MECHANICAL AND ELECTRO HYDRAULIC COUPLING DYNAMIC MODEL AND SIMULATIVE ANALYSIS FOR FULL HYDRAULIC DRILLING RIG

Guoqiang Geng
Henan University of Engineering, Henan 451191, China.
Email: guoqianggeng37485@126.com

ABSTRACT: This paper sets to research on the full hydraulic drill in order to further promote the development of drilling industry. Adopting the hybrid drive technology of horizontal directional drilling in deep underground, and signal acquisition and fault diagnosis system for hydraulic drilling, we constructed the coupling dynamics model and conducted simulative analysis. The results show that, thanks to research on relevant technologies, the output of the research models of the full hydraulic drill conform to the actual responses. And even when there is a sudden change to the load of the drill, the output speed of the drill will significantly decrease. In conclusion, the drilling speed is highly sensitive and subject to interferences.

KEYWORDS: CNC milling machine, hydraulic system, remote control system

1. INTRODUCTION

Full hydraulic drill is a typical key tool in the drilling industry. With this tool, theoretically, it should ensure smooth process of drilling. But in actual practices, the full hydraulic drill has very high energy consumptions, in addition, the drilling often takes place in challenging conditions, thus posing great challenges to the performance of the drilling machine. To ensure and improve the performance of the drill, it is important to conduct dynamic analysis of the drill rig, find the deficiencies of its operation, and propose improvement measures to remediate such deficiencies.

This paper first conducted a summary of the advantages of full hydraulic drills, and concluded that the drill has high drilling speed, is easy to operate on, and has steady performance. Based on these advantages as initial goals of research, we built a coupling dynamics model, and adopted the hybrid drive technology of horizontal directional drilling in deep underground, and signal acquisition and fault diagnosis system for hydraulic drilling in order to locate the deficiencies with the support of the simulative analysis.

2. LITERATURE REVIEW

Dombrowski invented the electric motor in 1888, and PixiiH made the permanent magnet generator in 1932, which marked the emergence of the electromechanical system. Ceccoli used the multi scale method to establish a dynamic model of mechanical coupling of a generator system. On the basis of the model, the resonance problem of the motion component under the action of reciprocating inertia force and self-excited inertia force was studied (Ceccoli, 2017). Gwen used the electromechanical disaster dynamics model to establish the dynamic model of the motorized spindle system and study the soft start characteristics of the system (Gwen, 2017). Junjiang and Min applied the translational excitation to stimulate the liquid in the cylindrical tank. The structure was studied theoretically and experimentally by the method of mechanical fluid combination, and a series of nonlinear dynamic phenomena were observed, such as the super-sum difference resonance and the amplitude frequency response (Junjiang and Min, 2017). Ortiz established a dynamic model for the flexible multibody system, considering the existence of the dynamic characteristics of the structure and the nonlinear factors in the liquid. Kyle and others studied the electrohydraulic micro-jet printing system under the action of DC voltage by electrohydraulic coupling method. By investigating the two parameters of solution concentration and voltage, the process of printing and deposition of the system were analyzed by mechanical and liquid misalignment (Kyle et al., 2017). Luo Fusheng studied the asymmetrical closed pump control system, and used the power bond graph to reconstruct the mathematical model of electro-hydraulic balance technology.

Electromechanical fluid mismatch is a typical multi field mismatch. As early as 1990, Sobiesczanski-Sobieski, a senior researcher of
nasa and American American mathematician of Poland, proposed a global sensitivity analysis method for the complex mechatronics system. Lebedev and Karabuta, based on the study of the structure and principle of handling manipulator, established the mechanical and electrical coupling dynamic model, and carried out the simulation analysis, and also studied the joint friction compensation and the precise trajectory tracking problem of the manipulator (Lebedev and Karabuta, 2017). Taking the six-degree-of freedom system of the robot as an example, Liao Daoxun established the optimization design model of the electromechanical hydraulic integration system, and improved the design efficiency of the robot by using the joint simulation technology.

