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Finite-Time Command Filtered Backstepping Algorithm-Based Pitch Angle Tracking Control for Wind Turbine Hydraulic Pitch Systems

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ABSTRACT In this paper, a novel finite-time command filtered backstepping control algorithm is proposed to address the problems of high order nonlinearity, noise and friction interference when tracking the pitch angles of wind turbine hydraulic pitch systems. Since taking derivative is not required, the system noise cannot be amplified in the algorithm design process. The finite-time command filter is first employed to filter the state variables of the hydraulic pitch systems to eliminate the interference caused by the noise and friction. Moreover, the filter is combined with backstepping design to approximate the derivatives of virtual control variables to avoid ''Differential expansion'' phenomenon. In addition, in order to ensure the accuracy of the filtered signals to approximate the virtual control variables, a finite-time error compensation mechanism is designed. Simulation results show the effectiveness and high-precision tracking performance of the proposed algorithm in this paper.

INDEX TERMS Wind turbine, hydraulic pitch system, backstepping control, finite-time command filter, position tracking.

I. INTRODUCTION

As large-scale wind turbines continues to increase, hydraulic pitch systems are widely used in the turbines. Compared with the electric servo pitch systems, the hydraulic pitch systems have many advantages such as faster response, longer service life, higher reliability[1]–[3]. The hydraulic pitch systems often use hydraulic cylinders as actuators, This system is the first choice for large wind turbines as it has the characteristics with high power density, small clearance and high reliability. The pitch systems driven by hydraulic cylinders convert the linear motion of the hydraulic cylinders into the circular motion of blades by a crank slider mechanism, thereby realizing pitch operation. However, the crank slider mechanism will complicate the pitch mechanism and prone to occur faults. In order to overcome the shortages of hydraulic cylinder pitch systems, we apply hydraulic motors as actuators in this paper. The hydraulic motor pitch system drives pinion gears to drive large ring gears at the blade root by the hydraulic motors, which has simpler structure and higher reliability than applying the hydraulic cylinders.

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To achieve effective tracking control of hydraulic systems, many scholars have completed a lot of researches using various advanced control strategies. Reference [4]–[6] proposed using genetic algorithm and fuzzy rule to optimize the parameters of PID controller to achieve adaptive control and improve the tracking performance of hydraulic systems. Reference [7]–[9] addressed the nonlinear dynamics problem of hydraulic systems by means of feedback linearization. Furthermore, Lyapunov theory was used to derive control laws, which makes the system stable. Reference [10]–[12] adopted fuzzy self-adjustment mechanism to adapt to sliding mode control parameters. Therefore, the system vibrations are reduced and convergence speed is improved. In [13], a hydraulic pitch controller with self-tuning fuzzy sliding mode compensation was developed to better compensate the influence of Coulomb friction in hydraulic pitch systems. Most of the above references adopted linearization methods. However, hydraulic pitch systems have complex dynamic characteristics, for instance, the system parameters will change with changing of pressure, oil temperature, and valve opening, which demand higher performance of the controller.

In order to deal with the complex dynamic characteristics of hydraulic systems, [14] proposed an ideal adaptive compensation method with the function of eliminating system noise, in which actual state feedback values were replaced by the expected values and the discontinuity caused by the symbol function was approximated by the continuous function in compensation loop. Thereby, the progressive tracking performance of hydraulic systems was improved. In [15]–[19], a self-adaptive integral control for hydraulic systems was proposed. The parameter uncertainty of systems was addressed by the adaptive control law of discontinuous projection, and the unmodeled interference was reduced by the integral robust feedback term based on the extended error. In recent years, many scholars have begun to apply backstepping design method to hydraulic systems. The backstepping control was proposed to solve the nonlinearity and parameter uncertainty of the system, and realize adaptive control for hydraulic systems in [20]–[24]. In addition, to realize progressive tracking in pitch systems, backstepping control was combined with adaptive rules to solve nonlinearity and parameter time-varying in pitch systems in [25] and [26]. However, it is difficult to guarantee the tracking accuracy for external interference and noise.

In order to realize the bounded position tracking performance of wind turbine hydraulic pitch systems and improve the tracking accuracy of pitch angle, we proposes a finite-time command filtered backstepping control strategy in this paper. It is well known that the backstepping control can achieve adaptive control of uncertain nonlinear systems. However, in the backstepping design process, the derivatives of virtual control variables need to be gained, which prone to the problem of ''differential explosion''. Therefore, we use a novel command filter to approximate the derivatives of the virtual control variables in the backstepping design process, avoiding direct derivation of the virtual control variables. Furthermore, we not only employ an error compensation mechanism for derivation error to reduce the deviation introduced by command filter but also combine the command filter with finite-time method to filter the actual state feedback variables to reduce the interference of noise and friction. So the system has better anti-interference performance, faster convergence speed and higher precision.

The algorithm designed in this paper has many advantages as follows.

(1) The problem of ''differential explosion'' caused by direct derivative calculation is overcome by the command filter technique. In addition, an error compensation mechanism is used to compensate the error.

(2) The derivatives of the state variables of hydraulic pitch systems are not taken, so that the noise is not be amplified, and the interference caused by noise and friction is reduced by the command filter.

(3) Compared with the PID control based on artificial bee colony algorithm and finite-time expansion differential backstepping control, the proposed finite-time command filtered backstepping control algorithm in this paper has faster convergence speed and stronger anti-interference capacity.

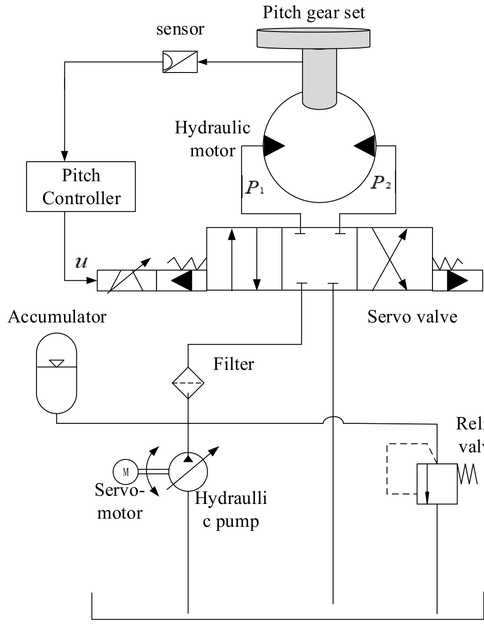


FIGURE 1. Hydraulic motor pitch system.

II. MODELLING FOR HYDRAULIC PITCH SYSTEMS

The hydraulic pitch system is mainly composed of a hydraulic motor, an electro-hydraulic proportional valve, an accumulator, an overflow valve, a hydraulic pump and a pitch controller, as shown in FIGURE 1. The electro-hydraulic proportional valve changes the flow direction of the oil according to the polarity of the output electric signals of the pitch controller to realize the positive and negative rotation of the hydraulic motor. In addition, the oil flow is controlled to adjust the rotation speed of the hydraulic motor in terms of the amplitude of electric signals. The hydraulic motor drives pinion gears to propel girth gear rings at blade root, which changes the pitch angle. The accumulator eliminates pressure pulsation by absorbing oil shock to improve the stability of the entire hydraulic systems. When system pressures are greater than the normal set value of the systems, the overflow valve acting as a safety valve will open and the pressure oil will return to the fuel tank to avoid excessive pressure in the hydraulic systems, which ensures not damage the hydraulic systems.

The relationship between the spool displacement and control command is usually approximated as a first-order linear formula, which is expressed as follows.

$$x_v = k_v u \tag{1}$$

where x_v is the electro-hydraulic proportional valve spool displacement, k_v is the proportional gain, u is the control signal.

