abstract. In order to realize the test purpose of high pressure and large displacement hydraulic pump and hydraulic motor, this paper designs a power recovery test system using fuzzy controller, and explains the working principle of the system. Through the theoretical calculation of its power recovery rate, this paper proposes how to maintain the maximum power recovery rate of the test system. Finally, the feasibility of the system was verified by theoretical simulation and experiments.

1. Introduction
The test of large displacement hydraulic pump and motor can be divided into direct test and power recovery test. The previous power recovery tests cannot keep the maximum power recovery rate of the test system. Based on this situation, this paper designs a split-type high-power hydraulic pump and hydraulic motor load test system. From theoretical analysis to simulation and experiment, the maximum recovery power of the test system is realized by virtual instrument technology and fuzzy control algorithm.

2. Test system analysis
The traditional power recovery methods are divided into mechanical compensation power recovery and hydraulic compensation power recovery. The mechanical power recovery method is mainly used for the hydraulic pump test. The power recovery motor converts the hydraulic energy into the mechanical energy of the shaft and feeds back to the pump under test to achieve the purpose of power recovery. The hydraulic compensation method is mainly used for the test of the hydraulic motor. The mechanical energy of the output shaft of the motor under test is converted into hydraulic energy by the power recovery pump to feed back to the motor under test for power recovery.

The test system of this paper combines the above two traditional power recovery methods. The test system schematic is shown in Figure 1.
From the test system schematic diagram, we get the working principle of the system: adjusting the test speed of the pump under test 1 by turning on the variable frequency main motor 7, and supplying the oil to the suction port of the pump under test 1 by the charge pump 3. The outlet of the pump under test 1 supplies oil to the inlet of the power recovery motor 2, and the compensation pump 4 is used to compensate the flow difference between the pump under test and the power recovery motor, so that the overflow of the proportional relief valve 5 is stabilized at a setting value. When testing the large displacement hydraulic pump, the outlet hydraulic oil of the power recovery motor 2 is returned to the suction port of the tested pump through the ball valve 8 to be replenished with the charge pump. The system combines two traditional power recovery methods to improve the performance of the system.

3. Theoretical analysis of system power recovery rate

The power recovery of the test is analysed by the schematic diagram of the test system in figure 1.

The realization of power recovery is mainly the feedback of the test motor and the test pump to the spindle and torque of the main motor. So the theoretical simplified formula of the power recovery rate is expressed as follows:

\[
\xi = \frac{N_{a2} - N_{a7}}{N_{a7}} = 1 - \frac{N_{a2}}{N_{a7}}
\]

(2.1)

In the formula:
- \(N_{a7}\) —— the output power of the main motor 7 when there is no power recovery;
- \(N_{a7}\) —— the output power of the main motor 7 during power recovery;

According to the spindle power balance can be obtained:

\[
N_{i1} = N_{a2} + N_{a7}
\]

(2.2)

In the formula:
- \(N_{i1}\) —— the input power of the test pump 1;
- \(N_{a2}\) —— the output power of the test motor 2;

Output power of motor without power recovery:

\[
N_{a7} = \frac{P \cdot n \cdot q_{1}}{\eta_{m1}}
\]

(2.3)

In the formula:
- \(P\) —— system pressure difference;
- \(n\) —— spindle speed;
- \(q_{1}\) —— the displacement of the test pump 1;
- \(\eta_{m1}\) —— mechanical efficiency of the test pump 1;
Output power of the test motor:

\[ N_{o2} = P \cdot n \cdot q_2 \cdot \eta_{m2} \]  \hspace{1cm} (2.4)

In the formula:  
- \( q_2 \) ——— the displacement of the test motor 2;  
- \( \eta_{m2} \) ——— the mechanical efficiency of the test motor 2;  

Motor output power during power recovery:

\[ N_{e2} = \frac{P \cdot n \cdot q_1 \cdot P \cdot n \cdot q_2 \cdot \eta_{m2}}{\eta_{m1}} \]  \hspace{1cm} (2.5)

According to the flow balance equation:

\[ Q_1 + Q_2 = Q_3 + \Delta Q \]  \hspace{1cm} (2.6)

