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Research on new type of fast-opening mechanism in steam turbine regulating system and optimization of operation tactic*

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Abstract: With the analysis on regulating system in 200 MW steam turbine, the necessity of appending the fast-opening function to the original system is set forth and a new type of fast-opening mechanism is devised. The mathematical model of system is built up. With the use of AMESIM software, the displacement curve of the piston, the force curve of the cartridge valve spool, the pressure curve and the flux curve in the regulation process are obtained based on simulation. The performances of three fast-opening systems composed of cartridge valves with different diameters are compared. Based on the analysis on factors that affect the execution time of fast-opening, the dead zone of the fast-opening system is put forward. To overcome the defect, different operation modes are adopted for different zones. The result shows that with the increase of the valve diameter, the regulating time in the dead zone significantly exceeds the fast-opening time in the whole journey. Accordingly, the optimization operation tactic in the dead zone and the qualification conditions are brought forward. The fast-opening system composed of 32 mm cartridge valves is taken as an example with use of the tactic. The simulation result shows that the maximum regulating time is shortened by 509 ms.

Key words: Steam turbine, Fast-opening, Regulating system, AMESIM
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INTRODUCTION

Fast valving, regarded as an effective measure to keep the electric power system running stably, was put forward in 1929 (Park, 1973). However, the practical research did not start until 1980s (Kundur *et al.*, 2003; Patel *et al.*, 2002; Karady and Mohamed, 2002). Fast valving includes fast-closing and fast-opening controls of the valve. Though fast-closing can restrain the power angle fluctuation in the first cycle (Delfino *et al.*, 1988), it fails to eliminate the possible fluctuations that appear subsequently. So a function is needed to work with the fast-closing to restrain the re-fluctuation of the power angle. Generally the fast-closing function is set on the middle pressure regulating valve. For the stabilization of the unit, there is a 0.3~1 s delay for the reduction of the rotor mechanical energy after the valve is closed. Then the valve is opened for a reset journey in 2 s. If the middle pressure regulating valve is not opened in time, a motion of the axis and the damage of the trust bearing tiles will be caused.

At present, the research on the fast valving mainly focus on two aspects. Some experts adopt control arithmetic and advanced model, such as neural network, robust fault-tolerance control and nonlinear model predictive control, to obtain the optimum control time to improve the dynamic feature of the electric network in malfunction (Zhang *et al.*, 2000; Chen *et al.*, 1995; Yu *et al.*, 2002; Xiao *et al.*, 2005). Others concentrate on the improvement of the fast valving mechanism, such as using the parallel

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valves or the new style valve to realize the fast valving (Hassan *et al.*, 1999; Li and Wei, 2003).

In this paper a fast-opening system is devised using the cartridge valve for its features of much flux and short response time. After analyzing and predigesting the actual load, a model of the whole regulating system is built up using AMESIM. By analyzing every part of the fast-opening system, the dead zone of the fast-opening is found. Different operation modes are adopted for different zones. Besides, an optimization operation tactic in the dead zone is brought forward. The simulation result shows that the maximum regulating time in the dead zone is shortened greatly.

LOAD ANALYSIS AND PRINCIPLES OF THE FAST-OPENING MECHANISM

In this paper the regulating system of the reheating steam turbine (N200-130/535/535) is studied. Because the uninstall-valve is adopted in the original system, the unbalanced steam force is not only small, but also varies little in the whole process. Therefore the force can be ignored while analyzing the load which can be predigested as the spring-mass system (Fig.1).

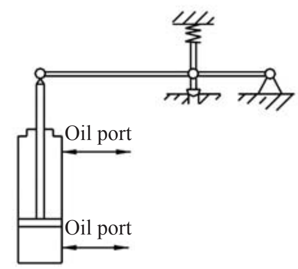


Fig.1 Graph of the system load

The servo regulation and the fast-closing function already exist in the original system. With the servo-proportional valve regulating the piston only, the moving time of the piston in the whole process will exceed 5 s in practice, since the diameter of the oil port in the servo-proportional valve is only 6 mm and the oil flux is small. Such a long time may lead to the second power angle peak within the second cycle, and even worse, losing step. Two cartridge valves and an electromagnetic switch valve are used to realize the fast-opening function. The oil circuit is shown in Fig.2. The pressure of the pilot oil cavities in the cartridge valve is controlled by the electromagnetic valve so that the opening and closing of the cartridge valves can be controlled through the electric signal. In order to guarantee the fast-opening performance, on one hand, the oil pump and the accumulator both supply oil; on the other hand, the drainage oil of the upper cavity in the oil servo motor is connected in differential mode to provide more oil to the nether cavity.