According to Newton's second law, the force balance formula of hydraulic motor is expressed as follows.

$$J\ddot{\theta} = D_m P_s - B_m \dot{\theta} - T_l - T_f \tag{2}$$

where θ is the rotation angle of the motor, J is the total moment of the inertia of the hydraulic motor and loads, D_m is the displacement of the hydraulic motor, $P_s = P_1 + P_2$ is the supply pressure of the hydraulic system, P_1 is the inlet chamber pressure of the hydraulic motor, P_2 is the return chamber pressure of the hydraulic motor, B_m is the damping coefficient of the hydraulic viscous damping, T_l is the load torque, T_f is the friction torque.

The dynamic formula of the load pressure of hydraulic motor is expressed as follows.

$$\frac{V_t}{4\beta_e}\dot{P}_l = -D_m\dot{\theta} - C_t P_l + Q_l \tag{3}$$

where V_t is the total compression volume of the two chambers and the connecting pipe of the hydraulic motor, β_e is the effective bulk elastic modulus, $P_l = P_1 - P_2$ is the load pressure, C_t is the total leakage coefficient of the hydraulic motor, Q_l is the load flow of the hydraulic motor.

The flow formula of the electro-hydraulic proportional valve is expressed as follows.

$$Q_l = C_d W x_v \sqrt{\frac{P_s - sign(x_v) P_l}{\rho}} \tag{4}$$

where C_d is the flow coefficient of the valve, W is the area gradient of the valve, ρ is the density of the hydraulic oil.

Symbolic function Sign($*$) is expressed as follows.

$$sign(*) = \begin{cases} 1, & if * > 0 \\ 0, & if * = 0 \\ -1, & if * < 0 \end{cases} \tag{5}$$

The pitch angle β is expressed as follows.

$$\beta = \frac{\theta}{i_g} \tag{6}$$

where i_g is the gear ratio, and its size must meet the following constraint.

$$\frac{w_{m\min}}{w_{p\min}} \le i_g \le \frac{w_{m\max}}{w_{p\max}} \tag{7}$$

where $w_{m\min}$ and $w_{m\max}$ are the minimum and maximum speeds of the hydraulic motor respectively, $w_{p\min}$ and $w_{p\max}$ are the minimum and maximum speeds of the pitch gear respectively.

The state variables in hydraulic pitch systems are defined as $x = [x_1, x_2, x_3]^T = \left[\beta, \dot{\beta}, P_l\right]^T$, then the state space formulas of the hydraulic pitch systems are expressed as follows.

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = \dfrac{D_m}{J_{i_g}} x_3 - \dfrac{B_m}{J_{i_g}} x_2 - \dfrac{T_l + T_f}{J_{i_g}} \\ \dot{x}_3 = \dfrac{4\beta_e C_d Wk}{V_t\sqrt{\rho}} u\sqrt{P_s - x_3\, sign(u)} - \dfrac{4\beta_e C_t}{V_t} x_3 \\ \quad -\dfrac{4\beta_e D_m i_g}{V_t} x_2 \\ x_1 = y_d \end{cases} \tag{8}$$

The formula (8) can be further expressed as follows.

$$\begin{cases} \dot{x}1 = x_2 \\ \dot{x}2 = b_1 x_3 - b_2 x_2 - b_3 \\ \dot{x}3 = b_4 g(x,u) u - b_5 x_3 - b_6 x_2 \\ x_1 = y_d \end{cases} \tag{9}$$

where $b_1 = \frac{D_m}{J_{i_g}}, b_2 = \frac{B_m}{J_{i_g}}, b_3 = \frac{T_l + T_f}{J_{i_g}}, b_4 = \frac{4\beta_e C_d Wk}{V_t\sqrt{\rho}}$, $b_5 = \frac{4\beta_e C_t}{V_t}, b_6 = \frac{4\beta_e D_m i_g}{V_t}, g(x,u) = \sqrt{P_s - x_3\, sign(u)}$, y_d is the input signal.

III. DESIGN FOR CONTROL LAWS

Hypothesis 1: the input signal y_d of the reference angle is continuous, n-order derivable and bounded.

Hypothesis 2: the supply pressure P_s of the hydraulic motor pitch system is a constant, and the friction torque T_f always hinders the system from moving.

Lemma 1 [27]: Suppose $V(x)$ is a smooth positive definite function on C^1 (defined $U \subset R^n$) and $\dot{V}(x) + \lambda V^\alpha(x)$ is a negative semidefinite function on $U \subset R^n, \alpha \in (0,1)$. There is a region of $U_0 \subset R^n$, in which any $V(x)$ can reach zero within a finite time. If the time required to reach $V(x) \equiv 0$ is T_r, then $T_r \le \frac{V^{1-\alpha}(x_0)}{\lambda(1-\alpha)}$.

where $V(x_0)$ is the initial value of the $V(x)$.

Lemma 2 [28]: For any real numbers $\lambda_1 > 0$, $\lambda_2 > 0, 0 < \gamma < 1$, the condition of the finite time stable Lyapunov is $\dot{V}(x) + \lambda_1 V(x) + \lambda_2 V(x)^\gamma \le 0$, and the stable time is estimated by $T_r \le t_0 + \left[1/\lambda_1(1-\gamma)\right] \ln\left[\left(\lambda_1 V^{1-\gamma}(t_0) + \lambda_2\right)/\lambda_2\right]$.

The first-order Levant differentiator is further expressed as follows [29], [30].

$$\begin{cases} \dot{\varphi}_1 = v_1 \\ v_1 = -r_1|\varphi_1 - \alpha_r|^{\frac{1}{2}} sign(\varphi_1 - \alpha_r) + \varphi_2 \\ \dot{\varphi}_2 = -r_2\, sign(\varphi_2 - v_1) \end{cases} \tag{10}$$

where α_r is the input signal. The following lemma is obtained when the parameters r_1 and r_2 are both appropriate.

Lemma 3 [29]: If the parameters r_1 and r_2 are chosen reasonably, the following formula is valid after a finite time transient process without noise interference. And the corresponding solution of the dynamic system is stable within a finite time.

$$\begin{cases} \varphi_1 = \alpha_{r0} \\ v_1 = \dot{\alpha}_{r0} \end{cases} \tag{11}$$

Lemma 4 [31]: If the inequality $|\alpha_r - \alpha_{r0}| \le k$ is valid in the case of containing input noise, then the following inequalities exist within a finite time.

$$\begin{cases} |\varphi_1 - \alpha_{r0}| \le \mu_1 k = \varpi_1 \\ \left|v_1 - \dot{\alpha}_{r0}\right| \le \lambda_1 k^{\frac{1}{2}} = \varpi_2 \end{cases} \tag{12}$$

where ϖ_1 and ϖ_2 are both positive normal numbers and their sizes depend on the design parameters of the first-order Levant differentiator.

Based on the formula (10), the design of the finite time command filter is expressed as follows.

$$\begin{cases} \dot{\varphi}_{i,1} = v_{i,1} \\ v_{i,1} = -r_{i,1} \left|\varphi_{i,1} - \alpha_i\right|^{\frac{1}{2}} sign\left(\varphi_{i,1} - \alpha_i\right) + \varphi_{i,2} \\ \dot{\varphi}_{i,2} = -r_{i,2}\, sign\left(\varphi_{i,1} - \alpha_i\right) i = 1, \cdots, n-1 \end{cases} \tag{13}$$

where α_i is the input of the virtual control signal, $x_{i+1,c} = \varphi_{i,1}$ and $\dot{x}_{i+1,c} = v_{i,1}$ are both the outputs of the finite time command filter.

