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Research on the Structural Design and Hydraulic Characteristics of the Intelligent Drilling Device for Drilling Machines

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Abstract

Mining hydraulic anchor drilling truck is a kind of high-pressure liquid as the power to carry out deep hole drilling, gas monitoring, and anchor rods after the coal mine production. The hydraulic drill trucks on the market at present can basically meet the production demand in terms of function, but there is still a lot of room for improvement in the replacement of drill rods. Based on this, this paper designs a rod changing device, mining hydraulic drilling car drill rod automatic loading and unloading system is the auxiliary equipment of hydraulic drilling car, which can automatically carry out the loading and unloading of drill rods and reduce the labor intensity of workers. At the same time for the device hydraulic servo system anti-interference ability is poor and other issues, designed a sliding mode variable structure controller, derive its valve motor mathematical model, and with MATLAB and amesim joint simulation verification, the results show that the sliding mode control effect is good.

Keywords

Fuzzy control; Anchor drilling rigs; Valve-controlled motors; Sliding mold variable structure.

1. INTRODUCTION

China is a large coal producing and consuming country, and coal occupies an important position in China's energy consumption and energy security. In recent years, coal mine underground roadway excavation using synthesized excavator or blasting and other methods of sectioning method of low efficiency, and easy to cause safety hazards, can not guarantee the safe and efficient production of coal mines, and gradually eliminated by the major coal mining enterprises. As an important part of the integrated technology of digging and anchoring, the anchor drilling rig has constraints on the speed and safety of roadway digging, but the traditional hydraulic anchor drilling rig has many shortcomings, such as: its impact cycle and energy can not be automatically adjusted, and it can only rely on manual adjustment by artificial experience[1-3], which is very poorly adapted to different geological conditions. In addition, the automation degree of the drilling rig is not high, the common hydraulic anchor drilling rig is controlled by relay-contactor and other logic, not only the circuit is complicated, but also the operation reliability is poor, and the later maintenance work is difficult.

Domestic researchers have been paying great attention to the research of anchor drilling car, and on the basis of fully absorbing foreign experience as well as their own exploration and practice, in recent years, a major breakthrough in technology has been realized and obvious results have been achieved. Among them, Meng Chuanming used finite element analysis software ANSYS/LS-DYNA to simulate the rock-breaking process of the anchor drill bit,

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analyzed the process of coal rock crushing and the force of the anchor drill bit, and summarized the rock-breaking process when drilling. In addition, it is also found that the speed of the anchor bit changes once in each rotary cycle under the influence of the number of drilling wings of the drill bit, which indicates that the number of drilling wings of the drill bit affects the working stability of the anchor bit. Anhui University of Science and Technology in China developed a monorail crane anchor drilling rig, mainly composed of operating platform and monorail crane, the operating platform to install two drilling rigs and the operating platform bottom plate using a slide rail connection, free sliding, for roof and side plate support.

But in the narrow tunnel with less space, the small hydraulic drilling truck is the best choice, however, this kind of small drilling truck itself is small in size and compact in structure, the existing automatic loading and unloading drilling rod device is not suitable for the modification of this kind of small drilling truck, and the flexibility of the modified drilling truck will be affected; moreover, in the process of construction of the drilling truck, a large amount of debris, dust and oil will be generated, especially in the construction of the mine, the crushed cinder with the water or hydraulic oil mixed with stains, the complex structure of the mechanism is easier to be affected, in the process of changing rods if the drill rod library card rod, will affect the efficiency of the work. Therefore, a kind of automatic loading and unloading system of drill rods for hydraulic drill truck with simple structure, small space occupation, and suitable for small coal mine or tunnel needs to be developed urgently.

2. STRUCTURAL DESIGN AND MODELLING OF AUTOMATIC ROD CHANGERS

For the anchor drilling machine automation degree is not high, workers manually drilling safety is too poor, this paper designs a small automatic pole changing drilling device, can realize can realize robot drilling, on the anchor and install anchor, and to be able to effectively improve the efficiency of the coal mine underground work, reduce the labour intensity of the underground staff, and improve the degree of automation of the underground operation.