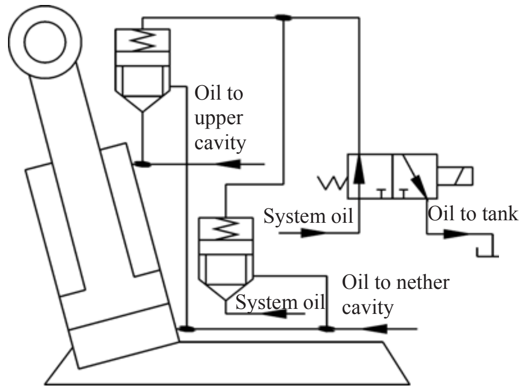


Fig.2 Design of the fast-opening system

MATHEMATICAL MODEL AND SIMULATION ANALYSIS OF THE FAST-CONTROL SYSTEM

The mathematical model of the regulating system is built up to analyze the characteristics of the fast-opening mechanism.

The servo-proportional valve is considered as the second-order form and its transfer function is:

$$W_{\mathrm{V}} = \frac{x_{\mathrm{V}}}{U} = \frac{K_{\mathrm{V}}}{S^2/\omega_{\mathrm{V}}^2 + 2\xi_{\mathrm{V}} S/\omega_{\mathrm{V}} + 1}, \tag{1}$$

where x_{V} is the displacement of the proportional valve spool; U is the input control signal; K_{V} is the proportion gain; S is complex variable; ω_{V} is the frequency response of the proportional valve; ξ_{V} is the hydraulic damping ratio.

The load-flux equation of the proportional valve assumes the following form:

$$Q_{\mathrm{L}} = K_{\mathrm{q}} x_{\mathrm{V}} - K_{\mathrm{c}} P_{\mathrm{L}}, \tag{2}$$

where Q_{L} is the load flux; K_{q} is the flux gain of the servo-proportional valve; K_{c} is the flux-pressure coefficient; P_{L} is the oil pressure of the load.

The load-flux equation of the oil servo motor can be expressed as:

$$Q_L = C_t(P_{down} - P_{up}) + \frac{V_{down}\dot{P}_{down} - V_{up}\dot{P}_{up}}{2\beta_e} + \frac{\dot{V}_{down} - \dot{V}_{up}}{2}, \tag{3}$$

where C_t is the leakage coefficient; P_{up}, P_{down} are the oil pressures of the upper and the nether cavities; V_{up}, V_{down} are the volumes of the upper and the nether cavities; β_e is the oil volume elasticity modulus.

Based on the force analysis, we have the force balance equation as follows:

$$m\ddot{x}_p = P_{down}A_{down} - P_{up}A_{up} - B_p\dot{x}_p - K_s x_p + F_z - f, \tag{4}$$

where A_{up}, A_{down} are the section areas of the upper and the nether cavities; B_p is the viscosity damping coefficient; x_p is the displacement of the piston; K_s is the elastic stiffness of the load; F_z is the component force of the piston gravity in the direction of motion; f is the friction between the piston and the oil servo motor; m is the mass of the load.

The flow equation of the upper cavity in the oil servo motor when fast-closing is:

$$Q_{up} = A_{up}\int \ddot{x}_p \mathrm{d}t = C_d A_p \sqrt{2(P_{sys} - P_{up})/\rho}, \tag{5}$$

where Q_{up}, Q_{down} are the flux of the upper and the nether cavities; C_d is the flux coefficient; A_p is the area of the passage; P_{sys} is the pressure of the oil of the system; ρ is the density of the oil.

