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Analysis of Torque Loading System Based on Hydraulic Energy Closed-loop

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Abstract

Power head with high power and large torque is the main power equipment of the rotary drilling rig. Aimed at the working condition, energy saving torque loading test program of power head was put forward and mathematical and simulation models of hydraulic energy closed-loop torque loading system were established. Combined with bond graph and state equations of torque loading system with single pump, effect of the logical relationship of discharge between the driving motor and loading pump and the relationship between the adjusting proportion of series and parallel pressure on the torque loading system was analyzed because both hydraulic flow and pressure can have an influence on the system stability. It proves that the torque loading system is stable when the adjusting proportion of series pressure is larger than that of the parallel pressure. Otherwise, it needs to consider the discharge trend between the power motor and loading pump.

Keywords: *hydraulic energy closed-loop, power head, torque load, bond graph, AMESim simulation, system analysis*

1. Introduction

Rotary drilling rig is an ideal construction drilling machine in the foundation building engineering with large output torque and high construction efficiency. Rotary drilling rig is a kind of soil pile hole construction machinery. It can drill pipe by cutting soil to the outer ground driven by rotary bucket and cyclic operation [1]. Power head is the most important part of the rig and its structure is shown in Figure 1.

Figure 1. Power Head Structure

Power head pressure plate directly contacts with components providing the axial pressure in the working process of rotary drilling rig. Axial steady-state load on the power head is constructed through the cylinder. Power head also bears axial impact load additionally because the feeding hole construction is in a harsh working environment. There exists various damage of power head and the working stability of power head of rotary drilling rig directly determines whether it can work normally. This paper presents the use of hydraulic energy closed and power feedback conservation program. It not only can complete the torque loading test of power head, and the use of energy recovery methods to realize energy saving can improve the utilization rate of power head test bench and prolong the service life of the. However, the power head test bench has large power, large torque, hydraulic power closed

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and complex system structure, so the key is to analyze the stability of power head torque loading system. Through the comparison between the series and parallel pressure, influences of pump and motor output and series-parallel pressure control ratio on the system stability are achieved by AMESim simulation. By time-sharing segmentation of parameter variation tendency method, it shows the effect of pressure discharge of the hydraulic system on the stability of the system and it verifies that when the series pressure control ratio larger than parallel pressure control ratio, the system keeps stable.

2. Scheme of Hydraulic Closed Torque Loading System

Energy feedback test generally has three kinds which are electric closed type [2], machinery closed type [3] and hydraulic closed type [4]. By references to some relevant articles about hydraulic system structure and construction methods [5-6]. In practical application, according to the specific circumstances to determine the implementation plan, the power head's torque is provided by the hydraulic motor, so the hydraulic energy recovery method is a natural choice. The principle of mechanical part in torque loading system shown in Figure 2, the power head is connected with the connecting plate, the power box gear transmission system is connected with the external transmission equipment and the whole system can transfer torque between the motor and pump. The principle of closed loop hydraulic loading system is shown in Figure 3. The hydraulic motor and the loading pump 2 make a rigid connection through the power head 6, gear box 5, a torque sensor 4. Energy in the transmission process, due to leakage, friction power loss is provided by hydraulic compensating pump to compensate the system process loss. Heating of the system decreases so as to recyclable use most of the energy and achieve energy saving to a large extent.

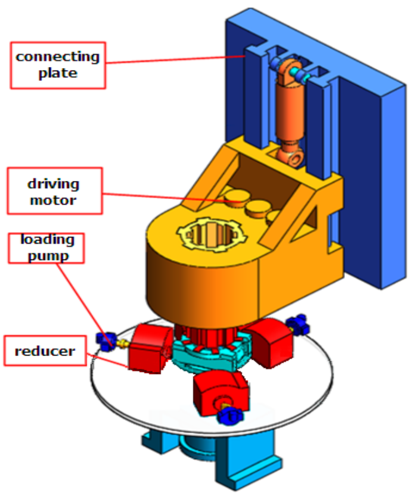
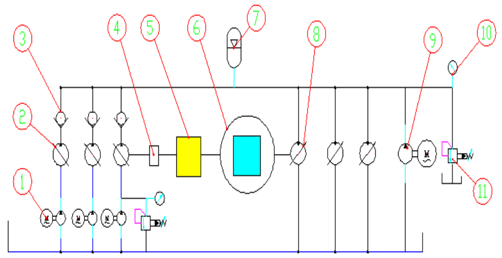