The tracking error of the hydraulic pitch systems are as $z_1 = x_1 - y_d$, $z_2 = x_2 - x_{2,c}$, $z_3 = x_3 - x_{3,c}$, which are further defined based on the outputs of the finite time command filter. y_d is the desired angle input signal, $x_{2,c}$ and $x_{3,c}$ are both the virtual control signals from the finite time command filter.

The error compensation signal can be expressed as $v_i = z_i - \delta_i$, $i = 1, 2, 3$, δ_i is the error compensation value.

Define Lyapunov function as follows.

$$V_1 = \frac{1}{2} v_1^2 \tag{14}$$

The derivative of function V_1 is as follows.

$$\begin{aligned} \dot{V}_1 &= v_1 \dot{v}_1 \\ &= v_1 \left(\dot{z}_1 - \dot{\delta}_1\right) \\ &= v_1 \left(z_2 + \left(x_{2,c} - \alpha_1\right) + \alpha_1 - \dot{y}_d - \dot{\delta}_1\right) \end{aligned} \tag{15}$$

Construct the virtual control variable α_1 as follows.

$$\alpha_1 = -k_1 z_1 + \dot{y}_d - s_1 v_1^{\gamma} \tag{16}$$

where k_1 and s_1 are both the positive parameters to be designed, γ is a normal number whose range is $0 < \gamma < 1$.

Define the error compensation signal as δ_1, then

$$\dot{\delta}_1 = -k_1 \delta_1 + \left(x_{2,c} - \alpha_1\right) + \delta_2 - h_1\, sign\left(\delta_1\right) \tag{17}$$

where h_1 is the parameter to be designed, $\delta_1(0) = 0$.

Substitute the formulas (16) and (17) into formula (15), then

$$\begin{aligned} \dot{V}_1 &= v_1 \left(z_2 + x_{2,c} - \dot{y}_d - \dot{\delta}_1\right) \\ &= v_1 \begin{pmatrix} z_2 + x_{2,c} - \dot{y}_d + k_1 \delta_1 - \left(x_{2,c} + k_1 z_1 - \dot{y}_d + s_1 v_1^{\gamma}\right) \\ -\delta_2 + h_1\, sign\left(\delta_1\right) \end{pmatrix} \\ &= v_1 \left(-k_1 v_1 + v_2 - s_1 v_1^{\gamma} + h_1\, sign\left(\delta_1\right)\right) \end{aligned} \tag{18}$$

Define Lyapunov function V_2 as follows.

$$V_2 = V_1 + \frac{1}{2} v_2^2 \tag{19}$$

The derivative of function V_2 is as follows.

$$\begin{aligned} \dot{V}_2 &= \dot{V}_1 + v_2 \left(\dot{x}_2 - \dot{x}_{2,c} - \dot{\delta}_2\right) \\ &= \dot{V}_1 + v_2 \left(-b_2 x_2 - b_3 + b_1 \left(z_3 + x_{3,c}\right) - \dot{x}_{2,c} - \dot{\delta}_2\right) \\ &= \dot{V}_1 + v_2 \left(-b_2 x_2 - b_3 + b_1 z_3\right) \\ &\quad + v_2 \left(b_1 \left(x_{3,c} - \alpha_2\right) + b_1 \alpha_2 - \dot{x}_{2,c} - \dot{\delta}_2\right) \end{aligned} \tag{20}$$

where $b_3 = -\frac{T_l + T_f}{J_{ig}}$ in which load torque T_l and friction torque T_f are both variables. Therefore, the range of b_3 is limited within $b_{3\min} \le b_3 \le b_{3\max}$.

$$\begin{aligned} \dot{V}_2 &\le \dot{V}_1 + v_2 \left(-b_2 x_2 - b_{3\min} + b_1 z_3\right) \\ &\quad + v_2 \left(b_1 \left(x_{3,c} - \alpha_2\right) + b_1 \alpha_2 - \dot{x}_{2,c} - \dot{\delta}_2\right) \end{aligned} \tag{21}$$

The virtual control signal α_2 is as follows.

$$\alpha_2 = \frac{1}{b_1}\left(-k_2 z_2 + \dot{x}_{2,c} + b_2 x_2 + b_{3\min} - z_1 - s_2 v_2^{\gamma}\right) \tag{22}$$

where k_2 and s_2 are both the parameters to be designed, $\dot{x}_{2,c}$ is the differential of α_1 from the finite time command filter.

Define the error compensation signal δ_2 as follows.

$$\dot{\delta}_2 = -k_2 \delta_2 + b_1\left(x_{3,c} - \alpha_2\right) + \delta_1 + b_1 \delta_3 - h_2\, sign(\delta_2) \tag{23}$$

where h_2 is the parameter to be designed, $\delta_2(0) = 0$.

Substitute formulas (22) and (23) into formula (21) as follows.

$$\begin{aligned} \dot{V}_2 &= -k_1 v_1^2 - k_2 v_2^2 - s_1 v_1^{\gamma+1} - s_2 v_2^{\gamma+2} \\ &\quad + v_1 h_1\, sign\left(\delta_1\right) + v_2 h_2\, sign\left(\delta_2\right) + b_1 v_2 v_3 \\ &= \sum_{i=1}^{2}\left[-k_i v_i^2 - s_i v_i^{\gamma+1} + v_i h_i\, sign\left(\delta_i\right)\right] + b_1 v_2 v_3 \end{aligned} \tag{24}$$

Define Lyapunov function V_3 as follows.

$$V_3 = V_2 + \frac{1}{2} v_3^2 \tag{25}$$

The derivative of function V_3 is as follows.

$$\begin{aligned} \dot{V}_3 &= \dot{V}_2 + v_3 \dot{v}_3 \\ &= \sum_{i=1}^{2}\left[-k_i v_i^2 - s_i v_i^{\gamma+1} + v_i h_i\, sign\left(\delta_i\right)\right] \\ &\quad + b_1 v_2 v_3 + v_3 \left(\dot{x}_3 - \dot{\delta}_3 - \dot{x}_{3,c}\right) \\ &= \sum_{i=1}^{2}\left[-k_i v_i^2 - s_i v_i^{\gamma+1} + v_i h_i\, sign\left(\delta_i\right)\right] \\ &\quad + b_1 v_2 v_3 + v_3\left(-b_5 x_3 - b_6 x_2 + b_4 g(x,u) u - \dot{\delta}_3 - \dot{x}_{3,c}\right) \end{aligned} \tag{26}$$

where $g(x,u) = \sqrt{P_s - x_3\, sign(u)}$ is an uncertain variable, further analysis is as follows.

$$P_s - P_l\, sign(u) = (P_1 + P_2) - (P_1 - P_2)\, sign(u)$$
$$2P_2 \le P_s - P_l\, sign(u) \le 2P_1 \tag{27}$$

In practical engineering applications, P_1 and P_2 are both much greater than 0, $g(x,u)$ is greater than 0 and is bounded. Then, we can defined an inequality as follows.

$$0 < p_{\min} \le g(x,u) \le p_{\max} \tag{28}$$

According to formula (28), the following expression can be gained.

$$\begin{aligned}&\dot{V}_3\\ &\le \sum_{i=1}^{2}\left[-k_i v_i^2 - s_i v_i^{\gamma+1} + v_i h_i\, sign(\delta_i)\right]\\ &\quad + b_1 v_2 v_3 + v_3\left(-b_5 x_3 - b_6 x_2 + b_4 p_{\max} u - \dot{\delta}_3 - \dot{x}_{3,c} + d_t\right)\end{aligned} \tag{29}$$

The final output control law u is as follows.

$$u = \frac{1}{b_4 p_{\max}}\left(-k_3 z_3 + \dot{x}_{3,c} + b_5 x_3 + b_6 x_2 - b_1 z_2 - s_3 v_3^{\gamma}\right) \tag{30}$$

where k_3 and s_3 are both constants which greater than zero, $\dot{x}_{3,c}$ is the differential of α_2 from the finite-time command filter.

