Moving Integrated Product Development to Service Clouds in the Global Economy
J. Cha et al. (Eds.)

doi:10.3233/978-1-61499-440-4-515

Stability Analysis and Optimal Design of Super-power Hydraulic Operating System

ZHANG Zenglei, WU Shijing[1], ZHAO Wenqiang, LAI Qiwei, HU Jicai
School of Power and Mechanical Engineering, Wuhan University, Wuhan 430072

Abstract. Running stability of hydraulic operating system with features of fast response, high flow rate and instantaneous super power greatly influences the normal and high effective action of ultra-high voltage circuit breaker. Mechanism of super-power hydraulic operating system in 1100 kV ultra-high voltage circuit breaker was analyzed to build a united simulation model, including multilevel control valves, accumulators, high-speed hydraulic cylinder buffering system and so forth, to investigate response features of control valves, buffering features of high-speed hydraulic cylinder and kinematic features of open and close action. Meanwhile, the accuracy of the model was validated with test data compared with design criterion, founding that there exist an extremely high pressure up to 107 MPa with a tremendous changing process within the hydraulic cylinder, and the velocity of the piston hitting the cushion sleeve at end of the cylinder was about 1.7 m/s which caused great impact on the sleeve. What's more, three optimal schemes for clearance distribution of the benched trunk piston were put forward to obtain a better cushion result reducing peak pressure and end speed of piston to a rate of more than 30%. Research methods, parameter analysis and optimal design results can act as guidance for further study and performance improvement of hydraulic operating system of ultra-high voltage circuit breaker.

Keywords. Hydraulic operating mechanism, Hydraulic cylinder, Stability analysis, Optimal design, Ultra-high voltage circuit breaker

Introduction

Serving as an important protection and operation device, ultra-high voltage circuit breaker carries out such tasks as controlling the grid transmission lines into or out of operation [1-2], cutting off faulty lines and so forth, in order to ensure normal reliable operation of the power system. Whereas, failure rate of operating mechanism among all failures of circuit breaker is the biggest, comprising about 43% of the major failures and 44% of the minor failures according to the statistical data from CIGRE [3-5]. Hydraulic operating mechanism is the only one successfully used in ultra-high voltage circuit breaker, owing to the super-large power needed and heavy current up to 63 kA, compared with other mechanisms such as magnetic operating mechanism, motor-driving mechanism, spring operating mechanism et al. [6-11].

Given this, it is necessary to study on running stability of the super-large power hydraulic operating mechanism of ultra-high voltage circuit breaker, to obtain dynamic characteristics of the mechanism and find corresponding problems. Then it is feasible to work out optimal schemes for the mechanism to improve stability of running process,

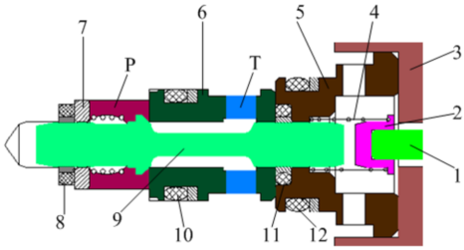
[1] Corresponding Author.

thus to guarantee the reliability of ultra-high voltage circuit breaker serving for improving transmission voltage level and constructing strong smart grid.

1. Mathematical model

1.1. Electromagnetic valve

Schematic diagram of electromagnet valve is shown in **Figure 1,** which is a direct action solenoid valve



1.ejector rod 2.spring seat 3.base 4.returning spring 5.valve sleeve 6.valve body 7.washer 8,10,11,12.sealing rings 9.valve plug

Figure 1. Schematic diagram of electromagnet valve

The constant high pressure port P is connected with return port T, when valve port of poppet valve is open with ejector rod pushing the plug moving left, to reduce pressure of controlling chamber in enlarging valve to zero bar. Spool of electromagnet valve is in the shape of cone, the schematic diagram of which is shown in **Figure 2** where *D* stands for diameter of inner bore, and *d* stands for diameter of rod, and *d*p stands for diameter of the plug end, and *da* stands for the effective diameter, and *Area* stands for the flow passing surface which is side face of a truncated cone, and *α* stands for half poppet angle.