Researchers at home and abroad made a deep study on the dynamics of drilling rigs. Nabo and so on, based on the multi-body dynamic method, studied and analyzed the dynamic simulation of the rotary motion of a crane type crane, and it was found that the rotation acceleration was the main factor affecting the dynamic characteristics of the crane (Nabo et al., 2017). Radoslaw and others used hydraulic simulation software to simulate the rotary hydraulic system of rotary drilling rig, analyzed the factors that had obvious influence on the performance of the system, and proposed an improved scheme (Radoslaw et al., 2017). Trent set up the mathematical model of the excavator based on the analysis of the structure and principle of medium excavator. The simulation analysis was carried out by AMESim and ADAMS software to obtain the change curve of the influence of the displacement, the moment of inertia and the deceleration ratio of the rotary motor on the spillway and the braking stationarity of the rotary platform (Trent, 2017). Based on the 3D modeling of bolting rig, Song Baoxin of Liaoning Technical University applied the multi rigid body dynamics and rigid flexible coupling dynamics module in ADAMS to analyze the dynamic response of the bolting rig, and obtained the maximum force on the hydraulic cylinder and the maximum force on the motion pair. Zhao Qian used the ADAMS software to simulate the dynamic simulation of the two typical working conditions of the virtual prototype of the SCG150 rubbering machine, and got the load change curve and the change curve of the contact force on the motion pair. Xiao Hua used AMES software and ADAMS software to carry out joint simulation and test verification of the rotary drilling rig, and applied the simulation results to improve the scheme. Finally, the effect of reducing the pressure fluctuation of the rotary drilling rig inlet pressure and reducing the high brake pressure was achieved.

With the development of modern digital network, the sources and types of data are more and more extensive, and the cost of obtaining data is getting smaller and smaller. A large number of data are produced in many fields. Due to the lack of effective data analysis tools, it is difficult to efficiently classify and deal with these data. In order to solve the problem of data redundancy, a new data analysis method - sensitivity analysis, came into being. Because of the superiority of sensitivity analysis method, it has been widely applied in many fields. Based on the classical two-mass spring damping dynamic model, Shen Qiang of Nanjing University of Aeronautics & Astronautics studied the sensitivity of the initial pressure of the low pressure cavity, the critical stroke position of the high and low pressure cavity conversion prop and the pre-pressure shrinkage of the tire. By applying the application matrix theory, a sensitivity analysis method with self-selected parameters was proposed by Zhang Jian and so on of Peking University, and the parameter sensitivity analysis and evaluation index were established in accordance with the actual working conditions. On the basis of developing a comprehensive analysis program for the overall design of airliner, Zhang Shuai, Nanjing University of Aeronautics & Astronautics, focused on the impact of the overall parameters on the performance of the airliner. Based on the principle of geometric control method, Xiong Shuzhang of Xihua University took a cross river bridge as the research object, and carried out the construction and sensitivity analysis of the main parameters such as structure weight, stiffness and other main parameters.

To sum up, through the continuous research and improvement of the domestic and foreign scholars, the analytical method based on the mechanical and electrical coupling dynamic model has been greatly developed. However, because of the mechanical, hydraulic and electronic control systems of the hydraulic drilling machine, the components of the mechanical, hydraulic and electronic control systems are many, and it is difficult to accurately express the relationship between the input and output of the model. The research on electromechanical fluid dynamics of drilling rigs at home and abroad remains to be further studied. Many scholars at home and abroad have studied deeply on the problem of complex electromechanical system. But the research point is to solve some serious and
obvious problems of misfortune and the research on the problem of multi input and multi output network calamity is seldom seen. As a result, there is a lack of design theory and method for the mechanism of global disaster coincidence. There are few studies on the sensitivity of system parameters by using the fully hydraulic electro-hydraulic kinetic model to find fault sensitive factors.

3. RESEARCH METHODS

3.1 Mechanical and Electro Hydraulic Coupling Dynamics Modeling Analysis for Full Hydraulic Drill

Following the research design of this paper, for the research on full hydraulic drill, it should start with the mechanical and electro hydraulic coupling dynamics modeling. To build this model, this paper adopted the load-sensitive value-controlled speed control system.