In the process of algorithm design, it is not necessary to calculate the derivatives of the hydraulic state variables, which not only reduces the calculation amount but also improves the operating efficiency of the algorithm, furthermore, avoids the noise of the hydraulic system being amplified for taking derivatives. Therefore, the tracking accuracy and stability of the hydraulic systems can be improved.

Define the error compensation signal δ_3 as follows.

$$\dot{\delta}_3 = -k_3\delta_3 - b_1\delta_2 - h_3\, sign(\delta_3) \tag{31}$$

where h_3 is the parameter to be designed, $\delta_3(0) = 0$.

Substitute formulas (30) and (31) into formula (29) as follows.

$$\dot{V}_3 = \sum_{i=1}^{3}\left[-k_i v_i^2 - s_i v_i^{\gamma+1} + v_i h_i\, sign(\delta_i)\right] \tag{32}$$

Based on Young's inequality, the expression can be gained as follows.

$$v_i h_i\, sign(\delta_i) \le \frac{1}{2}h_i v_i^2 + \frac{1}{2}h_i[sign(\delta_i)]^2 \le \frac{1}{2}h_i v_i^2 + \frac{1}{2}h_i \tag{33}$$

Then

$$\begin{aligned}\dot{V}_3 &\le -\sum_{i=1}^{3}\left(k_i - \frac{1}{2}h_i\right)v_i^2 - \sum_{i=1}^{3}s_i v_i^{\gamma+1} + \sum_{i=1}^{3}\frac{1}{2}h_i\\ &\le -mV_3 - nV_3^{\frac{\gamma+1}{2}} + r\end{aligned} \tag{34}$$

where $m = \min(2k_i - l_i)$, $n = \min(s_i)\cdot 2^{\frac{1+\gamma}{2}}$, $r = \sum_{1}^{n}\frac{h_i}{2}$, $i = 1, 2, 3$.

If $2k_i - h_i > 0$, we can determine that the range of v_i is $|v_i| \le \min\left\{\sqrt{\frac{2r}{(1-\theta^0)m}}, \sqrt{2\left(\frac{r}{(1-\theta^0)n}\right)^{\frac{2}{\gamma+1}}}\right\}$.

With $0 < \theta^0 < 1$, $z_i = v_i + \delta_i$, to prove that z_i converges to a small region, it is necessary to prove that δ_i is bounded within a finite time. The bounded proof of δ_i is as follows.

$$V'_3 = \frac{1}{2}\sum_{i=1}^{3}\delta_i^2 \tag{35}$$

$$\begin{aligned}\dot{V}'_3 &= \delta_1\dot{\delta}_1 + \delta_2\dot{\delta}_2 + \delta_3\dot{\delta}_3\\ &= -k_1\delta_1^2 + \delta_1\left(x_{2,c} - \alpha_1\right) + \delta_1\delta_2 - \delta_1 h_1\, sign(\delta_1)\\ &\quad - k_2\delta_2^2 + b_1\delta_2\left(x_{3,c} - \alpha_2\right)\\ &\quad - \delta_1\delta_2 + b_1\delta_2\delta_3 - \delta_2 h_2\, sign(\delta_2)\\ &\quad + \left(-k_3\delta_3^2 - b_1\delta_3\delta_2 - \delta_2 h_2\, sign(\delta_2)\right)\\ &= -\sum_{i=1}^{3}k_i\delta_i^2 - \sum_{i=1}^{3}\delta_i h_i\, sign(\delta_i) + \delta_1\left(x_{2,c} - \alpha_1\right)\\ &\quad + b_1\delta_2\left(x_{3,c} - \alpha_2\right)\\ &= -\sum_{i=1}^{3}k_i\delta_i^2 - \sum_{i=1}^{3}h_i|\delta_i| + \delta_1\left(x_{2,c} - \alpha_1\right)\\ &\quad + b_1\delta_2\left(x_{3,c} - \alpha_2\right)\end{aligned} \tag{36}$$

According to Lemma 3, $\left|\left(x_{i+1,c} - \alpha_i\right)\right| \le \varpi_{i1}$ can be obtained. With b_4 being a bounded constant, define $\eta \le b_4 g(x,u) \le \rho$, we can gain an expression as follows.

$$\begin{aligned}\dot{V}'_3 &\le -\sum_{i=1}^{3}k_i\delta_i^2 + |\delta_1|\left|\left(x_{2,c} - \alpha_1\right)\right| + |b_1||\delta_2|\left|\left(x_{3,c} - \alpha_2\right)\right|\\ &\quad - \sum_{i=1}^{3}h_i|\delta_i| + |\delta_3|\varpi_{31}\rho\\ &\le -k_0V'_3 - h_0V'^{\frac{1}{2}}_3 + \sqrt{2\times 3}\,\varpi'_1\rho V'^{\frac{1}{2}}_3\\ &\le -k_0V'_3 - \left(h_0 - \sqrt{2\times 3}\,\varpi'_1\rho\right)V'^{\frac{1}{2}}_3\end{aligned} \tag{37}$$

where $k_0 = 2\min(k_i)$, $h_0 = \sqrt{2}\min(h_i)$, $\varpi'_1 = \max\{\varpi_{i1}\}$.

According to Lemma 2, it can be proved that δ_i converges to zero within a finite time if $h_0 - \sqrt{2\times 3}\,\varpi_1\rho > 0$. Let $\delta_1 = 0$ and v_1 is bounded. Then, we can know $z_1 \le v_1 + \delta_1$. Thus, the pitch angle tracking error z_1 of the hydraulic pitch system converges to a small region.

IV. COMPARISON OF EXPERIMENTAL RESULTS

In this paper, the hydraulic components of the SimHydraulics module library in Matlab are used to build the experimental platform for hydraulic pitch systems, as shown in FIGURE 2. In the hydraulic pitch systems, the measurement noise with a

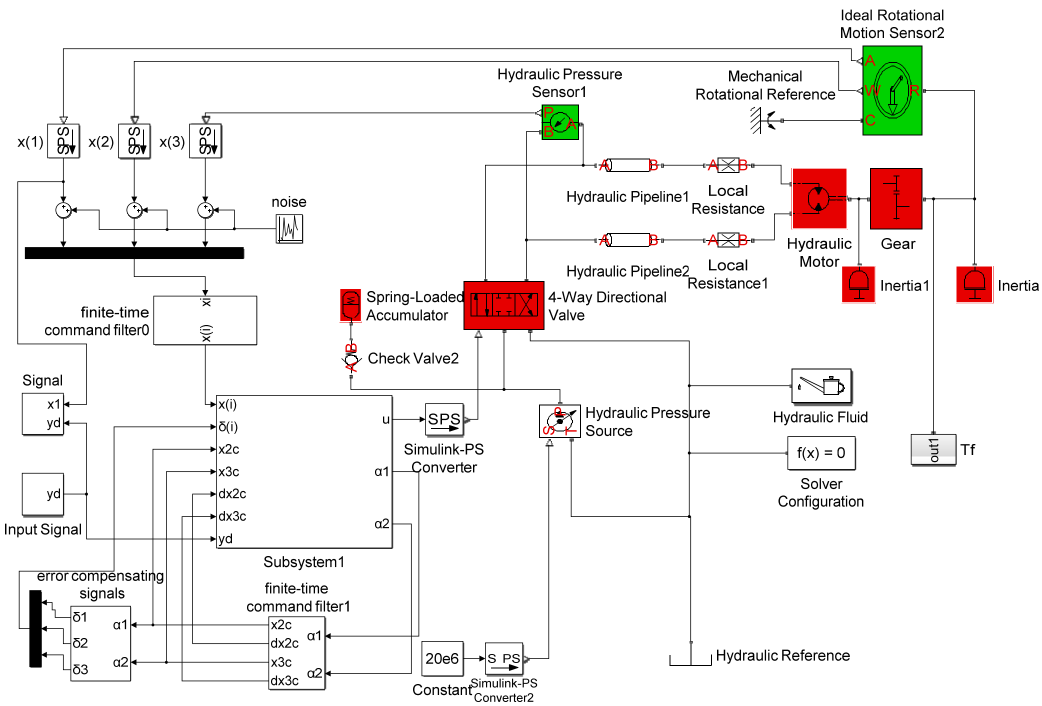


FIGURE 2. Hydraulic pitch system simulation platform.