Figure 2. Mechanical Schematic Diagram of Torque Loading System



1- series compensation pump 2- loading pump 3-check valve 4- torque sensor 5-gearbox 6- power head
7- accumulator 8-power head motor 9-parallel compensation pump 10- pressure gauge 11- relief valve

Figure 3. Simplified Hydraulic Schematic Diagram of Torque Loading System

3. Theoretical Analysis

The power head torque loading test bench is a kind of mechanical and hydraulic hybrid energy feedback complex test system. From simple to start, it firstly carries on the analysis to the single pump loading conditions. Loading pump and driving motor through a speed reducer, gear box form a closed loop, and parallel compensation pump and series compensation pump add power loss. According to the working principle diagram and based on the bond graph theory [7-10], the hydraulic closed torque loading test system bond graph model as shown in Figure 4.

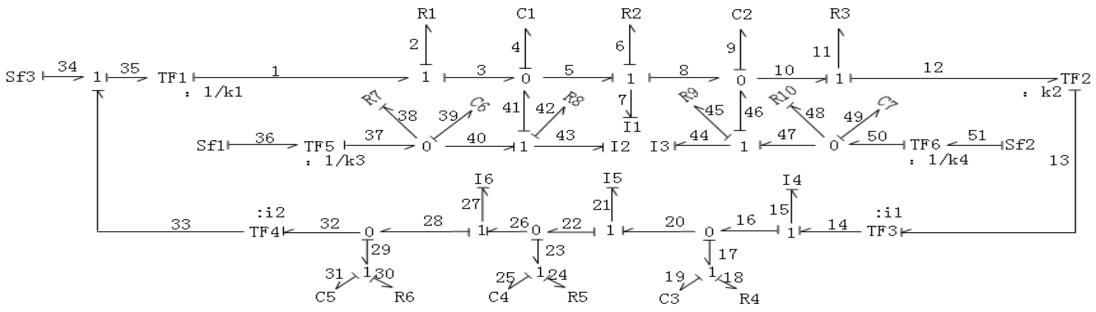


Figure 4. The Bond Graph of Load System with Single Pump

where,

transfer elements:
TF1—loading pump
k1—capacity of loading pump each radian
TF2—driving motor
k2—capacity of driving motor each radian
TF3—reducer
i1—reducer transmission radio
TF4—gear box
i2—gear box transmission radio
liquid capacity [11-12] elements:
C1—liquid capacity of loading pump
C2—liquid capacity of high pressure pipe
C3—torsional rigidity between small gear and big gear
C4—torsional rigidity between flywheel and big gear
C5—torsional rigidity between flywheel and gearbox
C6—liquid capacity of series compensation circle
C7—liquid capacity of parallel compensation circle
Inertial elements:
I1—liquid inductance of high pressure pipeline
I2—liquid inductance of series compensation circle
I3—liquid inductance of parallel compensation circle
I4—moment of inertia of small gear
I5—moment of inertia of big gear
I6—moment of inertia of flywheel
liquid resistance [13] elements:
R1—liquid resistance of loading pump
R2—liquid resistance of high pressure pipeline
R3—liquid resistance of driving motor
R4—torsional damping between small gear and big gear
R5—torsional damping between big gear and flywheel
R6—torsional damping between flywheel and transmission equipments
R7—liquid resistance of series circle relief valve opening
R8—liquid resistance of series compensation circle
R9—liquid resistance of parallel compensation circle
R10—liquid resistance of parallel circle relief valve opening
source elements:
Sf1—speed of series compensation pump
Sf2—speed of parallel compensation pump
Sf2—speed of loading pump

According to bond graph node 4 and total potential and total flow logic relations as well as the total energy [14] relationship, the system state equations are as follows.