Load-sensitive value-controlled speed control system is a revolutionary technology in the area of hydraulic transmission control. In response to the complex and ever-changing controlled system, the load-sensitive value-controlled speed control system, synergizing with the automatic control techniques, can provide accurate, fast, and steady output of required hydraulic dynamics for the controlled system. In the load-sensitive value-controlled speed control system, the core component is the direct connection uniform-pressure-drop valve. The output of hydraulic pump is constant, which can be changed via the control of the uniform-pressure-drop valve. In the load-sensitive pump-controlled system, the core control unit is the load-sensitive variance ram pump, which controls the traffic of output along with the load on the execution parts. This paper utilized the load-sensitive valve-controlled speed control system.

Considering the operational steps of drilling techniques in practices, we categorized the drill working stages into two, one is non-loaded status, the other loaded status. The non-loaded status is the starting stage of the drill before the drill rod enters the rocks when the load is 0; while the loaded status takes place when the drill rod enters the rocks, and the load is the torque required to break the rock. Their simple load characteristics are indicated as below.

3.2 Torque Balance Equations of Retarder

First, the retarder is a part placed on the dynamics part in the head of the full hydraulic drill. Its working principle is to form primary transmission via two sets of spur gears, and to conduct the following 11 motion steps using the high pressure hydraulic oil to drive the cycloid hydraulic motor.

Rotate, the output shaft of the cycloid hydraulic motor (5) drives double slip gear (2) rotating on the spline shaft (6), operate the sliding gear handle (9) to change the meshing position between the double slip gear (2) and the big gear wheel (4) and small gear wheel (3) on the principal shaft (7). The principal shaft (7) can be speed-controlled with two methods. The hydraulic closed chuck (8) transmits the moment power to the drilling rod and provides the torque needed for rock breaking. Fig 2 illustrates the detailed transmissions.
For the purpose of modeling, to simplify the models, we had below consumptions: (a) the machining error, installation error of the gear box, gear and shaft were discarded, and the installation gap between the bearing and gear as well as the meshing gaps were also discarded; (b) the elastic changes of gear, shaft and bearing were discarded, namely consuming that the gear, shaft and bearing were all full rigid solids. The transmission of the retarder was simplified as the mass-damping system. Its structure illustrated as below.

![Figure 2: Transmission details](image)

3.3

3.4 Mathemetic Model of Load-sensitive Proportional Electromagnet Valve

The electric-mechanic conversion part in the proportional reversing valve of the load-sensitive valve is the proportional electromagnet. It converts the current signals into force or displacement. The dynamics of proportional electromagnet is determined by the transient process of the coil current, electromagnet adhesion and armature displacement. Taking into consideration that this paper focused on the gyration loop executed by the cycloid hydraulic motors, it concerns one oil-way. But the load-sensitive proportional multi-way reversing valve only provides sensitivity to multi oil-ways. Here we discarded the load-sensitivity of the proportional reversing valve, and only considered the deduction of mathematic model based on mechanical and electro hydraulic proportional controls in the load-sensitive valve.

The load-sensitive proportional reversing valve is inbuilt with a proportional electromagnet that controls the main valve power via turning electrical energy into mechanical energy. We only considered the mass of the armature and spool, and discarded influence of other interferences. Taking the output force of the electromagnet as the input signal into the load-sensitive proportional reversing valve spool, and the displacement of the spool as the output, we arrived at the model for the load-sensitive proportional reversing valve spool.

3.5 Validation scheme of model accuracy and simulation reliability

Validation scheme of the model accuracy and simulation reliability: The main purpose of this scheme is to verify the accuracy of the electro-hydraulic coupling dynamics model of the full hydraulic drilling rig and the reliability of the simulation results.

The experimental steps are: check whether all the experimental equipment is connected normally. Close the shut-off valve so that the motor is leak-free; remove the load; start the motor. After the motor starts stably, use the controller to control the load sensitive proportional reversing valve in the right position to make the motor turn forward; observe the torque meter. After the speed fluctuation is stable, add the load and continue to observe the torque...
meter. After the speed reaches stability again, turn off the motor power; save experimental data and sort out experimental equipment; analyze experimental data.