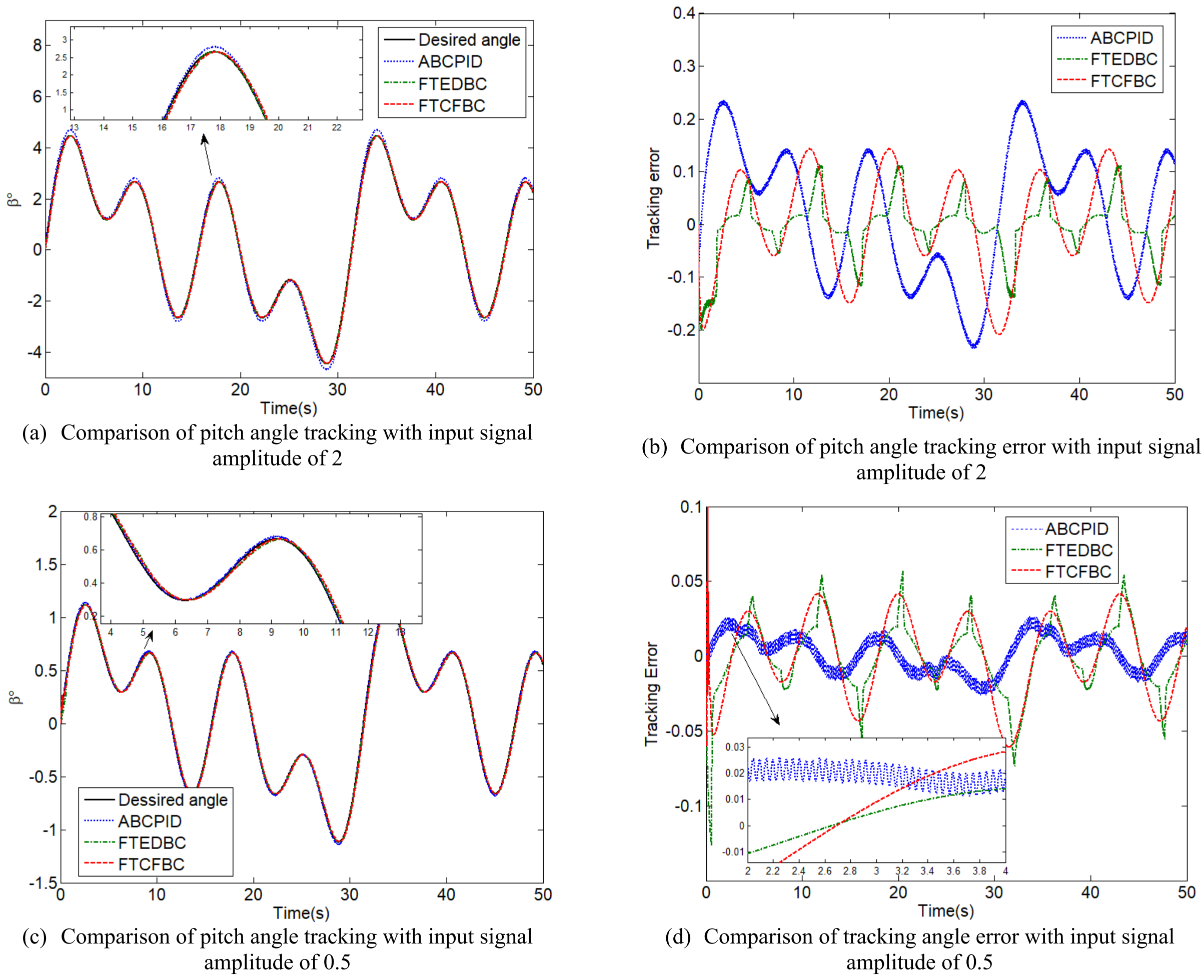
TABLE 1. Parameters designed of hydraulic motor pitch systems.

Parameter	Value
k_v	200
c_d	0.63
β_e	$6.86\times10^{8}P_a$
ρ	$947kg/m^3$
V_t	$4\times10^{-4}m^3$
D_m	$1.2\times10^{-4}m^3/rad$
J	$4\times10^{3}kg\cdot m^2$
c_t	$3.2\times10^{-12}m^5/(N\cdot s)$
B_m	$32N\cdot s\cdot m$
W	$0.01m$
P_s	$20\times10^{6}Pa$

TABLE 2. Parameter variation ranges of hydraulic motor pitch systems.

Parameter	Value$_{min}$	Value$_{max}$
b_1	$2\times10^{-9}m^2/(s^2\cdot N)$	$2.8\times10^{-9}m^2/(s^2\cdot N)$
b_2	$5.4\times10^{-4}/s$	$7.34\times10^{-4}/s$
b_3	$0.022m/s^2$	$0.034m/s^2$
b_4	$1.23\times10^{12}(N)^{1/2}/(m^3\cdot s)$	$1.64\times10^{12}(N)^{1/2}/(m^3\cdot s)$
b_5	$16.47/s$	$20.2/s$
b_6	$8.83\times10^{9}N/m^2$	$10.85\times10^{9}N/m^2$
p	$0.02\times10^{6}Pa$	$4.70\times10^{6}Pa$

power spectral density of $10^{-4}\,W/Hz$ is added to the feedback loop, with a sampling time of 1 ms. Hydraulic system piping is used to convey hydraulic oil, which parameters are shown as: the radius is 0.03 m, the length is 5 m, the polymer equivalent length of local resistance is 2 m, the geometry shape factor is 64. Hydraulic oil parameters are shown as: the model is Skydrol LD-4, the oil density is $947kg/m^3$. The electro-hydraulic proportional valve parameters are shown as: the maximum opening area of valves is $8\times10^{-5}m^2$, the maximum opening of valves is 0.03m, the critical Rayleigh number is 12, the gear ratio is 12, the simulation sampling time of hydraulic systems is 1ms.



(a) Comparison of pitch angle tracking with input signal amplitude of 2

(b) Comparison of pitch angle tracking error with input signal amplitude of 2

(c) Comparison of pitch angle tracking with input signal amplitude of 0.5

(d) Comparison of tracking angle error with input signal amplitude of 0.5

FIGURE 3. (a) Comparison of pitch angle tracking with input signal amplitude of 2; (b)Comparison of pitch angle tracking error with input signal amplitude of 2; (c)Comparison of pitch angle tracking with input signal amplitude of 0.5; (d)The comparison of tracking angle error with input signal amplitude of 0.5.