$$\dot{p}_7 = \frac{q_4}{C_1} - \frac{R_2}{I_1}p_7 - \frac{q_9}{C_2} + \frac{R_{11}-R_{12}}{I_3}p_{44} \tag{1}$$

$$\dot{p}_{15} = \frac{1}{i_1 k_2 C_2} q_9 + \frac{p_{44}}{C_3} q_{19} - \frac{R_3 + i_1^2 k_2^2 R_4}{i_1^2 k_2^2 Z_4} p_{15} + \frac{R_4}{I_5} p_{21} \quad (2)$$

$$\dot{p}_{21} = \frac{q_{19}}{C_3} + \frac{R_4}{I_4} p_{15} - \frac{R_4 + R_5}{I_5} p_{21} - \frac{q_{25}}{C_4} + \frac{R_5}{I_6} p_{27} \quad (3)$$

$$\dot{p}_{27} = \frac{R_5}{I_5} p_{21} - \frac{R_5}{I_6} p_{27} + \frac{q_4}{C_4} - \frac{R_6}{I_6} p_{27} + \frac{R_6}{i_2} Sf_3 - \frac{q_{31}}{C_5} \quad (4)$$

$$\dot{p}_{43} = \frac{q_{39}}{C_6} - \frac{q_4}{C_1} - \frac{R_8}{I_2} p_{43} \quad (5)$$

$$\dot{p}_{44} = \frac{q_{49}}{C_7} - \frac{q_9}{C_2} - \frac{R_{10}}{I_3} p_{44} \quad (6)$$

$$\dot{q}_4 = k_1 Sf_3 + \frac{p_{43}}{I_2} - \frac{p_7}{I_1} \quad (7)$$

$$\dot{q}_9 = \frac{p_7}{I_1} + \frac{p_{44}}{I_3} - \frac{1}{k_2 i_1 I_4} p_{15} \quad (8)$$

$$\dot{q}_{19} = \frac{p_{15}}{I_4} - \frac{p_{21}}{I_5} \quad (9)$$

$$\dot{q}_{25} = \frac{p_{21}}{I_5} - \frac{p_{27}}{I_6} \quad (10)$$

$$\dot{q}_{31} = \frac{p_{27}}{I_6} - \frac{1}{i_2} Sf_3 \quad (11)$$

$$\dot{q}_{39} = \frac{Sf_1}{k_3} - \frac{q_{39}}{C_6 \cdot R_7} - \frac{p_{43}}{I_2} \quad (12)$$

$$\dot{q}_{49} = \frac{Sf_2}{k_4} - \frac{q_{49}}{C_7 R_{10}} - \frac{p_{44}}{I_3} \quad (13)$$

Here,

P -- momentum variable corresponding to inductance element I. Variables such as p7, p43 and p44 represent the pressure momentum P corresponding to components of the hydraulic system, whose first derivative says oil pressure p. Variables such as p15, p21 and p27 represent the angular momentum L corresponds to the component of mechanical transmission system, whose first derivative says the corresponding torqueτ.

Q – changeable variable corresponding to the capacitive element C. Variables such as q4, q9, q39, q49 represent the oil volume V corresponding to the components of the hydraulic system, whose first derivative says oil flow rate Q corresponds to the oil. Variables such as q19, q25 and q31 represent angle displacement θ corresponding to the elements of the mechanical drive system, whose first derivative says corresponding angular velocityω.

From the bond graph and state equations of a single pump loading system, it can be seen above the key mechanical components, torque and hydraulic system components such as oil volume and mechanical components such as angular discharge have a relationship, and the system pressure of hydraulic elements influences each other. Among them, mutual influence on parallel compensation pressure and series compensation pressure is typical. Flow change is affected by the pressure momentum and the oil volume change. Obviously, the stability of the system is mainly affected by the two parts of the hydraulic system and mechanical system factors. The stability of the torque loading test bench is the key factor which can ensure the stability of the whole system, that is to say it should not lead to the rapid decline in pressure and lead to system speed decreases rapidly to zero because of a little outside interference. What’s more, for the hydraulic system due to the use of the closed-loop power recovery system, the perturbation caused by pressure and flow is likely to cause instability of the system, so the following analysis put the focus on the two aspects mentioned before which will have an influence on the stability of torque loading system.