4. RESULTS AND ANALYSIS

4.1 Analysis on the Mechanical and Electro Hydraulic Coupling Dynamics Model

We have arrived at five basic equations of full hydraulic drill rig, namely the retarder torque balance equation, the electro hydraulic proportional valve proportional electromagnet model equation, the electro hydraulic proportional valve flow continuity equation, the hydraulic motor two-cavity flow continuity equation, and the hydraulic motor torque balance equation. In the five equations, all physical quantities were set as the variations in the initial conditions. The five equations determined the dynamics features of the drill rig gyration loop. Below are the five equations:

Equation 1: retarder torque balance equation

\[ T_i(s) = J_m s \omega_m(s) + B_m \omega_m(s) + \frac{T_1(s)}{i} \] (1)

Equation 2: the electro hydraulic proportional valve proportional electromagnet model equation

\[ I(s) = \frac{ms^2 + Ds + ks + ky}{K_1} X_v(s) \] (2)

Equation 3: the electro hydraulic proportional valve flow continuity equation

\[ Q_e(s) = K_q X_v(s) - K_v P_L(s) \] (3)

Equation 4: the hydraulic motor two-cavity flow continuity equation

\[ Q_L(s) = D_m \omega_m(s) + C_m P_L(s) + \frac{V_m}{\beta_e} s P_L(s) \] (4)

Equation 5: the hydraulic motor torque balance equation

\[ P_L(s)D_m = J_m s \omega_m(s) + B_m \omega_m(s) + T_i(s) \] (5)

Given that the mechanical transmission are regarded as full rigid transmission, the output angle of the hydraulic motor equals the input angle of the gear retarder. Therefore the input angle and output angle of the retarder fits in below relationship:

\[ \omega_m(s) = \omega(s), \omega_L(s) = \omega(s)/i, \quad i.e., \]

\[ \omega_m(s) = i \omega_L(s) \] (6)

Based on the five equations, we had the mechanical and electro hydraulic coupling dynamics model of the full hydraulic drill.

\[ \omega_L(s) = \frac{K_i}{\beta K_p D_m} \left( \frac{J_m s^2 + D_s + K_s + K_y}{s^2 + D_s + K_s + K_y} s + D_v + B_v \right) \] (7)

Simulative Analysis of the Mechanical and Electro Hydraulic Coupling Dynamics of the Full Hydraulic Drill

With the above calculations, we have arrived at the mechanical and electro hydraulic coupling dynamics model for the full hydraulic drills, as well as the transfer function when the full hydraulic drill is non-loaded and loaded, namely, when the machine is operating non-loaded, the relationship between the output speed and the load-sensitive reversing valve controlled current, and when the machine is operating loaded, the relationship between the output speed and change of load. Based on the calculation, selection and determination of the parameters for the mechanical system, electronic control system, and hydraulic system, we laid the foundation for simulative analysis conducted in the next step. Table 1 lists the parameter value used in the coupling dynamics model.

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Symbol</th>
<th>Parameter values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum flowrate of hydraulic system</td>
<td>Q_{max}</td>
<td>6 L/min</td>
</tr>
<tr>
<td>System pressure under no load air rotation</td>
<td>P_z</td>
<td>1.6 MPa</td>
</tr>
<tr>
<td>The equivalent inertia of the driving shafting</td>
<td>i</td>
<td>1.194</td>
</tr>
<tr>
<td>The equivalent inertia of the load shafting</td>
<td>J_1</td>
<td>0.041 kg.m²</td>
</tr>
<tr>
<td>Viscous damping coefficient of driving shafting</td>
<td>J_L</td>
<td>2.02 kg.m²</td>
</tr>
<tr>
<td>The equivalent inertia of the reducer</td>
<td>J_e = iJ_1/J_1/i</td>
<td>1.74 kg.m²</td>
</tr>
<tr>
<td>Viscous damping coefficient of drive shafting</td>
<td>B_i</td>
<td>0.5 N.m.s/rad</td>
</tr>
<tr>
<td>Viscous damping coefficient of load shafting</td>
<td>B_L</td>
<td>0.5 N.m.s/rad</td>
</tr>
<tr>
<td>The equivalent damping coefficient of the reducer</td>
<td>B_e = iB_i + B_i/i</td>
<td>0.61 N.m.s/rad</td>
</tr>
<tr>
<td>Current force gain of proportional electromagnet</td>
<td>K_i</td>
<td>0.55 N/mA</td>
</tr>
</tbody>
</table>
Proportional electromagnet zeroing spring stiffness $K_{sy} = 10 \text{ N/mm}$