A. DESIGN OF PID CONTROLLER BASED ON ARTIFICIAL BEE COLONY ALGORITHM (ABCPID)

The PID controller is mainly composed of a proportional coefficient P, an integral coefficient I and a differential coefficient D. By adjusting three parameters of the PID controller, the output displacements of hydraulic systems can be controlled. And the desired position signals can be tracked. To reduce the adjustment time of the PID controller parameters, we implement artificial bee colony algorithm to offline optimize the parameters of the PID controller, which are $k_p = 25, k_i = 4.2, k_d = 1.5$ respectively. The control law expression is as follows.

$$u = k_p (x_1 - y_d) + k_d \frac{x_1 - y_d}{dt} + k_i \int (x_1 - y_d)\, dt \tag{38}$$

B. FINITE TIME EXPANSION DIFFERENTIATOR BACKSTEPPING CONTROL (FTEDBC)

In [24], a new backstepping control method based on extended differentiator was proposed. By applying a second-order differentiator with finite time convergence, the differential estimations of system state variables were obtained. The differential estimator was used to address the related information with uncertainty. At the same time, the derivatives of the virtual control variables in the backstepping design process are also approximated by the second-order differentiator with finite time convergence to avoid directly calculating derivatives of the virtual control variables.

The main design procedure of this algorithm is expressed as follows.

$$\begin{cases} z_1 = x_1 - y_d \\ z_2 = x_2 - \alpha_1 \\ z_3 = x_3 - \alpha_2 \end{cases} \tag{39}$$

where y_d is the input reference signal, which α_1, and α_2 are both the virtual control variables.

$$\begin{cases} \alpha_1 = \left(k_1 + \frac{1}{2}\right) z_1 + \dot{y}_d \\ \alpha_2 = -\frac{z_2}{b_1}\left(k_2 + 1 + \frac{b_1^2}{4\eta_1^2} + \frac{1}{4\eta_2^2}\left(y_{x2} - \dot{\alpha}_1\right)^2 + \frac{1}{4\eta_3^2} b_1^2 x_3^2\right) \\ u = \frac{1}{b_4 p_{\min}}\left(-\left(k_3 + \frac{1}{2} + \eta_1^2\right) z_3 + b_5 x_3 + b_6 x_2 + y_{\alpha 2}\right) \end{cases} \tag{40}$$

where y_{x2} is the approximate differential of x_2, $y_{\alpha 2}$ is the approximate differential of α_2, u is the control law of the system.

The main parameters are $k_1 = 15$, $k_2 = 15$, $k_3 = 15$, $\eta_1 = 0.3$, $\eta_2 = 0.2$, $\eta_3 = 0.2$.

C. FINITE TIME COMMAND FILTERED BACKSTEPPING CONTROL (FTCFBC)

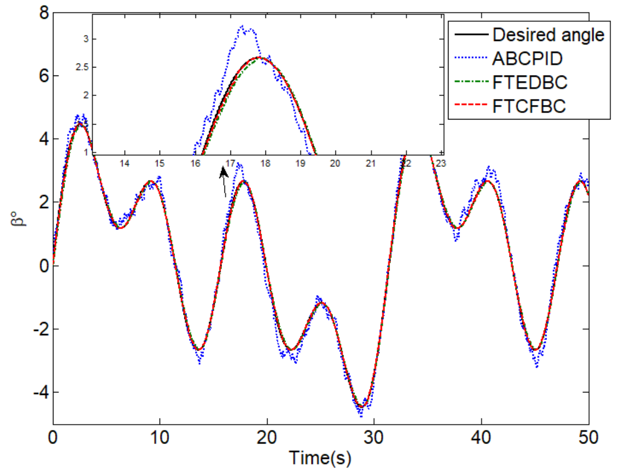
The control law in this paper is as shown in formula (30). The main design parameters are $k_1 = 15$, $k_2 = 15$, $k_3 = 40$, $\gamma = 0.6$, $h_1 = 5$, $h_2 = 5$, $h_3 = 10$, $s_1 = 25$, $s_2 = 30$, $s_3 = 30$. The design parameters of the finite time command filter are $r_1 = 400$, $r_2 = 100$.

According to the hydraulic module in simHydraulics, the experimental platform of hydraulic pitch systems is built to test the control performance of the algorithm proposed in this paper.

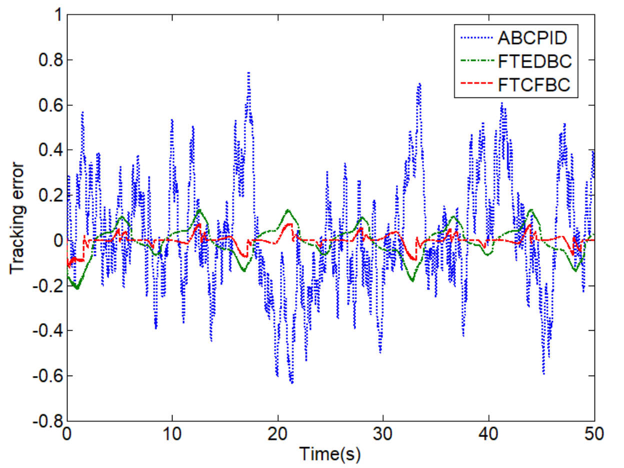
The tracking performance of the three control algorithms is tested in the environment without external interference and friction. The pitch angle input signals are shown in formula (41). FIGURE 3. (a) and (c) are both the comparison of the tracking results of the three control algorithms at different pitch angles. According to FIGURE 3. (b), the error of the PID control algorithm based on bee colony optimization is larger than that of the other two algorithms. The error curve of this proposed algorithm in this paper is smoother than that of the finite-time differentiator control algorithm. FIGURE 3. (d) is the tracking error of three control algorithms under low amplitude of 0.5, which show that PID controller based on bee colony optimization algorithm has better tracking performance in tracking low amplitude pitch angle signals except for about contrary trend pitch angle.

$$y_d = \begin{cases} 2\,(\sin(0.8t) + \sin(0.4t) + \sin(0.2t)) & 0 < t \le 50 \\ 0.5\,(\sin(0.8t) + \sin(0.4t) + \sin(0.2t)) & 0 < t \le 50 \end{cases} \tag{41}$$

Under the action of input noise and friction torque, $y_d = 2\,(\sin(t) + \sin(0.8t) + \sin(0.6t))$ is used as the input reference angle signal to test the tracking performance of the proposed algorithm and other two algorithms. The expression of the friction torque is shown in formula (42). It is well known from FIGURE 4.(b) that the pitch angle based on the PID control algorithm has large deviations under the action of the interference of noise and friction, which shows the lack of adaptive capability of the PID control algorithm. According to FIGURE 4. (b), the error of the other two algorithms is both only 20% of that of the PID control algorithm, which proves that the backstepping control based on the finite-time extended differentiator and the algorithm proposed in this paper both have good adaptive capability under the action of noise and friction interference. Compared with the finite-time extended differentiator backstepping control, the error of the proposed algorithm in this paper is about 60% of the former. On the one hand, the control accuracy is improved by the error compensation mechanism. On the other hand, the system state noise is not amplified for not calculating the derivatives of the state variables. Thus, the stability of the hydraulic system is enhanced.