4. Simulations

In Figure 5, the power head hydraulic closed torque loading system simulation model is created by means of hydraulic simulation software AMESim [15]. The software AMESim based on the bond graph theory is used for simulation of hydraulic system and mechanical system. Torque loading machine liquid mixing system created on the basis of the development, application of the software is professional to a certain extent. Based on this model, some analysis of the influence of different factors on the stability of the system under various conditions is shown in Figure 6.

Take method of fixed variable in different time, Figure 6 shows the system state changes when the different factors change corresponding to the perturbed system at different time and then judge whether the torque loading system can respond to a new stable state when there is a disturbance. Figure 6 lists of several typical conditions, respectively, starting from the change of flow of loading pump and motor and series-parallel output pressure, system stability can be judged through the change trend of motor rotation speed. In Figure 6(a), when the parallel pressure values set for the respective maximum values in Table 1, loading pump discharge is less than the power motor capacity, which will make the motor speed appear great fluctuation in zero, causing system instability. Otherwise, when loading pump output is greater than the power motor and parallel control pressure ratio is less than or equal to the series pressure proportional control ratio, motor speed will be stable at a certain value and the system is stable.

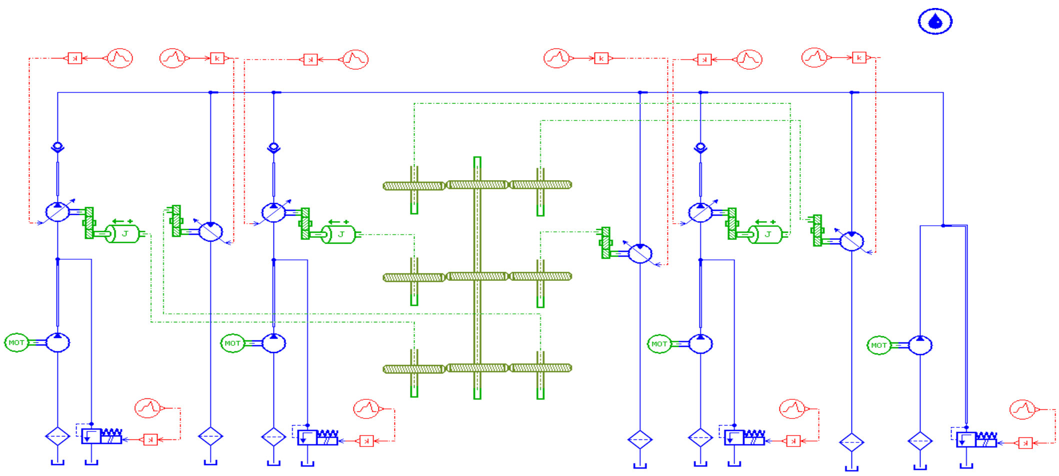
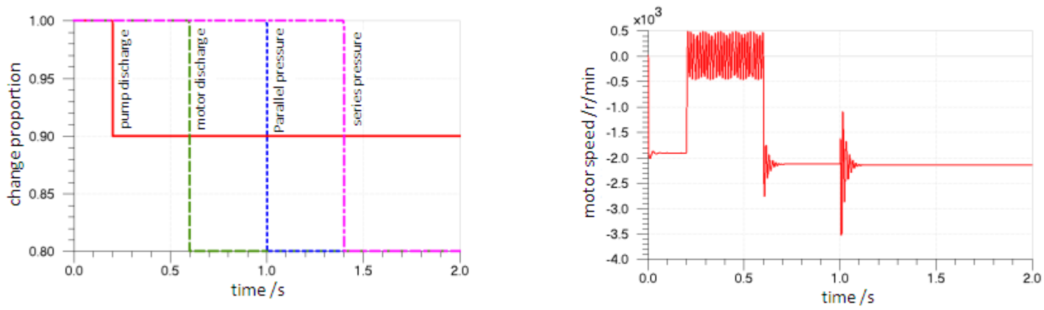