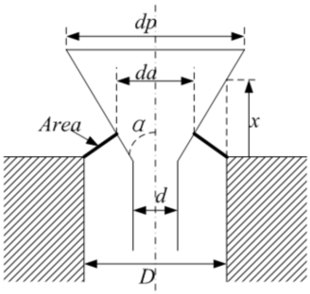


Figure 2. Schematic diagram of conical valve orifice

Flow equation of the valve port is defined as following [1-2],

$$Q_1 = C_{d1} A_1 \sqrt{\frac{2(p_0 - p_t)}{\rho}} \tag{1}$$

Area of flow-passing surface is defined as following,

$$A_1 = \pi \cdot x \cdot \sin\alpha \cdot (D - x \cdot \sin\alpha \cdot \cos\alpha) \tag{2}$$

where Q_1 is the flow rate of valve port, C_{d1} is the flow coefficient, A_1 is flow-passing surface, p_0 is pressure of port P, p_t is pressure of port T, ρ is density of oil, and x is opening of valve port

Force equilibrium equation of plug in electromagnetic valve is defined as following,

$$F_e - F_v - F_f - F_s = m_1 \ddot{x}_1 \tag{3}$$

$$F_s = C_{d1} \cdot \pi \cdot D \cdot x_1 \cdot \Delta p \cdot \sin 2\alpha \tag{4}$$

$$F_v = \frac{\pi \cdot D \cdot L \cdot v_1 \cdot \mu}{\Delta r} \tag{5}$$

where F_e is the acting force applied on spool by electromagnet, is viscous friction of oil, F_v is back spring force, F_f is steady state flow force, F_s is mass of spool, m_1 is differential pressure between inlet and outlet, Δp is contact length between valve spool and valve body, L is contact length between spool and valve body, x_1 is displacement of spool, v_1 is velocity of spool, μ is dynamic viscosity of oil, and Δr is fit clearance between spool and sleeve.

1.2. Main valve

Schematic diagram of enlarging valve is shown in **Figure 3**, where C3 stands for the pressure-constant chamber, port P is connected with head port of hydraulic cylinder, port T is connected with tank and port Z is connected with accumulator.

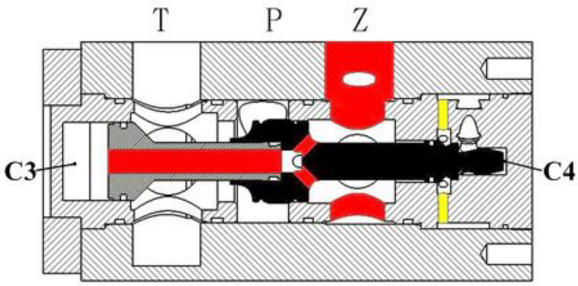


Figure 3. Schematic diagram of main valve

Flow equation of the valve port during open action is defined as following,

$$Q_3 = C_{d3} A_3 \sqrt{\frac{2\left(p_P - p_T\right)}{\rho}} \tag{6}$$

where Q_3 is flow rate of main valve, C_{d3} is flow coefficient of valve port, A_3 is flow-passing surface, p_p is pressure of port P, and p_T is pressure of port T.

Force equilibrium equation of plug in enlarging valve during open action is defined as following,

$$p_0 A_{c3} - p_0 A_z - p_{c4} A_{c4} - F_{v3} - F_d - F_{s3} = m_3 \frac{dx_3^{\ 2}}{dt^2} \tag{7}$$

$$F_d = C_{d3} \pi D_3 l \sqrt{2\rho \Delta p} v \tag{8}$$

where A_{c3} is effective area of spool in chamber C3, A_{c4} is effective area of spool in chamber C4, A_z is effective area of spool at port Z, p_{c4} is pressure of C4, F_{v3} is viscous friction, F_d is dynamic flow force, and F_{s3} is steady state flow force.