Aiming at the problems of low automation of anchor drilling rig and poor safety of manual drilling by workers, this paper designs a small automatic pole changing drilling device, which is capable of automatic pole loading, unloading, loading and pole changing and other functions. It realizes the integrated construction of underground anchor drilling and support, saves labor time cost, reduces the difficulty of workers' construction and improves the safety and automation of underground operation.

Since this device is based on the[6] original anchor drilling rig for automation, the overall structure must match the existing mechanical structure of the anchor drilling rig, without too much change and without interference. The overall structure after the modification is shown in Figure 1. In the figure, the feed beam is used for downhole drilling; the drill rod box is used to store the anchor rods and feed them into the claws of the manipulator; the manipulator delivers the anchor rods to the position after the rotation of the drill box, and then clamps the anchor rods to feed them into the feed beam; the connecting arm fixes the rod changing device to the anchor drilling rig, and the power head is used to drill the holes.

Compared with the traditional relay control which can only carry out switching control, PLC applies microelectronics and computer technology to carry out both switching control and analog control, and it can also be linked with computer network to realize hierarchical control. The introduction of PLC control in the traditional field of hydraulic and pneumatic pressure, you can give full play to the programming advantages of PLC, the use of minimal investment to achieve complex control, while the high reliability of the PLC ensures that the hydraulic and pneumatic system has a high degree of stability. At the same time, the high reliability of PLC ensures the high stability of the hydraulic and pneumatic system. This makes the design of PLC-based hydraulic control robot simple, stable and reliable. The intermediate detecting elements

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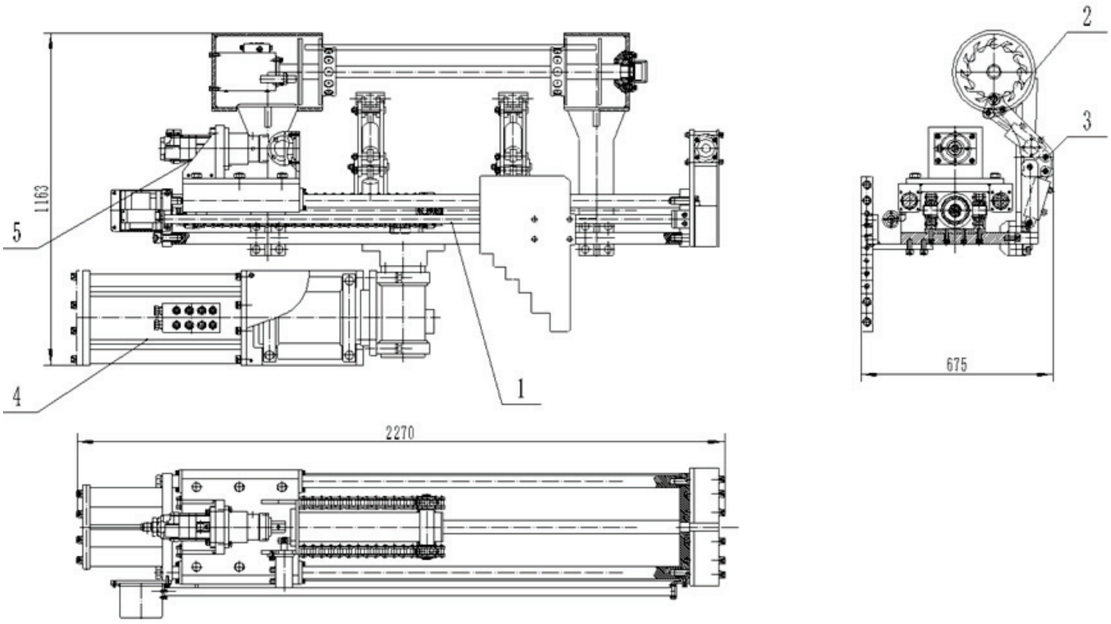
chosen in this paper are the rope displacement sensor to monitor the displacement of the feed cylinder, the absolute value rotary encoder for the angle of rotation of the storage box, the proximity switch to monitor the anchors in place or not, and the pressure transmitter to detect the pressure change of the cylinders at each place.