In the formula:  
- \( Q_1 \) ——— output flow of the test pump 1;  
- \( Q_2 \) ——— input flow of the test motor 2;  
- \( Q_4 \) ——— output flow of the compensating pump 4;  
- \( \Delta Q \) ——— overflow flow of proportional relief valve 5;  
- \( \eta_{s1} \) ——— the volumetric efficiency of the test pump 1;  
- \( \eta_{s2} \) ——— the volumetric efficiency of the test motor 2;  
- \( n_1 \) ——— motor speed of the compensation pump 4;  
- \( q_4 \) ——— displacement of the compensation pump 4;  
- \( \eta_{s4} \) ——— the volumetric efficiency of the compensation pump 4;

Calculated and compiled:

\[ \frac{q_2}{q_1} = \eta_{s1} \cdot \eta_{s2} + \frac{n_1}{n} \cdot \eta_{s4} \cdot \eta_{m2} \cdot \frac{q_4}{q_1} \cdot \frac{1000 \cdot \eta_{s2} \cdot \Delta Q}{n} \]  \hspace{1cm} (2.10)

The power recovery rate is calculated by substituting Equations 2.3, 2.6 and 2.11 into Equation 2.1:

\[ \xi = \eta_{s1} \cdot \eta_{s2} \cdot \eta_{m1} \cdot \eta_{m2} + \left( \frac{\eta_{s2} \cdot \eta_{s4} \cdot q_4}{n \cdot q_1} \cdot \frac{1000 \cdot \eta_{s2} \cdot \Delta Q}{n} \right) \cdot \eta_{m1} \cdot \eta_{m2} \]  \hspace{1cm} (2.11)

From the derived power recovery formula, it can be concluded that the recovery rate of the system can reach the theoretical maximum only when the overflow flow tends to zero and the motor speed of the compensation pump 4 are at the minimum required speed. Supposing the volumetric efficiency is 0.95, the mechanical efficiency is 0.9. Ignoring the influence of the compensation pump 4, the power recovery rate of the system under ideal conditions can be 0.73.

According to the above derivation, in order to maintain the maximum power recovery rate during the operation of the system, the overflow of the system should always maintain a stable and small flow. According to the design of the system, we adjust the compensation flow by controlling the speed of the compensation pump motor, and indirectly control the overflow.

4. Design of Fuzzy Control System

4.1. Control system selection

Based on the results of the above theoretical analysis, it is clear that the purpose of the control is to stabilize the overflow of the relief valve, and the control object is the speed of the motor. The mathematical model between the overflow and motor speed is complex, and is affected by the volumetric efficiency of the motor and the pump, so it is impossible to derive an accurate theoretical formula. Based on these conditions, we analyze the control algorithms which are used more at present and choose the appropriate algorithm for this system:

Fuzzy-PID controller: In essence, it is also the function of realizing the PID controller. Through fuzzy control, the three-loop parameters of the PID controller can be adjusted automatically, so that the three-loop parameters of the PID controller can be kept in the optimal state in real time.
Fuzzy controller: Fuzzy controller does not need to establish a mathematical model. It uses the mode-behaviour control mode to simulate and summarize the human response in this situation. It uses some uncertain concepts such as larger and smaller to summarize and express deterministic quantities. It has strong adaptive control ability for the system with large nonlinearity and delay.

After considering and trying all kinds of control methods, the test system finally chooses the fuzzy control system to control the motor speed, so that the overflow can be stabilized near a set value.

4.2. Fuzzy control system structure

As shown in figure 2, the fuzzy control system consists of a second-order fuzzy controller and the controlled object. The change rate is calculated through the change of the overflow of the overflow valve collected by the flow sensor. The change and the change rate are regarded as two input variables of the fuzzy controller. After the operation of the fuzzy controller, the speed of the motor is controlled by the output. Through D/A module, the control signal is sent to the variable frequency motor to adjust the real-time speed of the compensation pump motor to change the compensation flow. Through closed-loop control, the size of overflow can be corrected in real time, so that the overflow can be stabilized at a set value.