1.3. Hydraulic cylinder

Schematic diagram of hydraulic cylinder is shown in **Figure 4**, which is a differential cylinder with cushion plunger at the end of piston.

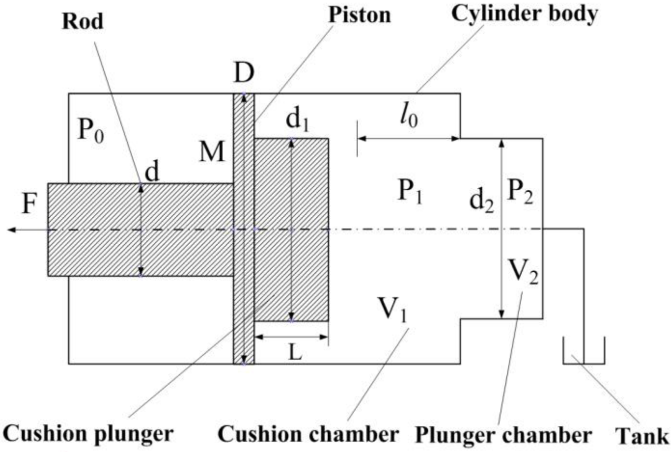


Figure 4. Schematic diagram of hydraulic cylinder

Force equilibrium equation of piston in hydraulic cylinder is defined as following,

$$P_0A_r - P_BA_p - F_c - F_\beta = m_4\frac{dx_4^{\ 2}}{dt^2} \tag{9}$$

where P_0 is pressure of rod port of hydraulic cylinder, P_B is cushion pressure, x_4 is displacement of piston, m_4 is equivalent mass of piston and drive mechanism, A_p is compression area of piston, A_r is compression area of rod port, F_c is load reaction force, and F_β is viscous friction

Flow continuity equation in open action is defined as following [12-17],

$$Q_4 = A_p\frac{dx_2}{dx} - \frac{V_B}{K}\frac{dp_B}{dt} \tag{10}$$

where V_B is volume of head rod port of hydraulic cylinder and K is bulk modulus of oil.

2. Results and discussion

The whole operating system simulation model, shown in **Figure 5**, including electromagnetic valves, enlarging valves, main valves, hydraulic cylinder and accumulator was established by connecting different hydraulic components which were built according to mathematical model above correspondingly, with appropriate pipe submodel.

Then the model was solved using parameters shown in **Table 1.** Meanwhile, test of physical prototype was implemented to obtain important characteristic curves, validating the accuracy of established simulation model.