1. Feed beam 2 Drill rod box 3 Manipulator 4 Connecting arm 5 Power head
Figure 1. Overall two-dimensional structure

The specific drilling workflow of the rod changing device is as follows: firstly, manually install the first anchor rod at the power head and install the drill bit at the top plate of the anchor rod; then the feed cylinder carries out hole alignment, the power head rotates, and drilling is carried out; after drilling of the first rod is completed, the gripper clamps the bottom of the anchor rod, and the feed cylinder is returned; the rod storage box rotates by 30°, and sends the anchor rod into themanipulator's claw, and the claw clenches the anchor rod; after that, the manipulator moves and feeds the rod into the power head; the power head buttresses the anchor rod, followed by drilling.

The unloading process is just the opposite: when the last rod is finished, the power head backs up in positive rotation, and the clamp loosens; when the first rod is pushed out of the hole, the clamp clamps the second rod at the end; the power head continues to back up in reverse rotation, separating the two anchor rods; when the power head is returned to the parallel part of the manipulator, the manipulator comes from the rod storage box to clamp the rods, the power head continues to back up to the initial position, separating the rods from the anchor rods, and the manipulator swings back to the rod storage box, separating the anchor rods from the rods. back to the storage rod box, send the anchor rods to the storage rod box, the storage rod box reverses 30° to leave the next empty position.

3. MATHEMATICAL MODELLING

Hydraulic motor according to the actual engineering needs of the choice of cycloidal hydraulic motor, at the same time, in order to facilitate the calculation of modelling, assuming that the servo valve and the motor of the oil circuit between the pressure loss of zero, the motor oil temperature and the volume of the elastic modulus is a constant, the motor inside and outside the leakage of laminar flow flow.

$$Q_L = K_q X_v - K_c P_L \tag{1}$$

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Where QL is the load flow rate of the hydraulic motor, m^3/s; Kq is the flow rate gain coefficient; Xv is the spool displacement of the servo valve, mm[4-5]; Kc is the flow rate pressure gain coefficient, m^3/pas; PL is the external load pressure, Mpa.

$$Q_L = D_m s\theta_m + C_{tm}P_L + \frac{V_t}{4\beta_e} sP_L \tag{2}$$

It is inevitable that hydraulic motors will have leakage problems in the actual working motion. Therefore, the motor load flow rate should be the sum of the theoretical flow rate of the motor, leakage flow rate and compressibility flow rate. The leakage flow of a hydraulic motor consists of the external leakage to the outside of the motor and the leakage through the gap to the internal oil chamber.

where Dm - motor displacement, L/r; θm - angle of rotation of the output shaft, °; Vt - equivalent total volume of the chambers and piping on both sides of the motor, m^3; Ctm - leakage coefficient, m^5/N · s; βe is effective volume Modulus of elasticity, Pa.

Neglecting the static friction and other non-linear friction and the quality of the oil, taking into account the inertial load of the motor shaft, the elastic load in motion and the applied loads, combined with Newton's second law to obtain the motor torque balance equations as follows

$$P_L D_m = J_t s^2\theta_m + B_m s\theta_m + G\theta_m + T_L \tag{3}$$

where Jt - equivalent rotational moment of inertia of the output shaft of the motor and the external load, kg · m^2; TL - external load torque acting on the motor shaft, N · m; Bm - viscous damping coefficient; G - spring stiffness of the load, N/mm.

The above equations (1)-(3), eliminating the intermediate variables, can be obtained from the motor output shaft angle formula:

$$\theta(s) = \frac{\dfrac{K_q}{D_m}X_v(s) - \dfrac{K_{ce}}{D_m^2}\left(1+\dfrac{V_t}{4\beta_e K_{ce}}s\right)T_L(s)}{\dfrac{J_t V_t}{4\beta_e D_m^2}s^3 + \left(\dfrac{J_t K_{ce}}{D_m^2}+\dfrac{B_m V_t}{4\beta_e D_m^2}\right)s^2 + \left(1+\dfrac{B_m K_{ce}}{D_m^2}+\dfrac{GV_t}{4\beta_e D_m^2}\right)s + \dfrac{GK_{ce}}{D_m^2}} \tag{4}$$

where Kce is the total flow-pressure coefficient (m3 /(s · Pa)) and Kce=Kc+Ctm.