(a) Comparison of pitch angle tracking under the combined action of friction and noise

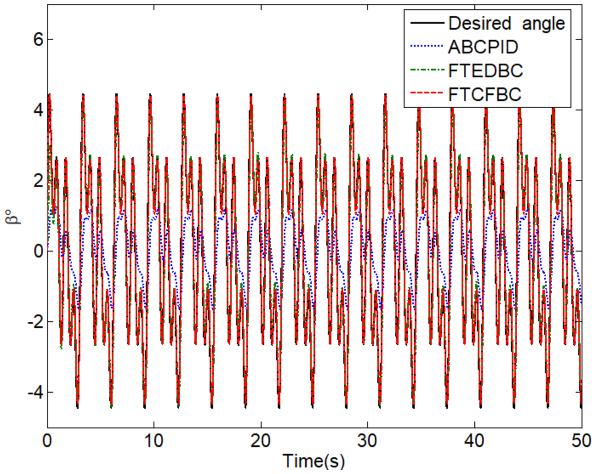


(b) Comparison of pitch angle tracking error under the combined action of friction and noise

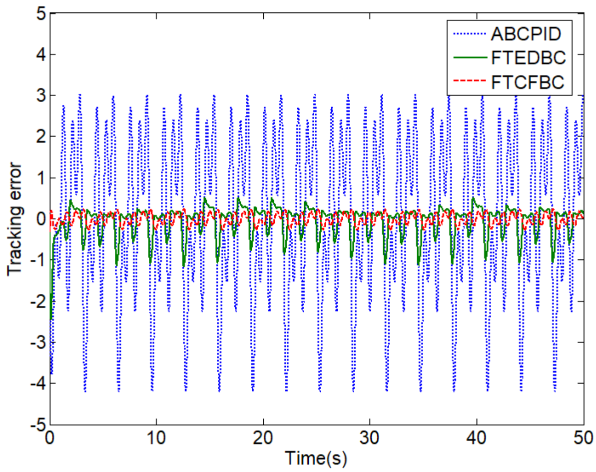
FIGURE 4. (a)Comparison of pitch angle tracking under the combined action of friction and noise; (b)Comparison of pitch angle tracking error under the combined action of friction and noise.

$$T_f = \begin{cases} 400 & w_p = 0 \\ 320sign(w_p) + 1.2\,w_p & w_p \ne 0 \end{cases} \tag{42}$$

In order to further test the response speed and tracking accuracy of the proposed algorithm, the pitch angle input signal of the faster frequency is adopted as $y_d = 2\,(\sin(8t) + \sin(4t) + \sin(2t))$, $0 < t \le 50$. We can know from FIGURE 5. (a) that the pitch angle outputs from the PID controller are completely out of the trajectory of the input pitch angle when the pitch angle changes fast, indicating that the PID control algorithm has slow response speed and insufficient self-adjusting capability. It can be concluded from FIGURE 5. (a) and (b) that the error range is $[-0.2\ 0.15]$ based on the proposed control algorithm in this paper while the error range is $[-0.8\ 0.4]$ based on finite-time differential backstepping control algorithm, which the



(a) Comparison of pitch angle tracking under the condition of fast change of input signal frequency

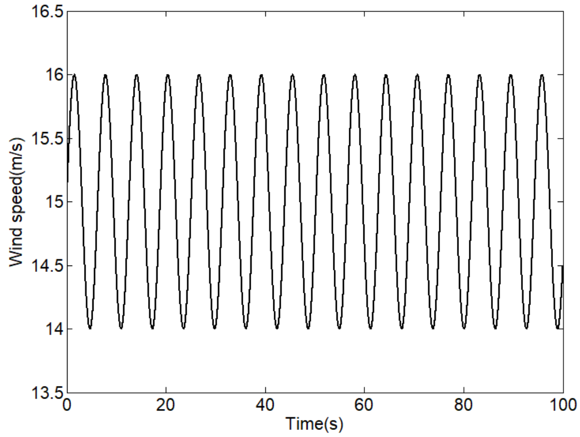


(b) Comparison of pitch angle tracking error under the condition of fast change of input signal frequency

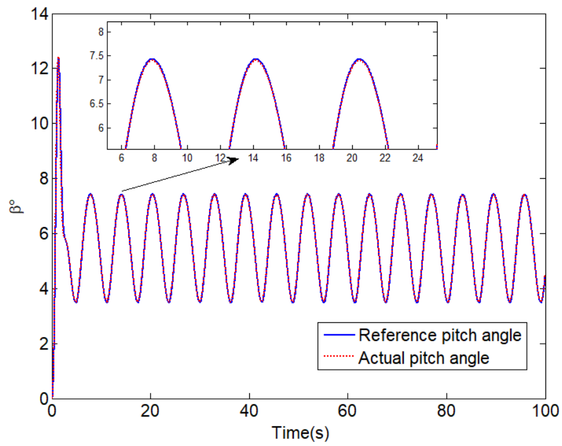
FIGURE 5. (a)Comparison of pitch angle tracking under the condition of fast change of input signal frequency; (b) Comparison of pitch angle tracking error under the condition of fast change of input signal frequency.

error of the former is about 29.2% of the latter. Accordingly, the the former has faster response speed and higher tracking accuracy than the latter. Furthermore, we can know from FIGURE 3.(b), FIGURE 4.(b) and FIGURE 5.(b) that the tracking accuracy of the algorithm based on the finite-time expansion differential backstepping is significantly reduced, however, the proposed algorithm in this paper still has high tracking accuracy.

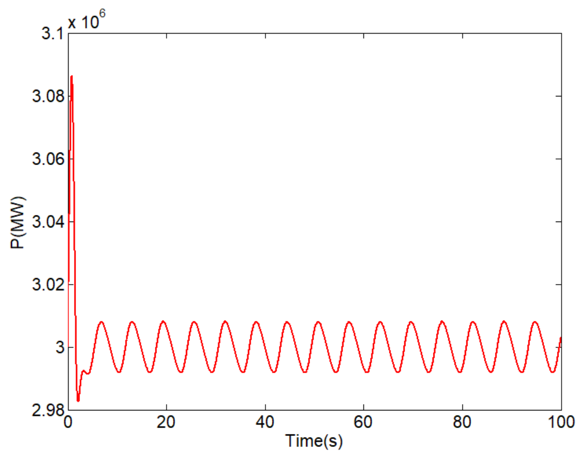
Based on the above experiments, the physical simulation models and control algorithms of the hydraulic motor pitch systems are applied in a 3MW wind turbine system for simulation test. FIGURE 6. (a) is the simulation wind speed curve with a maximum wind speed of 16m/s, a minimum wind speed of 14m/s and a rated wind speed of 12m/s. FIGURE 6.(b) and (c) show that the hydraulic motor pitch systems can track the pitch angle change on time, and the output power of the wind turbine is stable about the rated value.



(a) Sinusoidal wind speed curve



(b) Pitch angle tracking curve under sinusoidal wind speed



(c) Power output curve under sinusoidal wind speed

FIGURE 6. (a)Sinusoidal wind speed curve; (b)Pitch angle tracking curve under sinusoidal wind speed; (c)Power output curve under sinusoidal wind speed.

Thus, the effectiveness of the proposed control algorithm is further verified.

V. CONCLUSION

A novel finite-time command filtered backstepping control method is proposed for wind turbine hydraulic pitch systems to improve the position tracking performance. By using the command filtered technique and the error compensation mechanism in backstepping design, the problem of ''differential expansion'' caused by continuous derivative in backstepping control is overcome. The state trajectories of the system are bounded by means of adopting the finite-time convergence method, thus ensuring that the tracking error of the pitch angle converge to a small region. Through simulation experiments, the tracking performance of the proposed algorithm is respectively verified under the conditions of no interference, noise and friction working together, and the variation of the pitch angle frequency from FIGURE 3 to FIGURE 5. Further-more, comparing the tracking performance based on the proposed control algorithm in this paper with those of the other two control algorithms, the results show that the new algorithm has the best performance in tracking control in the more complicated case.

REFERENCES

[1] B. Boukhezzar, L. Lupu, H. Siguerdidjane, and M. Hand, ''Multivariable control strategy for variable speed, variable pitch wind turbines,'' *Renew. Energ.*, vol. 32, no. 8, pp. 1273–1287, Jul. 2007.

[2] D. Zhang, P. Cross, X. Ma, and W. Li, ''Improved control of individual blade pitch for wind turbines,'' *Sens. Actuators A, Phys.*, vol. 198, pp. 8–14, Aug. 2013.