Total stiffness of proportional electromagnet $K_Y = 24 \text{ N/mm}$

Quality of spool valve $m = 0.15 \text{ kg}$

Movement damping of the spool valve $D = 0.3 \text{ N/m/s}$

Flow gain coefficient of valve body $K_y = 0.0081 \text{ m}^2/\text{s}$

Flow - pressure coefficient of valve body $K_c = 1.2 \times 10^{-11} \text{ m}^2/\text{s} \text{ Pa}$

Bulk elastic modulus of fluid $\beta_e = 7.5 \times 10^8 \text{ Pa}$

Theoretical discharge of hydraulic motor $D_m = 2.5 \times 10^{-5} \text{ m}^3/\text{rad}$

Internal leakage coefficient of hydraulic motor $C_{im} = 0.58 \times 10^{-14} \text{ m}^3/\text{rad}$

External leakage coefficient of hydraulic motor $C_{em} = 0.35 \times 10^{-14} \text{ m}^3/\text{rad}$

Leakage coefficient of hydraulic motor $C_{tm} = C_{im} + 0.5 C_{em} = 1.28 \times 10^{-14} \text{ m}^3/\text{rad}$

High pressure chamber volume $V_m = 1.2 \times 10^{-3} \text{ m}^3$

Moment of inertia of hydraulic motor shaft $J_m = 6.7 \times 10^{-4} \text{ kg m}^2$

The viscous damping coefficient of the hydraulic motor $B_m = 0.5 \text{ N.m.s/rad}$

Equivalent inertia of the power head $J_d = iJ_m + J_e = 1.74 \text{ kg m}^2$

Equivalent damping coefficient of the power head $B_d = iB_m + B_e = 1.61 \text{ N.m.s/rad}$

MATLAB is an interactive large-scale software specially designed for scientific and engineering calculations produced by Math Works of the United States. It is a visual and powerful calculation tool that can perform various precise calculations and data processing. Simulink is a toolbox of MATLAB, which provides an integrated environment of dynamic system modeling, simulation and comprehensive analysis, enabling the digital simulation technology to enter a brand-new stage. It doesn't need to know too much about "numerical problems," but rather focuses on system modeling and analysis design. Simulink is closely integrated with MATLAB, which can directly access a large number of MATLAB tools for algorithm development and simulation analysis. The simulation model of the electro-hydraulic coupling dynamics model of the full hydraulic drilling rig is established with MATLAB/Simulink software, as shown in Figure 4.

![Figure 4: Simulink simulation model of mechanical and electro hydraulic coupling dynamic model of full hydraulic drilling rig](image)

In Figure 4, step 1 represents the No.1 input signal of the electro-hydraulic coupling dynamics model of the full hydraulic drilling rig, that is, the magnitude of the control current of the load-sensitive proportional multi-way reversing valve. Refer to the technical parameters of the selected load-sensitive proportional multi-way reversing valve, its control current is 0-610 mA, with the step function as input, and its corresponding curve is shown in figure 5.
Figure 5: The magnitude of the control current of the load-sensitive proportional multiple-way reversing valve

In Figure 5, Step 2 represents the No. 2 input signal of the electro-hydraulic coupling dynamics model of the full hydraulic drilling rig, that is, the load on the main shaft of the full hydraulic drilling rig (the reaction torque that the drill pipe receives when it breaks rock). Referring to the technical parameters of the full hydraulic drilling rig, the maximum output torque of the drilling rig is 500 N·m. In the simulation, the load value is 50N·m, and the step function is also input. The corresponding curve is shown in Figure 6.