When the effect of external load stiffness is not taken into account, the external load is considered to be rigidly connected to the motor, so by taking G = 0, which is usually much less than 1, the above equation can be simplified.

$$\theta(s) = \frac{\dfrac{K_q}{D_m}X_v(s) - \dfrac{K_{ce}}{D_m^2}\left(1+\dfrac{V_t}{4\beta_e K_{ce}}s\right)T_L(s)}{s\left(\dfrac{s^2}{\omega_h^2}+\dfrac{2\xi_h}{\omega_h}s+1\right)} \tag{5}$$

Since the action of sending anchor rods from the storage box to the manipulator requires precise position and smooth movement, the movement process is indirectly controlled smoothly by controlling the angular velocity of the hydraulic motor. The hydraulic motor angular velocity control transfer function is:

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$$\omega(s)=\theta_{\mathrm{m}}s=\frac{\frac{K_{\mathrm{q}}}{D_{\mathrm{m}}}X_{v}(s)-\frac{K_{\mathrm{ce}}}{D_{\mathrm{m}}^{2}}\left(1+\frac{V_{\mathrm{t}}}{4\beta_{\mathrm{e}}K_{\mathrm{ce}}}s\right)T_{\mathrm{L}}(s)}{\frac{s^{2}}{\omega_{\mathrm{h}}^{2}}+\frac{2\xi_{\mathrm{h}}}{\omega_{\mathrm{h}}}s+1} \tag{6}$$

Where, ω_{h} −undamped hydraulic intrinsic frequency, rad/s ,$\omega_{\mathrm{h}}=\sqrt{\frac{4\beta_{\mathrm{e}}D_{\mathrm{m}}^{2}}{J_{\mathrm{t}}V_{\mathrm{t}}}}$; ξ_{h}- hydraulic damping ratio (dimensionless). $\xi_{\mathrm{h}}=\frac{K_{\mathrm{ce}}}{D_{\mathrm{m}}}\sqrt{\frac{J_{\mathrm{t}}\beta_{e}}{V_{\mathrm{t}}}}+\frac{B_{\mathrm{m}}}{4D_{\mathrm{m}}}\sqrt{\frac{V_{\mathrm{t}}}{J_{\mathrm{t}}\beta_{e}}}$, In the hydraulic system, the general load viscosity coefficient B_m is relatively small, can be ignored, so can be expressed in the following formula $\xi_{\mathrm{h}}=\frac{K_{\mathrm{ce}}}{D_{\mathrm{m}}}\sqrt{\frac{J_{\mathrm{t}}\beta_{e}}{V_{\mathrm{t}}}}$.

In establishing the electro-hydraulic servo valve transfer function, the effect of the servo valve on the control system cannot be ignored, so it is simplified to a second-order oscillatory link:

$$G_{\mathrm{v}}(s)=\frac{Q(s)}{I(s)}=\frac{K_{\mathrm{q}}}{\frac{s^{2}}{\omega_{\mathrm{v}}^{2}}+\frac{2\xi_{\mathrm{v}}}{w_{\mathrm{v}}}s+1} \tag{7}$$

Where ωsv is the equivalent undamped self-oscillation frequency (rad/s) of the electro-hydraulic service valve; ξsv is the equivalent damping coefficient of the electro-hydraulic servo valve (dimensionless).

Since the frequency band of the servo amplifier is higher than the hydraulic intrinsic frequency, the dynamic performance of the servo amplifier is ignored when performing the system design, so the servo amplifier is simplified to a proportional link, then

$$\frac{I(s)}{U(s)}=K_{a} \tag{8}$$

In the equation, Ka is the gain of the amplifier, A/V.