Figure 5. AMESim Simulation Model of Hydraulic Closed System

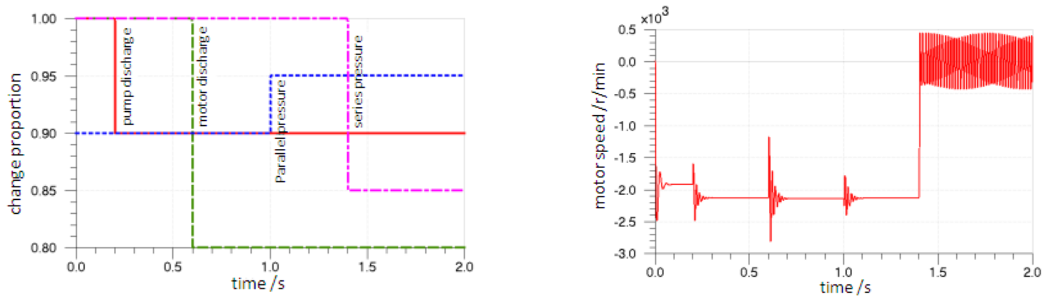
Table 1. Parameters of Simulation Model

Serial number	Factors name	Values
1	The maximum discharge of pump ml/r	200
2	The maximum discharge of motor ml/r	200
3	The rotation speed of pump r/min	5000
4	The rotation speed of motor r/min	5000
5	The transmission ratio of gearbox	0.031
6	The transmission ratio of reducer	32.7
7	The discharge of series compensation pressure ml/r	80
8	The discharge of parallel compensation pressure ml/r	80
9	The maximum pressure of the series relief valve MPa	72
10	The maximum pressure of the parallel relief valve MPa	35
11	Transmission ratio of the power box	5.64

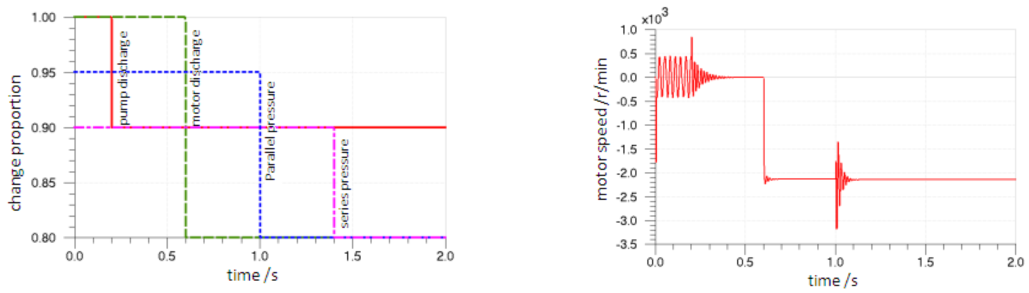
In Figure 6(b), different from Figure 6(a), there is a different phenomenon that when the pump discharge is below the motor while the parallel stable pressure control ratio is less than the series pressure proportional control ratio, the loading system is stable. When the parallel pressure is higher than the series parallel control pressure control ratio, even if the pump output fow is greater than that of motor, motor speed still appears zero oscillation. In Figure 6(c), it can be seen in the premise of loading pump and motor discharge is not equal, regardless of whether the relationship between the parallel compensation pressure control proportion and series compensation pressure control proportion, the system is stable in this case. Using the same method of analysis, it can be seen in Figure 6(d) and (e) with two different trends, system is in stable state in each stage and they have some similarities that is the parallel pressure control ratio is larger than that of series pressure.



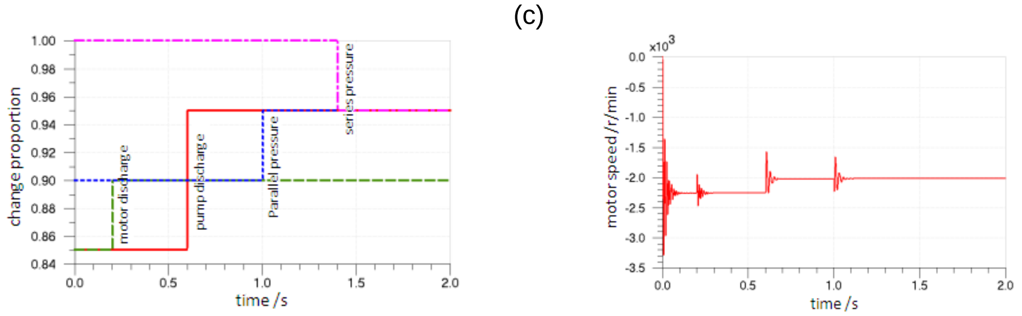
(a)