The flow equation of the nether cavity in the oil servo motor when fast-closing is:

$$Q_{down} = A_{down}\int \ddot{x}_p \mathrm{d}t = C_d A_p \sqrt{2P_{down}/\rho}. \tag{6}$$

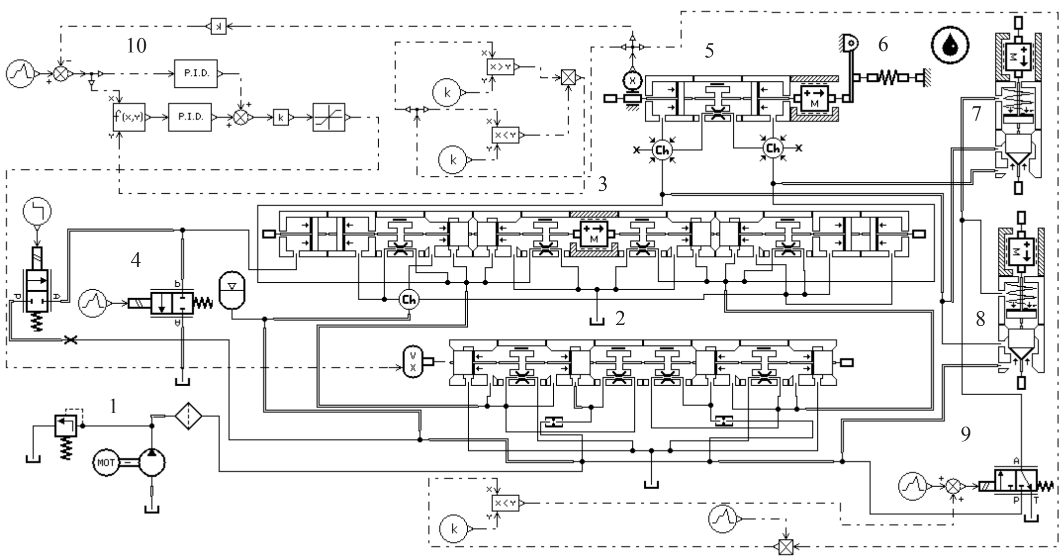
The flow equation of the upper cavity in the oil servo motor when fast-opening is:

$$Q_{up} = A_{up}\int \ddot{x}_p \mathrm{d}t = C_d A_p \sqrt{2P_{up}/\rho}. \tag{7}$$

The flow equation of the nether cavity in the oil servo motor when fast-opening is:

$$Q_{down} = A_{down}\int \ddot{x}_p \mathrm{d}t = C_d A_p \sqrt{2(P_{sys} - P_{down})/\rho}. \tag{8}$$

The AMESIM software of the IMAGINE corporation of France is adopted for the model building and the simulation of the hydraulic system. This software takes into account many factors, which are difficult to simulate in other softwares, such as friction, oil quality, circumstance temperature, etc. The whole model is shown in Fig.3. Normally, the



1: oil source; 2: servo-proportional valve; 3: oil servo motor valve; 4: switch valve for fast-closing; 5: oil servo motor; 6: system load; 7,8: cartridge valve; 9: electromagnetic switch valve; 10: integral-separated PI

Fig.3 Regulating system model building in AMESIM software circumstance

servo-proportional valve is used to support the oil in both the upper and the nether cavities of the oil servo motor. When fast-closing, the switch valve (4 in Fig.3) discharges oil at the bottom of oil servo motor valve (5 in Fig.3). After the falling of the valve spool, the nether cavity discharges oil quickly. The cartridge valves (7, 8 in Fig.3) are used to realize the fast-closing function.

The fast-opening system composed of 16 mm cartridge valve is taken as an example. Curve 1 in Fig.4 shows the process during which the servo-proportional valve is used to regulate the piston. The rise time is 5.6 s and the overshoot is 1.3%. It takes 6.5 s to attain stabilization. Curves 2 and 3 show the process of twice continuous fast-controls, both of which cost 2.4 s. The fast-closing of the whole process costs 0.4 s. After a time lag of 1 s, the fast-opening system begins to operate. It takes 0.98 s to complete the fast-opening of the whole journey and the overshoot is 2.3%.

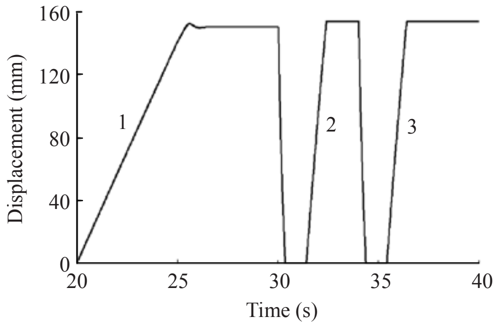


Fig.4 Displacement curve of the piston
1: servo-proportional valve regulation; 2,3: fast-control regulation

The forces of two cartridge valve spools are shown in Fig.5, in which Curve 1 and Curve 2 indicate the spool of cartridge valve 7 and cartridge valve 8 in Fig.3, respectively. It can be seen that the wallop is high both at the beginning of the fast-opening and at the end of the fast-closing. The instantaneous wallop of cartridge valve 7 reaches 375 N.