After consulting the relevant manuals and selection calculations can be obtained, the motor displacement for Dm = 3.91x10-5 m3/rad, the oil source pressure of 21MPa; the total volume that the hydraulic motor, servo valves and oil piping and Vt = 7.05x10-4 m3; hydraulic oil volume modulus of elasticity for βe = 7 × 108 N/m2; the external load inertia for Jt =0.31kg·m2;

$$\omega_{\mathrm{h}}=\sqrt{\frac{4\beta_{\mathrm{e}}D_{\mathrm{m}}^{2}}{J_{\mathrm{t}}V_{\mathrm{t}}}}=140rad/S \tag{9}$$

The electrohydraulic servo valve model FF106/100. has a rated current of 40 mA and a rated pressure of 21 MPa. The equivalent damping coefficient of the electrohydraulic servo valve is ξsv = 0.6, and the system damping ratio ξh = 0.17. The servo amplifier gain is Ka = 0.0014 A/V.

Then the hydraulic motor angular velocity to spool displacement and external load transfer function are respectively:

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$$\frac{\omega(s)}{X_v(s)}=\frac{\frac{K_q}{D_m}}{\frac{s^2}{\omega_h^2}+\frac{2\xi_h}{\omega_h}s+1}=\frac{931}{0.51\times10^{(-4)}s^2+2.42\times10^{(-3)}s+1}$$
$$\frac{\omega(s)}{T_L(s)}=\frac{\frac{K_{ce}}{D_m^2}\left(1+\frac{V_t}{4\beta_e K_{ce}}s\right)}{\frac{s^2}{\omega_h^2}+\frac{2\xi_h}{\omega_h}s+1}=\frac{0.0079(1+0.021s)}{0.51\times10^{(-4)}s^2+2.42\times10^{(-3)}s+1} \tag{10}$$

The transfer function of the servo valve is:

$$G_V(s)=\frac{Q(s)}{I(s)}=\frac{K_q}{\frac{s^2}{\omega_V^2}+\frac{2\xi_V}{w_V}s+1}=\frac{0.0364}{1.01\times10^{-5}s^2+3.8\times10^{-3}s+1} \tag{11}$$

4. SIMULATION ANALYSIS OF HYDRAULIC SYSTEM OF ROD CHANGER

4.1. Joint simulation system modelling

The invariance of the sliding-mode dynamics in sliding-mode variable-structure control makes the controlled object ideally robust to system uncertainties caused by nonlinear factors. While conventional control systems usually use continuous control, the "structure" of a sliding mode variable structure control system is a discontinuous control that exhibits switching characteristics, which are time-dependent. The general term "sliding mode" motion refers to the high-frequency, small-amplitude up-and-down motion according to a predetermined state trajectory under the effect of the system's switching characteristics described above. The application of sliding mode variable structure control to the control of valve-controlled motors will greatly compensate for the influence of nonlinear factors on the performance of the hydraulic system, improve the robustness of the system, and achieve a better steady-state accuracy.AMESim has a rich library of models, which has a great advantage in the modeling and simulation of hydraulic systems. It is more accurate to model the control system in MATLAB/Simulink, and the application of joint simulation technology of AMESim and Simulink can not only give full play to the advantages of AMESim in building the hydraulic system, but also give full play to the powerful digital processing capability of MATLAB/Simulink to complete the simulation.

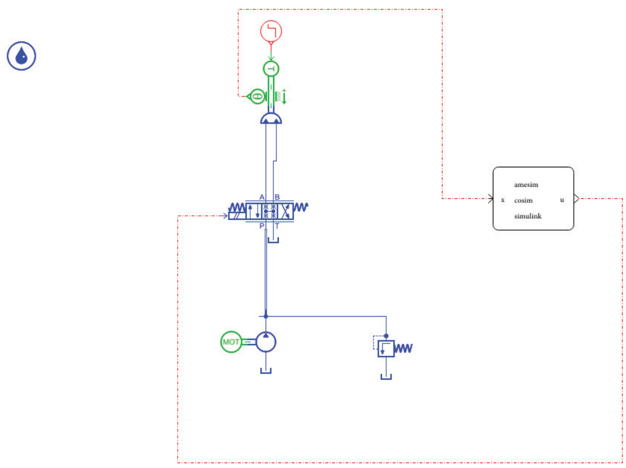

Figure 2. AMEsim hydraulic simulation model of valve-controlled motor

Firstly, the valve-controlled motor system shown in Fig. 2 is modelled in AMEsim. Then a simulink interface is created in the co-simulation interface interface. In order to let the AMEsim

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model run in Simulink, the input and output of the interface module are set up as interfaces for the mutual information exchange between AMEsim and Simulink, respectively. The input of the input is the angular velocity signal of the hydraulic motor which is fed back from the AMEsim model and transferred to the Simulink model, and the output signal of the output is calculated and then transferred to the AMEsim servo valve. The output signal is calculated and transferred to the AMEsim servo valve.