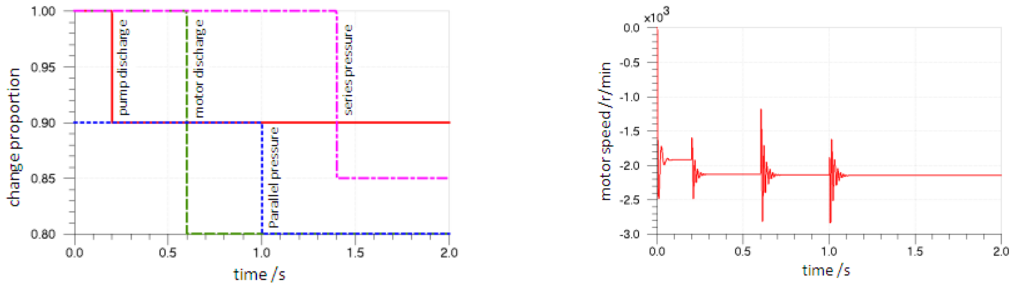
(b)



(c)

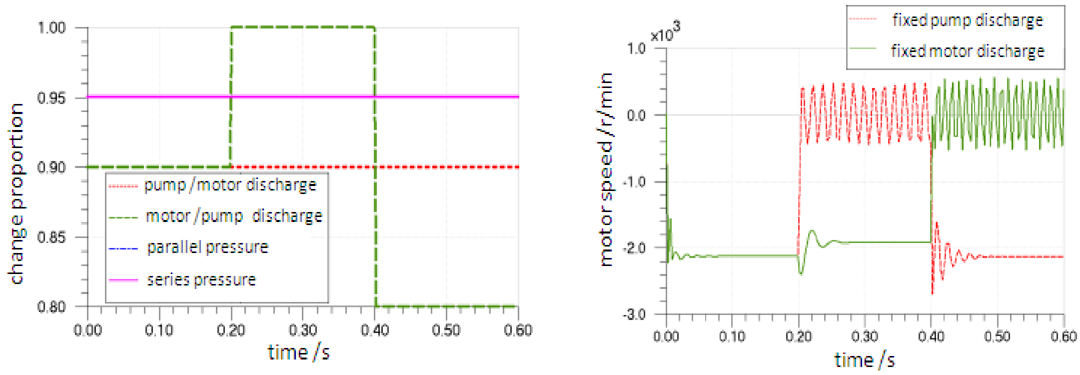


(d)

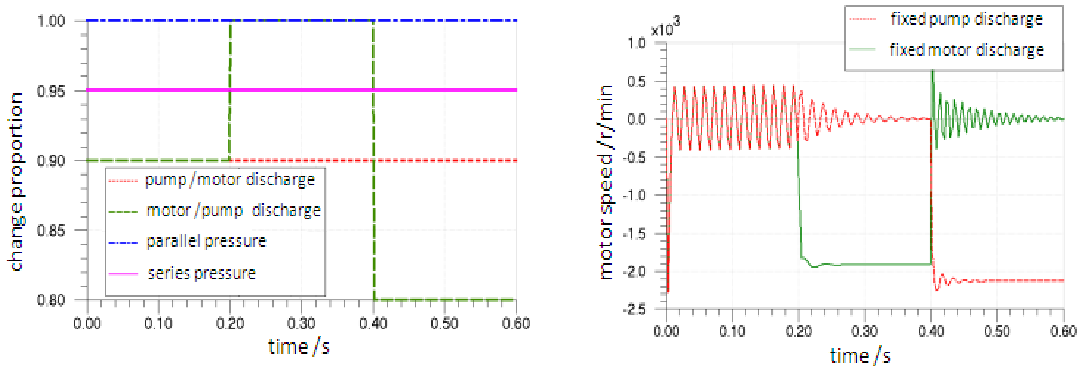


(e)

