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Technical note

Design, modeling and implementation of a novel pitch angle control system for wind turbine

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A B S T R A C T

A novel pitch angle control system is proposed to smooth output power and drive-train torque fluctuations for wind turbine. This system is characterized by an outer open control loop for enhancing the direct pitching motion and an intrinsic hydro-mechanical position control loop offering the benefit of sensor-less pitch control. A pragmatic design procedure is provided and several key design parameters are determined or optimized. Modeling, stability analysis and dynamic characteristics of this pitch control system are also presented. Comparative experimental results have validated the effectiveness and efficiency of this system in power and torque regulations.

1. Introduction

Pitch angle control systems are commonly employed in medium to large wind turbines to keep the captured wind power close to the rated value above the rated wind speed, bringing about the benefits of better control flexibility and power quality. Such systems can also alleviate the structural wind loads and protect the wind turbine from fatigue damage during strong wind gusts. Thus, these systems have an immediate influence on the wind power regulation and are of significant importance for variable pitch wind turbines. Moreover, high performance and reliability oriented advanced pitch control systems can meet the ever increasing stringent performance requirements specified by modern turbines and hence are essential to enhance the competitiveness of wind energy technology [1].

There are fundamentally two types of such systems: electromechanical and hydraulic types. For the electromechanical type, the pitch actions can be achieved by using an electric motor. This system has been extensively investigated in the literature including system design [2], analysis of dynamic characteristics [3], double closed-loop control [4], direct torque pitch control [5], adaptive pitch control [6], and fuzzy logic pitch control [7]. Although relatively compact and accurate, the robustness and power-mass ratio of this system could be relatively low.

For the hydraulic pitch system, a value controlled hydraulic cylinder is commonly employed to generate the final pitch actions through a slider-crank mechanism [8]. Recent studies on this system mainly include pitch control strategies [9,10], reliability evaluation [11], system modeling [12], and independent pitch control [13]. Chiang et al. [14] developed a variable-speed pump-controlled hydraulic pitch control system and an adaptive fuzzy pitch controller. However, despite of such various control methods in the literature, sufficiently detailed dynamic analysis of this system was not provided. Although the hydraulic pitch control system may be advantageous in high power/mass ratio and relatively high reliability, the control accuracy of this system is relatively poor due to the use of the slider-crank mechanism [8].

The major contributions of this paper are a novel pitch angle control system and the detailed analysis methods such as design procedure, system modeling, stability and dynamic analysis. By integrating the fundamentally different working mechanisms of the aforementioned two types, this novel system holds the advantages of both the two types while overcoming their well-known practical performance limitations. The power/mass ratio of the conventional electromechanical pitch system can be enhanced by using a hydraulic motor in the proposed system where the electric motor is used for control, not for actuation in the electromechanical system. The pitch control accuracy of the conventional hydraulic type can be significantly improved in the novel system by

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incorporating a rotary hydraulic servo instead of a slider-crank mechanism. This is because the resulting pitch angle is directly proportional to the angular displacement of the hydraulic servo and can be precisely controlled by an intrinsic hydro-mechanical position closed-loop in the servo. Therefore, this novel system not only holds the advantages such as compactness, high power/mass ratio, higher reliability, and good control precision in particular but also can be utilized in medium or large scale wind turbines with better precision as compared with the conventional pitch systems. Hence, such significant performance improvements make the proposed pitch control system a promising choice for industrial applications.

2. System description

As shown in Fig. 1, this novel pitch control system, consisting of a digital electric motor, a hydraulic servo and a pitch gear set mounted in the nacelle, is an integrated electro-hydraulic position servo system with an intrinsic feedback closed-loop. The digital electric motor acts as a converter transmitting the digital pitch control command to the hydraulic servo. The pitch gear set is used to adapt the high shaft speed of the hydraulic servo to the relatively low pitch rate.

The hydraulic servo consists of a spool type rotational valve, a screw and nut combination and a rotary actuator. The rotational motion from the digital electric motor opens the valve and ports the oil flow from a constant hydraulic power supply to this actuator whose motion is fed back through the screw and nut combination and subtracted from the input motion so as to stroke the valve and close this loop. Hence, such motions automatically create an intrinsic hydro-mechanical position control loop in the hydraulic servo. The rotary actuator can be sized to handle expected pitch loads and has the hydraulic natural frequency large enough to meet overall response requirements. Generally, a hydraulic axial piston motor can be employed as the rotary actuator.