![Figure 2. Schematic diagram of the fuzzy control system](image)

4.3. Establish fuzzy control rules

The key of the fuzzy control system used in this paper is to establish the fuzzy relationship between the flow error value and the error change value and the motor speed of the output compensation pump. Then the flow error e and the error change ec are continuously detected through closed-loop control during the testing process. The corrected value of the motor speed ∆n at this time is deduced by the fuzzy rules, which makes the system have good dynamic and static performance.

Control law: After manual testing, the output flow of the compensation pump increases by 0.055L/min with the increase of motor speed. The change of the overflow and motor speed basically meet the proportional relationship of variable slope.

According to the method of fuzzy reasoning, we fuzzify the three parameters of error e, error change ec and output ∆n, and express them by fuzzy sets. The fuzzy sets of three parameters {e, ec, ∆n} are represented by {NB NM NS ZO PS PM PB} with letters denoting: Negative big, Negative middle, Negative small, Zero, positive small, positive middle, positive big. After determining the fuzzy linguistic variables, the domain of the fuzzy subset of the three variables is determined. The domain of the fuzzy subset of the input variables {e, ec} is chosen to be {-6, -5, -4, -3, -2, -1, 0, 1, 2, 3, 4, 5, 6}, and the domain of the fuzzy subset of the output variable ∆n is {-7, -6, -5, -4, -3, -2, -1, 0, 1, 2, 3, 4, 5, 6, 7}. The basic domain of the flow error e is E: [-2, 2] L / min, and the quantization factor of the error is: $K_e = \frac{6}{2} = 3$.

The basic domain for determining the flow error variation ec is EC: [-5, 5], and the quantization factor of the error change is determined as follows: $K_c = \frac{6}{5} = 1.2$ . The basic domain for determining the output speed increment ∆n is N: [-10, 10], and the proportional factor of output $K_u = \frac{7}{10} = 0.7$ . Therefore, the membership functions of flow error e, error variation ec and output ∆n are determined as shown in Figure 3-5.
After determining the membership function of input and output, the fuzzy rules of the control system are determined. According to the parameter setting rules and the previously established fuzzy sets and membership functions, the table of the fuzzy control rules output speed variation can be obtained as shown in Figure 6. The change of output ∆n under different values of e and ec can be seen intuitively from the three-dimensional graph shown in Figure 7.

![Figure 6. Output ∆n fuzzy rule](image)

![Figure 7. Three-dimensional map](image)

4.4. Simulation verification
The simulation system is built under Labview programming environment. The simulation experiment is carried out with the sinusoidal signal with amplitude of 10L/min and frequency of 0.05Hz as the simulation signal to determine the response speed and control deviation of the system. The test results are shown in Figure 8 (left), and the control error is shown in figure 8 (right). The analysis shows that the response time of the control system is related to the control cycle time of the system. There is basically no lag in the theoretical simulation. It can be seen from the figure that the response is very fast and meets the control requirements. It can be concluded from the error graph that the instantaneous maximum error of the system is less than 0.9L/min, and the root mean square value of the dynamic error calculated by the computer is 0.4L/min. Based on the above simulation results of the sinusoidal signal with noise, it is concluded that the control system meets the control requirements of power recovery.

![Figure 8. Controller with noise sinusoidal signal test curve (left) and error curve (right)](image)

5. Analysis of experimental results
The pressure-flow curve of the final power recovery pump and motor test results is shown in figure 9.
During the test process, the recorded data is shown in figure 10. It is observed that the overflow of the relief valve can be stabilized at about 5L/min, and the error is within 1.5L/min. The actual control effect of the fuzzy controller is good to meet the test requirements. The real-time power recovery rate calculated by the measurement and control program is about 0.67, and the power recovery effect is good, which has a good practical application reference value.

![Figure 9. Pump motor test results pressure flow curve](image)

![Figure 10. Data recorded during the experiment](image)

Through the above test results, it can be concluded that the fuzzy control can make the power recovery hydraulic pump and motor test achieve the maximum power recovery rate in the whole test process, reduce the waste of energy as far as possible, and achieve good control effect. The actual test system lags behind in flow acquisition and speed control, which results in larger errors in actual test than in theoretical simulation. In the later stage, we can continue to try to use a faster response servo motor to drive the compensation pump. The test system runs well in practice. It provides practical application reference for hydraulic power recovery system, and has certain theoretical and practical value.

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