The flux curve of the upper cavity in the oil servo motor is shown in Fig.6. In the process of the servo-proportional valve regulation, the oil flux is 21.4 L/min. When the fast-closing is executed, the oil pump and the accumulator both supply oil in a short time. At 30.1 s, the instantaneous flux reaches 568 L/min. The flux in the process of the fast-opening is also great. At 32.8 s it reaches 128 L/min.

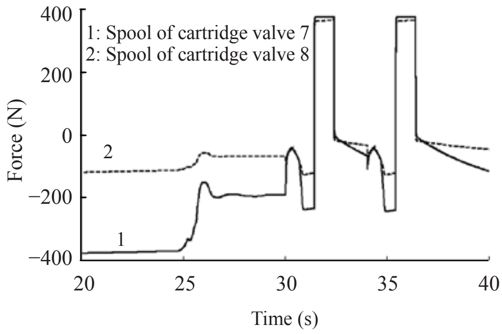


Fig.5 Force curve of the cartridge valve spool

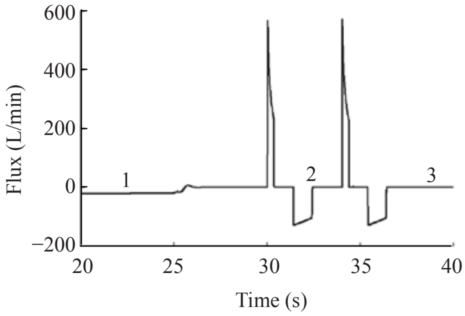


Fig.6 Flux curve of the upper cavity in oil servo motor
1: servo-proportional valve regulation; 2,3: fast-control regulation

The change of the oil pressure of the system is shown in Fig.7. At the beginning, the oil pressure is 2.0 MPa. It falls to 1.98 MPa in the process of the servo-proportional regulation (Curve 1). When the fast-control is executed, there is a remarkable decrease in the oil pressure because much flux is needed in a short time. The oil pressure falls to 0.68 MPa at the end of the fast-closing. At 32.38 s, the first fast-control comes to an end, while the oil pressure of the system is 1.95 MPa and fails to return to the normal value. So if the second fast-control begins right after that, it is unlikely to obtain the performance of the first one. From the result of the simulation, the oil pressure recovers completely at 32.48s, after that time the same performance of fast-control can be carried out. The change of the oil pressure of the upper and nether cavities in the oil servo motor is shown in Fig.8.

After simulation, the fast-opening time of the three systems is made up of 16 mm, 25 mm and 32 mm (in diameter) cartridge valves correspond to 0.98 s, 0.75 s and 0.68 s, and their overshoot is 2.3%, 3.5% and 4.2%, respectively. It shows that the fast-opening time decreases with the increase of the diameter of the cartridge. However, a larger diameter brings about a bigger dead zone in the system. We will address this problem in the following part.

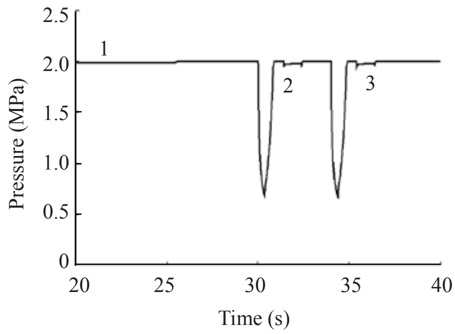


Fig.7 System oil pressure curve
1: servo-proportional valve regulation; 2,3: fast-control regulation

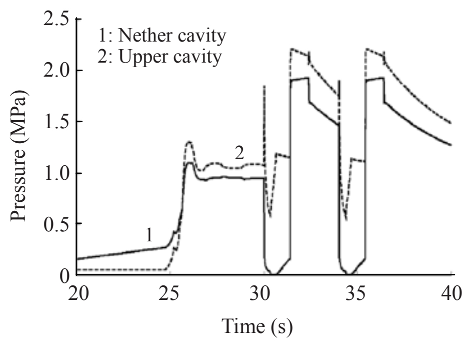


Fig.8 Oil pressure curves of the upper and the nether cavities in the oil servo motor