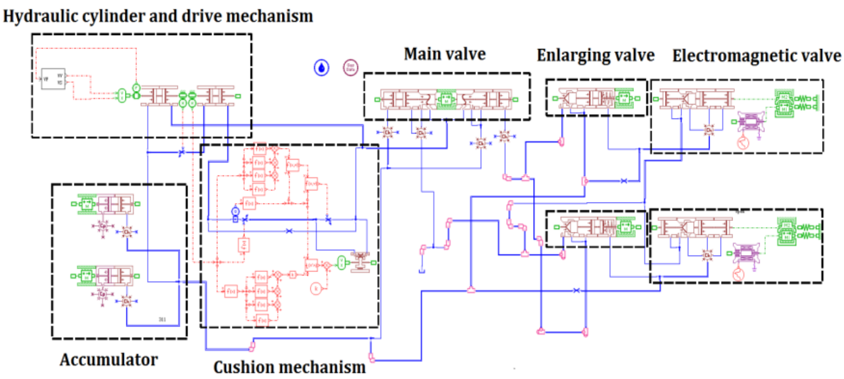


Figure 5. Simulation model of hydraulic operating system

Table 1. Parameters of simulation model

Parameter	Value
Coil turns of electromagnet N/tr	2580
Internal resistance $R/\,\Omega$	180
Stroke of ejector rod s_0/mm	3
Stroke of electromagnetic valve s_1/mm	1.5
Inner diameter of electromagnetic valve d_1/mm	6
Stroke of enlarging valve s_2/mm	3.2
Inner diameter of enlarging valve d_2/mm	9
Stroke of main valve s_3/mm	20.4
Inner diameter of inner valve d_3/mm	54
Underlap at open location of main valve l_1/mm	10.2
Underlap at close location of main valve l_2/mm	7.2
Stroke of hydraulic cylinder L_s/mm	230
Inner diameter of hydraulic cylinder D/mm	90
Diameter of piston D_r/mm	45
Volume of accumulator V_c/L	37.5
Precharged pressure of accumulator P_c/MPa	21.5
Inner diameter of main pipe d_0/mm	40
Rated pressure P_0/MPa	32.6
Operating temperature T_0/℃	20

Displacement curves of three level control valves are shown in **Figure 6** and **Figure 7.** The time needed for spool of three level control valves moving from one end to the other end are different during open and close action. Specifically, operation time of electromagnetic valve, enlarging valve and main valve are 8.4 ms, 7.6 ms and 10.4 ms, respectively during open action, while 8.4 ms, 8.2 ms and 10.4 ms, respectively during close action. It is obviously that movement of the main valve spool in open action is faster than that in close action due to different working principals of main valve between open and close action, because of that it is a pressure-losing process during open action and a pressure-rising process during close action on the contrary.

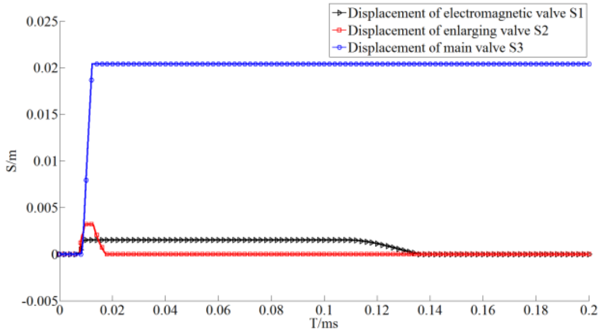


Figure 6. Response features of control valves during open action

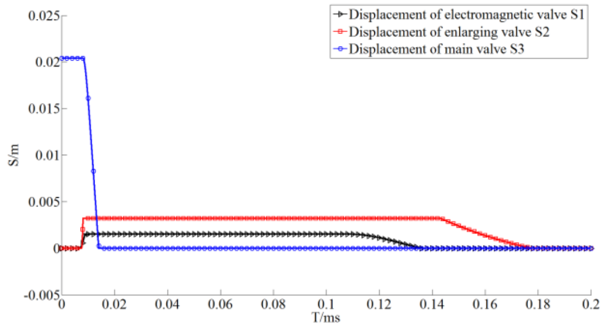


Figure 7. Response features of control valves during open action

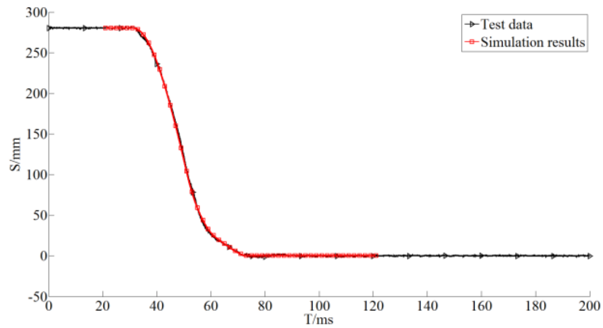


Figure 8. Displacement features of moving contact during open action

Of all the kinematic features, open time and close time, open velocity and close velocity, are two group important parameters related with the normal action of the system. According to the criteria of IEC, open time is the time needed from receiving actuating signal to the moment moving contact and fixed contact separated with each other. And open velocity is average speed within ten milliseconds after the two contacts separated [1-4]. From **Figure 8**, the two kinematic features can be calculated. Close time and close velocity can be defined according to the same criteria, respectively.

Specific kinematic features are calculated from simulation and test results shown in **Table 2**. It can be drawn from the results that the system simulation model built in **Figure 5** is of high accuracy.