Significant features of this pitch control system are as follows.

(a) The pitch angle control can be eventually achieved by a rotary hydraulic actuator with high payload capability and high power to weight ratio rather than an electric motor or a hydraulic cylinder in other pitch systems. Thus, relatively high pitch control accuracy and power/mass ratio make this system suitable for large-scale wind turbines.
(b) The intrinsic hydro-mechanical closed control loop enables this system to avoid the necessity to measure or feedback the pitch angle signals and hence allow sensor-less pitch control, whereas a large variety of sensors or transducers are always used in other pitch systems.
(c) Compact structure and integration design make this pitch system appropriate for individual pitch control.
(d) This system can be controlled directly by a host computer without using additional controllers or sensors, which provides significantly cost-effective potentials.
(e) The novel pitch control system works in the outer open-loop with an internal hydro-mechanical position control loop and hence has a wider range of pitch rate and lower maintenance cost as compared with other conventional pitch systems.

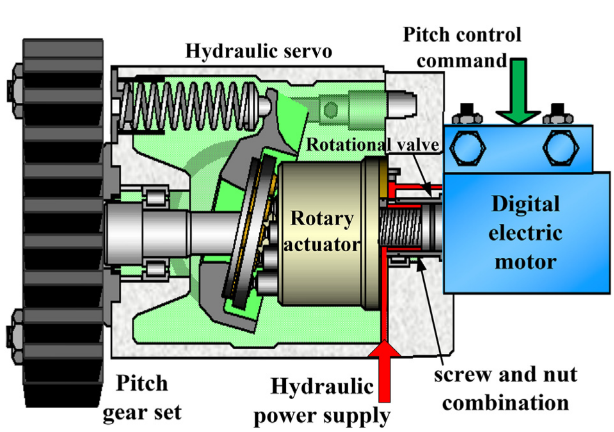

Fig. 1. Schematic of the proposed pitch control system.

3. System design

3.1. Pitch loads

The calculation of pitch loads is first presented as it is an important prerequisite for the system design. Such loads mainly stem from aerodynamics, gravity and dynamic interactions [15]. In particular, the inertia moment arising from the blade centrifugal force is the predominant source of loads associated with the pitch actions and is discussed in detail as follows.

As illustrated in Fig. 2, the plane of rotation is aligned with the axis o-x and perpendicular to the surface of this paper. The pitch axis passes through the center of gravity of each blade cross section and lies in the plane of rotation. The first principal axis of the blade cross sections lies along the chord line for this symmetric aero-foil [16]. Two coordinate systems are established at the same origin O. The reference frame (x, y) is rotated about the frame (x_1, y_1) with the pitch angle β between them.

Consider an incremental part of the blade at a radius r from the rotational axis and the point B with an incremental mass dm. The incremental centrifugal force acting on this point is

$$\mathrm{d}F_c = r_B\omega^2\mathrm{d}m \tag{1}$$

and the radius r_B can be represented as

$$r_B = \frac{L_{OB}\cdot\sin\gamma}{\sin\varphi} \tag{2}$$

Substituting equation (2) into (1) yields

$$\mathrm{d}F_c = \frac{L_{OB}\cdot\sin\gamma}{\sin\varphi}\omega^2\mathrm{d}m \tag{3}$$

where

ω − the angular velocity of the wind rotor;
L_{OB} − the length of the line segment O−B;
γ − the angle between the line O−B and the plane of rotation;
φ − the angle between the pitch axis and the incremental centrifugal force dF_c.

The force dF_c can be decomposed into a component dF_n perpendicular to the blade cross section and a component dF_t parallel to this cross section. The force dF_t can be described as

$$\mathrm{d}F_t = \mathrm{d}F_c\cdot\sin\varphi = \frac{L_{OB}\cdot\sin\gamma}{\sin\varphi}\cdot\omega^2\cdot\mathrm{d}m\cdot\sin\varphi = \omega^2\cdot L_{OB}\cdot\sin\gamma\cdot\mathrm{d}m \tag{4}$$

The resulting inertia moment about the pitch axis due to the centrifugal force is