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Adaptive sliding mode back-stepping pitch angle control of a variable-displacement pump controlled pitch system for wind turbines

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ABSTRACT

A variable-displacement pump controlled pitch system is proposed to mitigate generator power and flap-wise load fluctuations for wind turbines. The pitch system mainly consists of a variable-displacement hydraulic pump, a fixed-displacement hydraulic motor and a gear set. The hydraulic motor can be accurately regulated by controlling the pump displacement and fluid flows to change the pitch angle through the gear set. The detailed mathematical representation and dynamic characteristics of the proposed pitch system are thoroughly analyzed. An adaptive sliding mode pump displacement controller and a back-stepping stroke piston controller are designed for the proposed pitch system such that the resulting pitch angle tracks its desired value regardless of external disturbances and uncertainties. The effectiveness and control efficiency of the proposed pitch system and controllers have been verified by using realistic dataset of a 750 kW research wind turbine.

1. Introduction

Pitch systems are normally used to keep the captured wind power around the rated value for medium to large size wind turbines. These systems are effective in protecting the turbines and the supporting structure from damage caused by strong wind gusts. These systems can also be employed to significantly improve the power conversion efficiency at low wind speeds and limit the maximum captured power in high wind gusts as compared with other aerodynamic power control systems such as active stall control system and passive stall control system [1]. Furthermore, the pitch systems react faster than the active-stall control system and provide better controllability. However, the turbulent operating characteristics and structural flexibility of wind turbines can cause failures and faults in the pitch systems and will lead to inappropriate pitch angles that will directly result in rotor over-speed and extreme turbine loads. Therefore, the pitch systems are expected to be more robust against time-varying operating conditions and more effective in turbine power and load regulations.

The pitch systems of wind turbines differ mainly in the types of the employed pitch drives. Hydraulic or electromechanical pitch drives can both be used to turn the blades in their longitudinal axis, thereby changing the blade pitch angles. The hydraulic pitch systems can rotate the adjustable turbine blades through mechanical linkages. The hydraulic power supply of the systems is commonly housed at a fixed location in the nacelle. A hydraulic system and the associated pitch controller were deigned in [2] to enhance the pitch control performances. However, the full state variables in the controller were not practically available. A hydraulic pitch system driven by two differential hydraulic cylinders was designed in [3] to achieve high energy efficiency for wind turbines. However, the pitch system had poor control accuracy due to the use of the pump controlled hydraulic cylinders. The valve controlled hydraulic motor-based pitch systems were proposed in [4–7] to reduce the turbine power fluctuations. However, the problem of relatively low operating efficiency due to the valve throttling control also existed in these systems. The hydraulic pitch systems generally suffer from the problems of relatively low operating efficiency and inherent nonlinear relationship between the cylinder displacement and the final pitch angle [2].

Generally, an electric motor can be employed in the electromechanical pitch systems to generate the pitch motions through the gears located in the rotor hub. Although relatively compact and cost-effective, the electromechanical pitch systems have the major

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disadvantages including a certain power instability with high turbulence and the relatively large power fluctuations during strong wind gusts.

In this paper, a variable-displacement pump controlled pitch system is presented for reducing generator power and turbine load fluctuations. An adaptive sliding mode pump displacement controller and a back-stepping stroke piston controller are also designed for the proposed pitch system to track the desired pitch angle in the presence of external disturbances and uncertainties. Since the valve throttling control losses can be significantly reduced by using the hydraulic pump, the pitch system has the advantages of high operating efficiency, high energy-saving capability, high torque to weight ratio and compactness. The control accuracy and asymptotic stability of the designed pitch controllers can also be guaranteed based on Lyapunov functions. Therefore, significantly improved pitch control performances can be reasonably achieved by using the proposed pitch system.

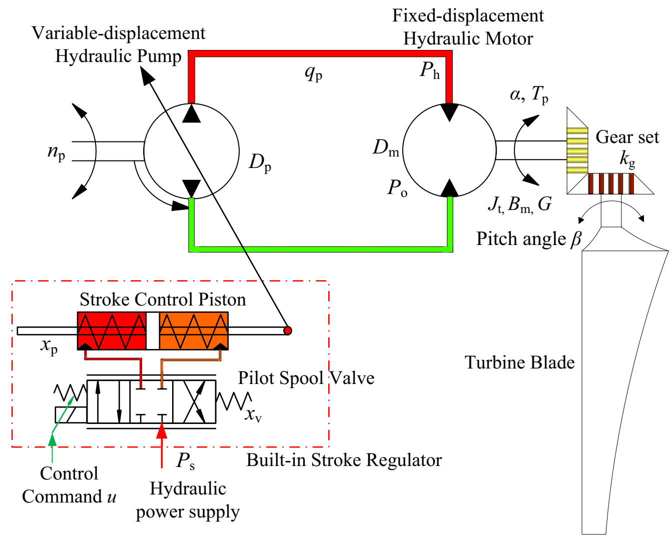

Fig. 1. Schematic of the proposed pitch system.