4.2. Sliding mode controller design

The motion trajectory of the sliding mode control is mainly divided into two aspects: (1) the stage of the system's arbitrary initial state moving towards the sliding mode surface; (2) the stage of the system arriving at the sliding mode surface and slowly tends to stabilize. Therefore, the design of the sliding mold variable structure controller, corresponding to the two stages of the system motion, can be divided into two parts: the first part, the design of the sliding mold surface; the second part, the design of the control law.

The original transfer function is deformed into:

$$\begin{aligned} \dot{x} &= \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & -\omega_h^2 & -2\xi_h\omega_h \end{bmatrix} x + \begin{bmatrix} 0 \\ 0 \\ K_h\omega_h^2 \end{bmatrix} u \\ y &= \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} x \end{aligned} \tag{12}$$

The designed sliding mode switching function is:

$$s = c_1e_1 + c_2e_2 + e_3 \tag{13}$$

For satisfying, $s = c_1e_1 + c_2e_2 + e_3 = 0$;Simplifying the differential equation of sliding mode motion:

$$s = c_1e_1 + c_2e_2 + e_3 = c_1e_1 + c_2\dot{e}_1 + \ddot{e}_1 = 0 \tag{14}$$

For the sliding mode variable structure controller is designed as follows:

$$u = u_{eq} + u_{sw} \tag{15}$$

u_{eq} It is the equivalent control that enables the tracking of the system state, i.e., the state of the system is always kept on the sliding mold surface; u_{sw} is the switching control that converges the state of the system to the sliding mold surface to weaken the jitter of the system, using the exponential convergence law:

$$\dot{s} = -\varepsilon\,\mathrm{sgn(s)} - \mathrm{ks} \tag{16}$$

included among these

$$\mathrm{sgn(s)} = \begin{cases} 1 & s > 0 \\ -1 & s < 0 \end{cases}$$

So the switching control is:

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$$u_{sw} = -\varepsilon\,\mathrm{sgn}(s) - ks \tag{17}$$

Figure 3 shows the sliding mode variable structure control model of the valve-controlled motor servo system of the rod changer.

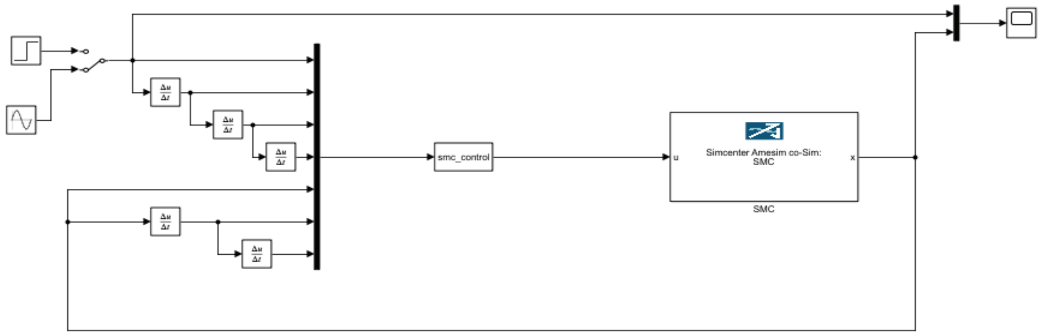

Figure 3. Sliding mode control model of valve-controlled hydraulic motor system

4.3. Simulation results analysis

By designing the sliding mode controller and using the above joint simulation model, the simulation results can be obtained by inputting a step signal to the system as shown in Fig. 4. From the figure, it can be seen that when a step signal is input to the system at 1 second, the response time of the valve-controlled motor system is 1s, and the overshoot is 1.1% at the same time. The image shows that the system response is fast and there are no sudden changes or large oscillations. The results show that the stiffness of the sliding mode control strategy is very good, and the accuracy in steady state is very high.