Table 2. Results comparison of simulation and test

Parameter	Criteria	Test	Simulation
Open velocity m/s	12.5±1.0	11.68	11.97
Open time ms	20±2	20.8	18.86
Close velocity m/s	5.0±0.5	4.52	4.95
Close time ms	85±10	94	86.13

Apart from kinematic characteristics mentioned above, cushion feature of hydraulic cylinder is another target used to evaluate quality of operating mechanism. What is specially needed to be pointed out is that the designed open velocity is up to 12 m/s with a big equivalent mass up to 150 kilogram which will generate extremely high pressure causing great damage to cushion sleeve. Velocity characteristic of piston and pressure characteristic of hydraulic cylinder are shown in **Figure 9** and **Figure 10**, respectively. From the results it can be found that simulation results and test data are almost in according with each other. Final speed of piston hitting against cushion collar is about 1.7 m/s, and peak pressure of cushion chamber is up to 107 MPa, which are not ideal cushion results.

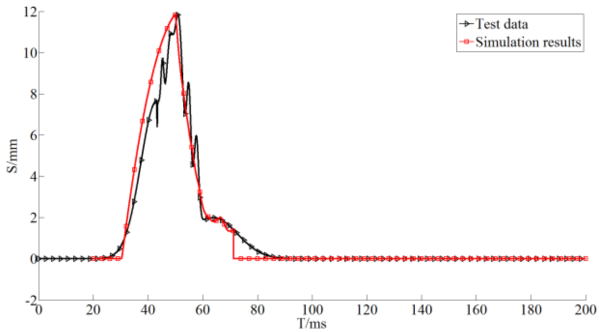


Figure 9. Velocity features of piston during open action

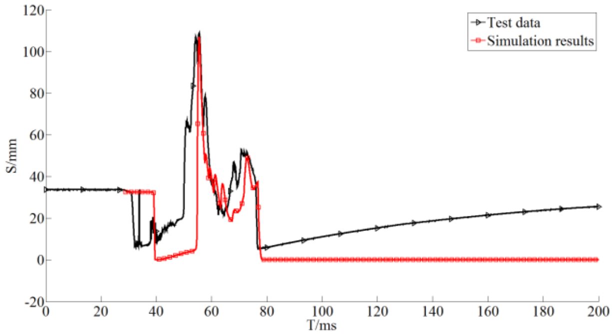


Figure 10. Pressure features of hydraulic cylinder during open action

3. Parameter analysis and optimal design

Characteristics of hydraulic operating system from simulation results are almost the same as that from test results. Meanwhile it is necessary to make optimal design of structure and parameter to promote performance of the system [18-20].

Structure and parameter of cushion plunger cause greatest influence on cushion performance. The final speed of piston and the peak cushion pressure of cylinder can be effectively limited by a reasonably designed cushion plunger which is type of multiple steps used to achieve the goal of gradient throttling.