DELAYED RESPONSE TIME AND DEAD ZONE ANALYSIS OF THE FAST-OPENING SYSTEM

In practice, only after the response time of the middle relay, the switch valve and the cartridge valve, the oil would begin to flow into the nether cavity of the oil servo motor. So the delayed response time of the whole fast-opening system is approximately equal to the sum of that of these three parts. The response time of the middle relay is 10 ms and that of the switch valve is 25 ms under 20 Hz. As shown in (Lu, 2002), when the pilot flux is 8 L/min, the response time of the 16 mm, 25 mm and 32 mm (in diameter) cartridge valve is 28 ms, 67 ms and 110 ms, respectively. So the delayed response time of the three systems would be 63 ms, 102 ms and 145 ms, respectively (Assuming that the valve is in an ideal on-off state).

As shown in Fig.9, after the start delayed response time the system begins to execute the fast-opening. The minimum fast-opening time is zero. Then it comes into a stop delayed response time, which is fixed and equal to start delayed response time. Even if the fast-opening time is zero, the fast-opening system continues supplying oil for the nether cavity of the oil servo motor in the stop delayed response time. So after the stop delayed response time, piston has moved for some distance, which is the dead zone journey of the fast-opening system. Because the fast-opening system cannot make the piston move in the closing direction, the regulation in the dead zone won't be realized with it.

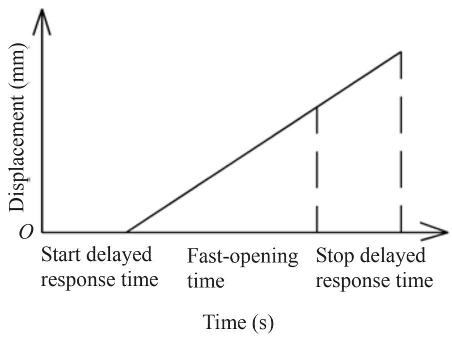
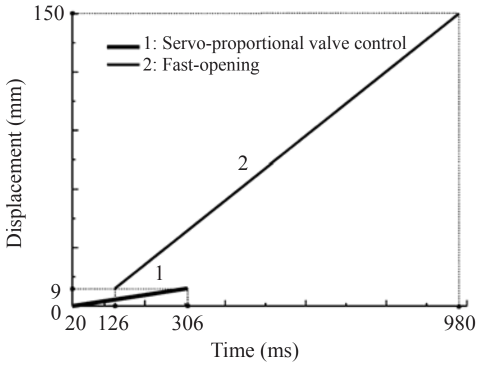


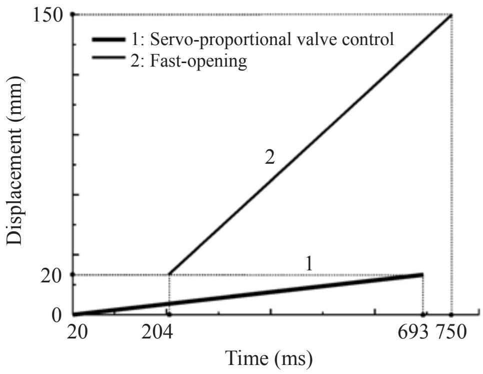
Fig.9 Analysis of actual fast-opening time

RESEARCH ON OPTIMIZATION OPERATION MODE OF THE FAST-OPENING SYSTEM

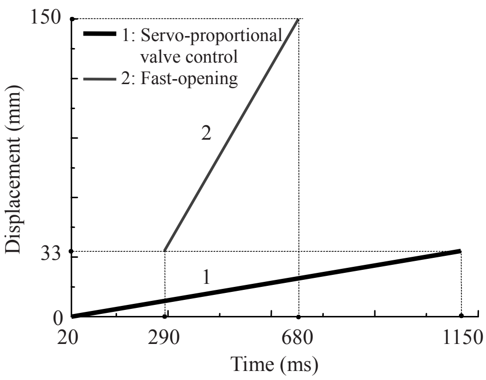
Due to the influence of the dead zone, different operation modes are adopted for different zones. If the reset journey is not in the dead zone, the fast-opening system can be used. Otherwise, the servo-proportional valve has to be used instead. Due to the small flux of the servo-proportional valve, it takes a relatively long time for the piston to complete the preset distance. Three fast-opening systems made up of 16 mm, 25 mm and 32 mm (in diameter) cartridge valves were simulated in this mode (Fig.10). Taking the 16 mm cartridge valve fast-opening system as an example, Curve 1 represents the dead zone of the fast-opening, and the corresponding displacement is 9 mm. The servo-proportional valve is used to adjust the piston in this zone and the maximum regulation time is 306 ms. Curve 2 is the fast-opening operation zone, from which we can see that the minimum time to perform the fast-opening is 126 ms and the time for the fast-opening of the whole process is 980 ms. As the three figures show, the fast-opening time decreases with the increase of the cartridge valve diameter. But the dead zone increases due to the increase of the cartridge valve response time, resulting in a longer time for the servo-proportional valve to regulate in the dead zone. When the diameter of the cartridge valve is 32 mm, the maximum regulating time in the dead zone exceeds the fast-opening time of whole process significantly.