2. System design and modeling

2.1. System design

As illustrated in Fig. 1, the proposed pitch system mainly includes a variable-displacement hydraulic pump, a fixed-displacement hydraulic motor and a gear set. The variable-displacement hydraulic pump rotates at constant speed n_p and employs a built-in stroke regulator to regulate its displacement. The stroke regulator is essentially a pilot spool valve controlled stroke control piston with spring return. A four-way critical-center spool valve can be used as the pilot spool valve and can be reasonably controlled to change the position of the stroke control piston and hence to vary the pump displacement to regulate the pump flow. The hydraulic motor is coupled to the pump through hydraulic pipes and its speed can be continuously regulated by controlling the pump displacement and flows. The gear set with the gear ratio k_g is normally required to adapt the low speed of the turbine blade to the high speed of the hydraulic motor [8].

The use of this bidirectional-rotating hydraulic pump significantly reduces the throttling control losses and heat generation in the pitch system. Therefore, the pitch system has the advantages of high operating efficiency, high energy-saving capability, high torque to weight ratio and compactness.

2.2. Dynamic modeling

For the built-in stroke regulator, the pilot spool valve displacement x_v is approximately proportional to the control command u. Thus,

$$x_{\mathrm{v}} = k_{\mathrm{v}} \cdot u \tag{1}$$

where x_v, k_v and u denote the pilot spool valve displacement, the valve control gain and the input control command, respectively.

The flow continuity equation of the pilot spool valve controlled stroke control piston can be formulated as [9]

$$k_{\mathrm{d}} \cdot x_{\mathrm{v}} \cdot \sqrt{P_{\mathrm{s}} - P_{\mathrm{L}} \cdot \mathrm{sgn}(x_{\mathrm{v}})} = A_{\mathrm{p}} \cdot \dot{x}_{\mathrm{p}} + \frac{V_{\mathrm{p}}}{4\beta_{\mathrm{e}}} \dot{P}_{\mathrm{L}} \tag{2}$$

where k_d, P_s and P_L denote a constant valve flow coefficient, hydraulic supply pressure and load pressure of the stroke control piston, respectively, A_p, x_p, V_p and β_e denote the effective piston area, piston displacement, total volume of the pilot spool valve controlled piston combination and the effective bulk modulus, respectively.

The function $\mathrm{sgn}(x_v)$ can be defined as

$$\mathrm{sgn}(x_{\mathrm{v}}) = \begin{cases} 1, & \text{if } x_{\mathrm{v}} \geq 0 \\ -1, & \text{if } x_{\mathrm{v}} < 0 \end{cases} \tag{3}$$

The force balance equation of the stroke control piston can be described as

$$A_{\mathrm{p}} \cdot P_{\mathrm{L}} = m_{\mathrm{p}} \cdot \ddot{x}_{\mathrm{p}} + B_{\mathrm{p}} \cdot \dot{x}_{\mathrm{p}} + k_{\mathrm{s}} \cdot x_{\mathrm{p}} + F_{\mathrm{p}} \tag{4}$$

where m_p, B_p, k_s and F_p denote the effective piston mass, piston viscous damping coefficient, spring stiffness and the external disturbance, respectively.

For this variable-displacement hydraulic pump, the pump displacement D_p is proportional to the displacement of the stroke control piston x_p. Therefore,

$$D_{\mathrm{p}} = k_{\mathrm{p}} \cdot x_{\mathrm{p}} \tag{5}$$

where k_p denotes the displacement gradient of the pump stroke control loop.

Assuming that the two pressures P_h and P_o ($P_h > P_o$) are uniform, no pipe losses or pressure saturation. The pump flow can be described as

$$q_{\mathrm{p}} = D_{\mathrm{p}} \cdot n_{\mathrm{p}} \tag{6}$$

where q_p, P_h and n_p are the pump flow rate, pump pressure and speed, respectively.

The flow equation of the hydraulic motor can be formulated as [10]

$$q_{\mathrm{p}} = D_{\mathrm{m}} \cdot \frac{\mathrm{d}\alpha}{\mathrm{d}t} + \frac{V_{\mathrm{o}}}{\beta_{\mathrm{e}}} \cdot \frac{\mathrm{d}(P_{\mathrm{h}} - P_{\mathrm{o}})}{\mathrm{d}t} \tag{7}$$

where α, P_o, V_o and D_m denote the angular displacement, replenishing pressure, total volume and displacement of the hydraulic motor, respectively.

The torque balance equation of the hydraulic motor can be described as

$$D_m \cdot (P_h - P_o) = J_{\mathrm{t}} \cdot \frac{\mathrm{d}^2\alpha}{\mathrm{d}t^2} + B_{\mathrm{m}} \cdot \frac{\mathrm{d}\alpha}{\mathrm{d}t} + G \cdot \theta_{\mathrm{m}} + T_{\mathrm{p}} \tag{8}$$

where J_t, B_m, G and T_p denote the inertia, viscous damping coefficient, pitch spring gradient and external pitch loads of the hydraulic motor, respectively.