Figure 6: The size of the load on the full hydraulic drill

According to the Simulink simulation model of the electro-hydraulic coupling dynamics model of the full hydraulic drilling rig, the response output of the model under the action of the dual input signal can be obtained, that is, the output speed of the main shaft of the full hydraulic drilling rig, as shown in figure 7.
It can be concluded from figure 7 that the full hydraulic drilling rig is in the state of stop operation within 0-1s, and the output speed of the main shaft is 0 r/min; in 1-2s, the full hydraulic drilling rig is in the no-load stage, and its output speed rises linearly. When the speed reaches 26r/min, the speed will fluctuate slightly; in 2-4s, under the action of viscous damping, the speed of the full hydraulic drilling rig is stable at 26r/min; within 4-5s, the full hydraulic drilling rig is in a loaded stage, its output speed drops sharply, and fluctuates up and down at a speed of 19r/min; within 5-7s, under the action of load and viscous damping, the speed of the full hydraulic drilling rig stabilizes again and remains at 19r/min. When the rig has a load, the output speed of the rig will decrease obviously.

### 4.2 Verification results of model accuracy and simulation reliability

The verification results of model accuracy and simulation reliability are shown in Table 2.

<table>
<thead>
<tr>
<th>State</th>
<th>Load (N.m)</th>
<th>Output speed (r/min)</th>
<th>Error</th>
<th>Simulation value</th>
<th>Experiment value</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unloaded</td>
<td>0</td>
<td>26.6</td>
<td>0%</td>
<td>26.6</td>
<td>23.3</td>
<td>12.41%</td>
</tr>
<tr>
<td>Loaded</td>
<td>50</td>
<td>55.6</td>
<td>11.2%</td>
<td>19.9</td>
<td>18.1</td>
<td>9.05%</td>
</tr>
</tbody>
</table>

In the no-load stage, the simulation value of the output speed of the drill is 26.6 r/min and the experimental value is 23.3 r/min, and the error between the two is 12.41%. The reasons for the error may be: after the detection element is added to the hydraulic system, the damping of the hydraulic system is increased, which leads to the decrease of system flow and the decrease of output speed; impurities appear in the lubrication of the main shaft bearing of all-hydraulic drill, which leads to increase of friction damping and decrease of output speed; after long-term operation of the hydraulic motor, the internal rotor and stator may wear, resulting in leakage of hydraulic oil and reduced output speed.

In the loaded stage, the simulation value of rig load is 50n.m and the experimental value is 55.6 N.m, and the error between the two is 11.2%. The cause of error may be the load caused by the crushing rock of the drill, which is determined by the rock hardness and feeding speed of the drill and has certain uncontrollable factors. The simulation value of the output speed of the drill is 19.9 r/min, and the experimental value is 18.1 r/min. The error between the two is 9.05%, and the reason for the error may be: it contains the
cause of the no-load phase; in the experiment, the load is larger than that in the simulation, resulting in the reduction of rotating speed.

According to the comprehensive analysis, the experimental results of the all-hydraulic drill are basically consistent with the model simulation results. It can be determined that the mechanical and electro hydraulic coupling dynamic model of full hydraulic drilling rig is correct and the simulation results are reliable.

5. CONCLUSION

Full hydraulic drilling rig is a classic complex mechanical, electro and hydraulic equipment that integrates mechanics, electricity control, and hydraulic pressure. This paper firstly conducted research analysis of the full hydraulic drilling rig, and set to understand the structure and working principle of the drilling rig and gave detailed description of the major structures of the drilling system, such as load-sensitive proportional multi-way reversing valve, cycloid hydraulic motor, retarder, feeding device, gripper, and programmable controller. Secondly, with the building of simulative model, we analyzed the drilling speed of coupling dynamics for the full hydraulic drilling rig.

The full hydraulic drilling rig has two control systems, the valve control and pump control. Due to the limitations, this paper only focused on the load-sensitive valve control system for the purpose of modeling for mechanical and electro hydraulic coupling dynamic model of full hydraulic drilling rig. To comprehensively understand the features of the full hydraulic drills, it calls for modeling for the pump-control system, and comparative research of two models. In addition, this paper only selected 5 principle parameters for analysis instead of all the parameters for parameter sensitivity analysis. To fully understand the responsiveness features of the full hydraulic drilling rig, the sensitivity analysis should be conducted on all the parameters for the full hydraulic drilling rigs.

6. REFERENCES

Ceccoli S., 2017, Explaining attitudes toward u.s. energy extraction: offshore drilling, the keystone xl pipeline, and hydraulic fracturing, Social Science Quarterly, 99(2), 644-664, DOI: doi.org/10.1111/ssqu.12447