(a)



(b)



(c)

Fig.10 Journey curve in the fast-opening system made up of 16 mm (a), 25 mm (b) and 32 mm (c) (in diameter) cartridge valves under zoning operation mode

In the situation that the target position is in the dead zone for most of the time, especially when the fast-opening system with large diameter cartridge valve is used, it takes a relatively long time to regulate the piston to the target position. In order to reset the system faster, an optimization operation tactic by which the fast-opening regulation cooperates with the servo regulation is put forward. A distance limit of the process is set. When the reset displacement is shorter than the limit, the servo-proportional valve is used (Curve 1 in Fig.11). Otherwise, the fast-opening (Curve 2 in Fig.11) is carried out within the minimum operation time (including the start and stop delayed response time only). After the response time (Curve 3 in Fig.11) of the servo-proportional valve, the piston begins to move in the reverse direction by the servo-proportional valve (Curve 4 in Fig.11) and the piston would move to a specific point above the limit. The difficulty of this operation mode is how to set the limit. Two conditions need to be satisfied:

$$s_1+s_2=s_d, \tag{9}$$

where s_1 is the piston displacement executed by the servo-proportional valve in the forward direction; s_2 is the displacement executed by the servo-proportional valve in the reverse direction after the fast-opening; s_d is the total distance of the dead zone.

$$t_1+20=t_2+t_{q\text{-}o}+20, \tag{10}$$

where t_1 is the regulation time in the forward direction by the servo valve; t_2 is the regulation time in the

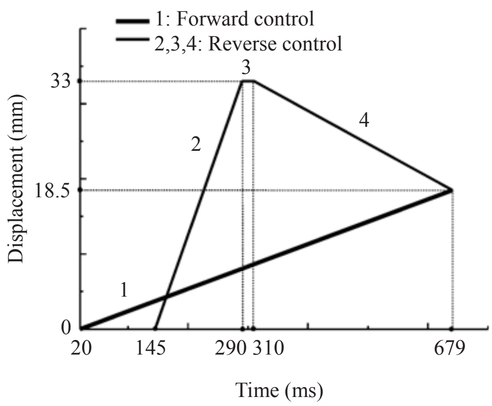


Fig.11 Optimization control curve of dead zone in the fast-opening system made up of 32 mm (in diameter) cartridge valve

reverse direction by the servo valve; $t_{q\text{-}o}$ is the minimum operation time of the fast-opening system.

Taking the fast-opening system made up of the 32 mm cartridge valve as an example (Fig.11), s_d is 33 mm and $t_{q\text{-}o}$ is 290 ms. The limit is calculated to be 18.5 mm. The regulation time of the piston taken to move to 18.5 mm in either forward or reverse directions is 641 ms. Compared to the operation mode in Fig.10c, the maximum regulation time in the dead zone is shortened by 509 ms.

CONCLUSION

The simulation results show that the fast-opening mechanism in this paper can drive the piston at a high speed and continuous fast-control can be executed after a short time interval. After comparing the three fast-opening systems made up of cartridge valves of different diameters, we find that with the use of larger diameter, the fast-opening time is shortened. By analyzing every part of the fast-opening system, we find the dead zone in which the piston cannot be regulated. Different operation modes are adopted for different zones. However, it takes a longer time to open the piston in the dead zone. As a result, an optimization operation tactic in the dead zone is brought forward. The simulation result shows that the maximum regulation time is shortened significantly. Our research is valuable for the over-speed protection of the steam turbine and the improvement of the transient stability of the electric net.

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