[3] X.-X. Yin, Y.-G. Lin, W. Li, Y.-J. Gu, S. Lou, and H.-W. Liu, ''Study on variable pitch-controlled technology based on electro-hydraulic planetary bevel gear motor,'' *Zhejiang Univ Eng. Sci.*, vol. 48, no. 2, pp. 206–213, Feb. 2014.

[4] K. M. Elbayomym, J. Zongxia, and Z. Huaqing, ''PID controller optimization by GA and its performances on the electro-hydraulic servo control system,'' *Chin. J. Aeronaut.*, vol. 21, no. 4, pp. 378–384, Aug. 2008.

[5] D. Q. Truong and K. K. Ahn, ''Force control for press machines using an Online smart tuning fuzzy PID based on a robust extended Kalman filter,'' *Syst. Appl.*, vol. 38, no. 5, pp. 5879–5894, May 2011.

[6] J.-M. Zheng, S.-D. Zhao, and S.-G. Wei, ''Application of self-tuning fuzzy PID controller for a SRM direct drive volume control hydraulic press,'' *Control Eng. Pract.*, vol. 17, no. 12, pp. 1398–1404, Dec. 2009.

[7] H. A. Mintsa, R. Venugopal, J.-P. Kenné, and C. Belleau, ''Feedback linearization-based position control of an electrohydraulic servo system with supply pressure uncertainty,'' *IEEE Trans. Control Syst. Technol.*, vol. 20, no. 4, pp. 1092–1099, Jul. 2012.

[8] J. Seo, R. Venugopal, and J.-P. Kenné, ''Feedback linearization based control of a rotational hydraulic drive,'' *Control Eng. Pract.*, vol. 15, no. 12, pp. 1495–1507, Dec. 2007.

[9] B. Ayalew and K. W. Jablokow, ''Partial feedback linearising force-tracking control: Implementation and testing in electrohydraulic actuation,'' *IET Control Theory Appl.*, vol. 1, no. 3, pp. 689–698, May 2007.

[10] O. Cerman and P. Hušek, ''Adaptive fuzzy sliding mode control for electro-hydraulic servo mechanism,'' *Expert Syst. Appl.*, vol. 39, no. 11, pp. 10269–10277, Sep. 2012.

[11] H. T. Do, H. G. Park, and K. K. Ahn, ''Application of an adaptive fuzzy sliding mode controller in velocity control of a secondary controlled hydrostatic transmission system,'' *Mechatronics*, vol. 24, no. 8, pp. 1157–1165, Dec. 2014.

[12] T. H. Ho and K. K. Ahn, ''Speed control of a hydraulic pressure coupling drive using an adaptive fuzzy sliding-mode control,'' *IEEE/ASME Trans. Mechatronics*, vol. 17, no. 5, pp. 976–986, May 2012.

[13] M.-H. Chiang, ''A novel pitch control system for a wind turbine driven by a variable-speed pump-controlled hydraulic servo system,'' *Mechatronics*, vol. 21, no. 4, pp. 753–761, Jun. 2011.

[14] J. Yao, W. Deng, and W. Sun, ''Precision motion control for electro-hydraulic servo systems with noise alleviation: A desired compensation adaptive approach,'' *IEEE/ASME Trans. Mechatronics*, vol. 22, no. 4, pp. 1859–1868, Aug. 2017.

[15] G. Yang, J. Yao, G. Le, and D. Ma, ''Adaptive integral robust control of hydraulic systems with asymptotic tracking,'' *Mechatronics*, vol. 40, pp. 78–86, Oct. 2016.

[16] A. Mohanty and B. Yao, ''Indirect adaptive robust control of hydraulic manipulators with accurate parameter estimates,'' *IEEE Trans. Control Syst. Technol.*, vol. 19, no. 3, pp. 567–575, May 2011.

[17] W. Ma, W. Deng, and J. Yao, ''Continuous integral robust control of electro-hydraulic systems with modeling uncertainties,'' *IEEE Access*, vol. 6, pp. 46146–46156, 2018.

[18] J. Yao, W. Deng, and Z. Jiao, ''RISE-based adaptive control of hydraulic systems with asymptotic tracking,'' *IEEE Trans. Autom. Sci. Eng.*, vol. 14, no. 3, pp. 1524–1531, Jul. 2017.

[19] C. Guan and S. Pan, ''Nonlinear adaptive robust control of single-rod electro-hydraulic actuator with unknown nonlinear parameters,'' *IEEE Trans. Control Syst. Technol.*, vol. 16, no. 3, pp. 434–445, May 2008.

[20] D. Won, W. Kim, D. Shin, and C. C. Chung, ''High-gain disturbance observer-based backstepping control with output tracking error constraint for electro-hydraulic systems,'' *IEEE Trans. Control Syst. Technol.*, vol. 23, no. 2, pp. 787–795, Mar. 2015.

[21] Q. Guo, Y. Liu, D. Jiang, Q. Wang, W. Xiong, J, Liu, and X. Li, ''Prescribed performance constraint regulation of electrohydraulic control based on backstepping with dynamic surface,'' *Appl. Sci.*, vol. 8, no. 1, p. 76, Jan. 2018.

[22] X. Li, Z.-C. Zhu, G.-C. Rui, D. Cheng, G. Shen, and Y. Tang, ''Force loading tracking control of an electro-hydraulic actuator based on a nonlinear adaptive fuzzy backstepping control scheme,'' *Symmetry*, vol. 10, no. 5, p. 155, May 2018.

[23] Q. Guo, P. Sun, J.-M. Yin, T. Yu, and D. Jiang, ''Parametric adaptive estimation and backstepping control of electro-hydraulic actuator with decayed memory filter,'' *ISA Trans.*, vol. 62, pp. 202–214, May 2016.

[24] X. Yang, X. Zheng, and Y. Chen, ''Position tracking control law for an electro-hydraulic servo system based on backstepping and extended differentiator,'' *IEEE/ASME Trans. Mechatronics*, vol. 23, no. 1, pp. 132–140, Feb. 2018.

[25] X. Yin, Y. Lin, and W. Li, ''Integrated pitch control for wind turbine based on a novel pitch control system,'' *J. Renew. Sustain. Ener.*, vol. 6, no. 4, Jul. 2014, Art. no. 043106.

[26] X. Yin, Y. Lin, and W. Li, ''Adaptive back-stepping pitch angle control for wind turbine based on a new electro-hydraulic pitch system,'' *Int. J. Control*, vol. 88, no. 11, pp. 2316–2326, May 2015.

[27] S. Bhat and D. Bernstein, ''Finite-time stability of continuous autonomous systems,'' *SIAM J. Control Optim.*, vol. 38, no. 3, pp. 751–766, Mar. 2000.

[28] S. Yu, X. Yu, and B. M. Z. Shirinzadeh, ''Continuous finite-time control for robotic manipulators with terminal sliding mode,'' *Automatica*, vol. 41, no. 11, pp. 1957–1964, Nov. 2005.

[29] A. Levant, ''Robust exact differentiation via sliding mode technique,'' *Automatica*, vol. 34, no. 3, pp. 379–384, Mar. 1998.

[30] A. Levant, ''Higher-order sliding modes, differentiation and output-feedback control,'' *Int. J. Control*, vol. 76, no. 9–10, pp. 924–941, Nov. 2003.

[31] J. Yu, P. Shi, and L. Zhao, ''Finite-time command filtered backstepping control for a class of nonlinear systems,'' *Automatica*, vol. 92, pp. 173–180, Jun. 2018.

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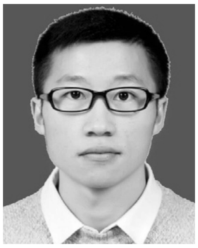

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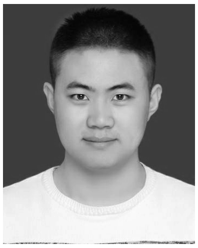

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