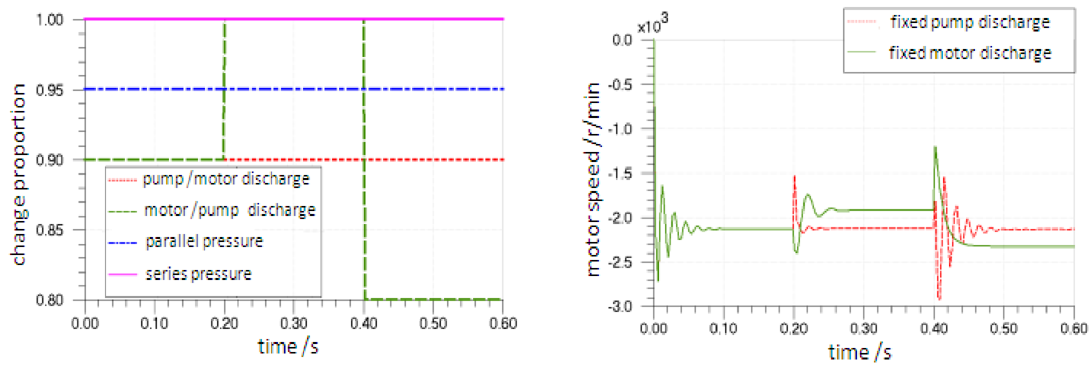
Figure 6. Analysis of Several Typical Control Conditions



(a)



(b)



(c)

Figure 7. Analysis of the Factors Affecting System Stability

Based on the above analysis, draw a conclusion: pump and motor output size and series parallel pressure control proportion size of the system don't have certain influence on the stability of the system imbalance. The reason of causing the system occur instability when a parameter mutation is parameters change trend, rather than numerical variation. Based on the previous work, the following plan summary to verify the influence factors is formulated. By fixed variables method considering the changing trend of pump and motor discharge and series-parallel pressure control proportion, it can analyze the influence of all the factors mentioned before on the stability of the system.

From the Figure 7, pump and motor output size and series parallel pressure control proportion size of the system don't have certain influence on the stability of the system imbalance. The reason of causing the system occur instability when a parameter mutation is parameters change trend, rather than numerical variation. As shown in Figure 7(a), it shows when the series and parallel pressure control proportion is equal, motor discharge suddenly exceeds pump discharge or pump discharge suddenly lower than motor discharge, motor speed decreases rapidly and appears oscillation at zero, which shows system instability. On the contrary, the system can immediately respond to a new stable state. In Figure 7(b), when the parallel pressure control ratio is less than series pressure control proportion, if and only if the motor output unchanged, pump rate changes from small to large or pump output unchanged, the motor output from large to small, motor speed can reach to a new stable state. In Figure 7(c), different from (b), no matter how the motor and pump discharge size change trend, as long as the parallel pressure control ratio is less than the series pressure control ratio, the whole system can still return to a new steady state in the parameter perturbation. In short, in order to make the system stable, the premise is the series compensation pressure control ratio is larger than the parallel compensation pressure control ratio and there is no necessarily linkage with the pressure difference.

5. Conclusion

Bond graph model and state equations of the torque loading system with single pump shows clearly the energy transfer logic relation between components in the energy feedback type power head torque loading system and the factors such as pressure, flow rate changes in the hydraulic system is the typical factors affecting the stability of the hydraulic energy closed-loop system. Influences of pump and motor output and series-parallel pressure control ratio on the system stability are achieved by AMESim simulation. By time-sharing segmentation of parameter variation tendency method and the combination of AMESim simulation and bond graph, it shows the effect of pressure discharge of the hydraulic system on the stability of the system and it verifies that when the series pressure control ratio larger than parallel pressure control ratio, the system keeps stable, and it has nothing to do with the motor and pump discharge output size. The discharge change trend of motor and pump should be considered and if and only if the discharge change trend of the pump is not less than that of motor, the system is stable. The using of AMESim silulation and comparison between the series and parallel pressure can directly prove the system stability to some extent. Through the analysis, the development of test beches can save much time and economic cost. The results can help the formulation and construction of the technology scheme of the power head torque loading system. This analysis provides reference for the regulation of the hydraulic equipments such as variable motor and variable pump and overflow valve. It makes preparation for the following torque loading tests.

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