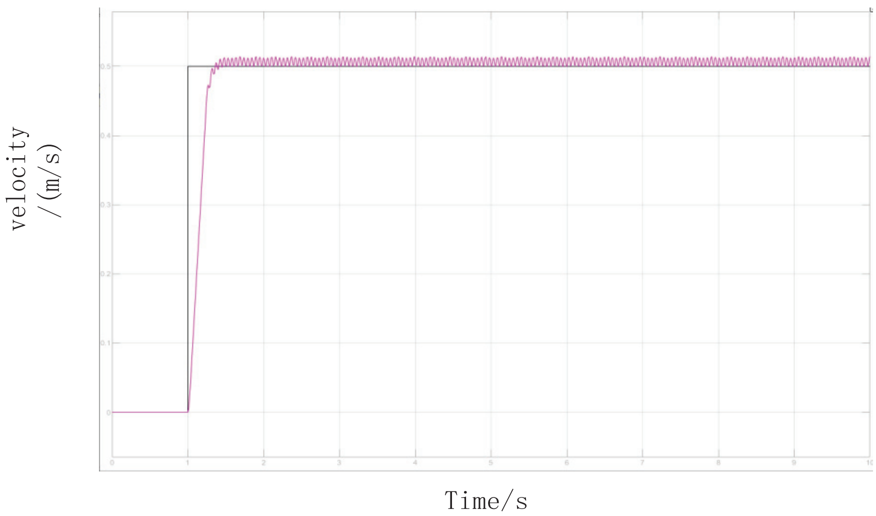

Figure 4. Step response simulation curve

The sinusoidal response performance of the joint simulation model is simulated using the sliding mode variable structure fear value, the simulation time is set to 10S, and the sinusoidal signal is applied at the moment of 0S, and[7] the results are shown in Fig. 4, and the image results show that the sliding mode control strategy can satisfy the dynamic and static performances as well as ensure the system stiffness and robustness.

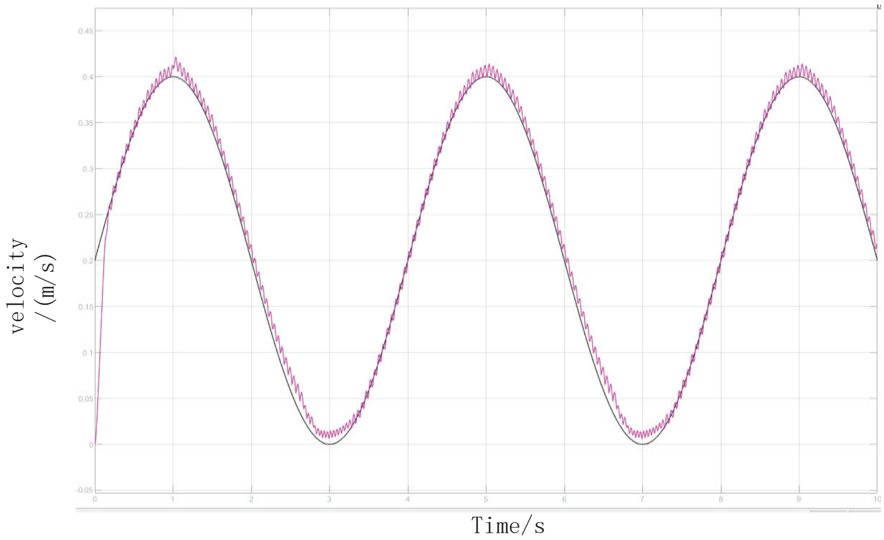

Figure 5. Sinusoidal tracking simulation curve

ACKNOWLEDGMENTS

In this paper, the structure of downhole drilling rig rod changing device is firstly summarized, and its specific structure and control process are introduced. Afterwards, a transfer function mathematical model was established [8] for the valve-controlled motor servo system diagram. And then designed the sliding mode controller, for the valve-controlled motor system through the establishment of AMEsim/Simulink joint simulation model for the sliding mode controller simulation verification. The simulation results show that the sliding mode controller designed in this paper meets the design requirements of the rod changer.

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