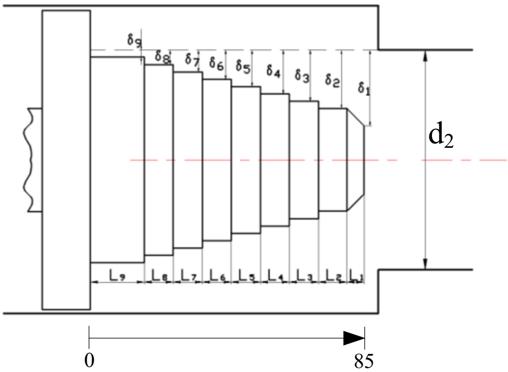


Figure 11. Schematic diagram of open action with benched trunk piston

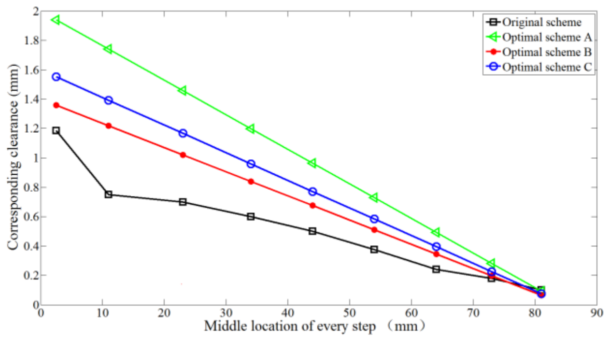


Figure 12. Clearance distribution along trunk piston of different schemes

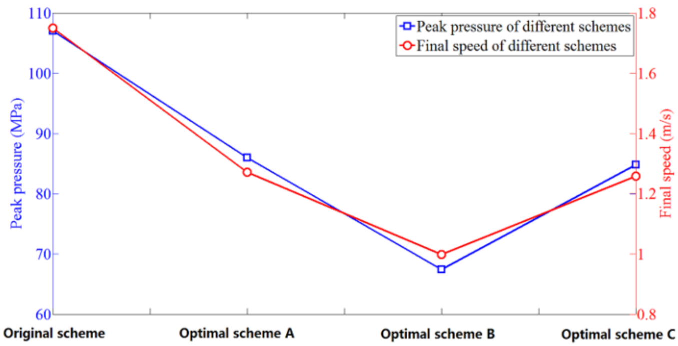


Figure 13. Comparison of cushion results between optimal scheme and original scheme

Clearances between nine level steps and cushion hole were studied to find a better clearance distribution. Schematic diagram of open action with benched trunk piston is shown in **Figure 11.** Three design schemes were obtained by varying different clearance between plunger and cushion hole after massive calculation, and clearance distribution along the plunger is shown in **Figure 12.** Cushion results of the three schemes were got using simulation model built above and compared with the original structure, which is shown in **Figure 13.**

From the results, it can be found that cushion performance of the three optimal schemes are better than that of the original scheme, limiting peak pressure within 90 MPa and final speed within 1.3 m/s. Specifically, scheme B is the best, reducing peak pressure to 67 MPa by as much as 37.38% and reducing final speed to 0.98 m/s by as much as 44%.

Comparing clearance distribution between scheme B and original scheme, it can be found that limiting the first two clearances and appropriately enlarging the other clearances can make transformation of kinetic energy and pressure energy more gently, which effectively limits the peak pressure in hydraulic cylinder and final speed of the piston. Nonetheless, it still needs test of prototype to validate scheme B and further research on influence of multiple-link transmission mechanism on cushion performance with a more accurate integrated simulation model.

4. Conclusion

The main function of a circuit breaker is to open up a circuit within a predefined time so as to prevent a sudden surge in current that could damage equipment with excessive high heat. This contribution presents an integrated model for 1100 kV ultra-high voltage circuit breaker, analyzing dynamic characteristics of the mechanism in open and close action with test data validating the model, to find the existing problems and make corresponding optimal design scheme to improve running stability of the mechanism. In the simulation model, flow rate equation and force equilibrium equation of corresponding spool of three level valves were accurately built, and cushion model of hydraulic cylinder was built according to the real benched trunk piston, but the precise process of cushion still need further research. Despite kinematic characteristics of simulation results agreeing with test data, cushion peak pressure and final speed of the piston are too high to ensure sealing performance of cylinder and small vibration of transmission system. Therefore, influence of structural parameter of cushion plunger on cushion process was studied to find a better combination of the diameter and the length. What's more, clearances of benched trunk piston were redesigned to find a better scheme of clearance distribution, effectively reducing peak pressure and final speed of piston. In future work, a more accurate united model considering multi-link transmission mechanism is to be built using a technique of co-simulation between AMESim and ADAMS in which the transmission mechanism is to be built taking into account clearance between adjacent links which is a complicated contact-impact dynamics problem, and the cushion process will be studied with method of CFD using steady analysis as well as transient analysis using technique of moving mesh. All the methods used and results obtained in this paper can provide guidance for stability analysis and optimal design of hydraulic operating system of ultra-high voltage circuit breaker.

References

[1] Xu Guozheng, Zhang Jierong, Qian Jiali, et al, *Theory and application of high voltage circuit breaker,* Beijing, Tsinghua University Press, 2000 : 1-12 (in Chinese).

[2] Yuan Shun, *Hydraulic operating system of high voltage circuit breaker*, Beijing, China Machine Press, 2000, 123-130 (in Chinese).

[3] Chen Jie, *Research on nonlinear characteristics of ultra-high voltage circuit breaker drive mechanism*, Wuhan,Wuhan University, 2012 (in Chinese).

[4] Wang Junwei. *Co-simulation of hydraulic operating mechanism of ultra-high voltage circuit breaker*, Wuhan, Wuhan University, 2012(in Chinese).

[5] Yang Wu, Rong Mingzhe, Wang Xiaohua, et al, Dynamic simulation of high-voltage circuit breaker concerning electrodynamic force, *Proceedings the CSEE*, 23(5) (2003) 103-107 (in Chinese).

[6] Liu Wei, Xu Bing, Yang Huayang. Characteristic analysis of high voltage circuit breaker with hydraulic operating mechanism, *Chinese Journal of Mechanical Engineering*, 46(10) (2010) 148-155 (in Chinese).

[7] Jin Zhaoming. *Hydraulic mechanics*, Beijing, National defense industry Press, 1994 : 223-227 (in Chinese).

[8] Zhang Jingshu, Chen Degui, Liu Hongwu. Dynamic simulation and optimum design of low-voltage circuit breaker , *Proceedings the CSEE*, 24(3) (2004) 102-107(in Chinese).

[9] Chang Guang, Wang Yi, Wang Wei, Mechanical fault diagnosis of high voltage circuit breakers utilizing zero-phase filter time-frequency entropy of vibration signal, *Proceedings the CSEE*, 33(3) (2013) 152-162 (in Chinese).

[10] Wang Lianpeng, Wang Erzhi, Investigation of joint simulation under no load breaking for hydraulic actuator and arc-quenching chamber of SF_6 circuit breaker, *Proceedings the CSEE*, 25(20) (2005) 118-123 (in Chinese).

[11] Wei Junmei, Lin Xin, Pressure character and mechanism character coupling computation of SF_6 high voltage circuit breaker, *Proceedings the CSEE*, 27(15) (2007) 118-123(in Chinese).

[12] Bartosz Rusek, Gerd Balzer, Timings of high-voltage circuit breaker, *Electric Power System Research*, 78(2) (2008) 2011-2016.

[13] Xiang Zhang, Ernst Gockenbach. Reliability estimation of high voltage circuit breaker by statistical analysis on the basis of the field data[J]. *Electric Power System Research*, 103(3) (2013) 105-113.

[14] Keli, M.A.Mannan, Electro-hydraulic proportional control of twin-cylinder hydraulic elevators, *Control Engineering Practice*, 9 (2001) 367-373.

[15] Chin-Kun Teng, Chao-Yin Hsiao, The effect of the guiding directions of the guiding cylinders of an area ratio modified machine cushion on the capability of suppressing impact and vibration, *Advances in Engineering Software*, 40 (2009) 991-999.

[16] Chen Fuchen, Dynamic response of spring-type operating mechanism for 69 kV SF_6 gas insulated circuit breaker, *Mechanism and Machine Theory*, 38 (2003) 119-134.

[17] Seong-Kwan Park, Kyong-Yop Park, Hi Jun Choe, Flow field computation for the high voltage gas blast circuit breaker with the moving boundary, *Computer Physics Communication*, 177 (2007) 729-737.

[18] Chen Fuchen, Tzeng Yih-fong, On the dynamics of spring-type operating mechanism for 69 KV SF6 gas insulated circuit breaker in open operation, *Computers and Structures*, 80 (2002) 1715-1723.

[19] Hao Qingchang, Yang Guodong, Development of the control system and communication network for intelligent SF6 high voltage circuit breaker, *Procedia Engineering*, 29 (2012) 1690-1696.

[20] Niu Xin, Zhao Xiaoxia, The study of fault diagnosis of the high-voltage circuit breaker based on neural network and expert system, *Procedia Engineering*, 29 (2012) 3286-3291.