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Modeling and loading compensation of a rotary valve-controlled pitch system for wind turbines*

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Abstract: A rotary valve-controlled pitch system is proposed to regulate the generator power and smooth power fluctuations for a wind turbine. Design details and dynamic modeling of this pitch system are presented and analyzed. A practical loading compensation approach is synthesized and involved in the pitch system to compensate for the external uncertain pitch loads. The proposed pitch system and loading compensation approach have been experimentally evaluated in terms of generator power smoothing and control accuracy. As demonstrated by the comparative experimental results, the proposed pitch system can be used to significantly smooth the generator power fluctuations and hence to improve the power quality as compared with a servo valve-controlled pitch system under the same operating conditions. The loading compensation approach can also be used to significantly attenuate the effects of external pitch loads and improve the robustness and reliability of the pitch system. The proposed pitch system features good control accuracy and cost-efficiency and hence is attractive for applications in modern large-scale wind turbines.

Key words: Wind turbine; Pitch system; Rotary valve; Loading compensation
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1 Introduction

Pitch systems are increasingly used in medium- to large-scale wind turbines to adjust the turbine's aerodynamic characteristics by adequately changing the blade pitch angle (Anaya Lara *et al*., 2009). When the wind speed exceeds the rated value, the pitch systems can be activated to regulate and limit the output power and torque, thus keeping the power generation within the designed capability. Pitch systems are also effective in protecting wind turbines from damages and hence are crucial in increasing the turbine's reliability and stability. Therefore, more reliable, efficient, and cost-effective pitch systems are essential for significant improvements in wind power quality and economic viability (Yin *et al*., 2014a; 2014b).

The pitch systems can be broadly classified as electric and hydraulic types. The electric pitch system generally consists of an electric motor for regulating the pitch rate and a planetary gear for lowering the rotating speed of the motor shaft. The electric pitch system is usually placed in the rotor hub together with a battery for safety purposes. Although relatively simple and cost effective, the electric pitch system has the disadvantages of relatively low torque/weight ratio and low robustness against wind-loading disturbances. The hydraulic pitch system commonly uses a hydraulic valve-controlled cylinder to change the bladed pitch angle. A slider-crank

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mechanism can also be placed between the end of the cylinder and the blade shaft to transform the linear displacement of the cylinder into the rotational pitch angle (Zhang, 2013). The hydraulic pitch system is advantageous in the relatively large power/weight ratio and high robustness. However, the major drawbacks of this pitch system are the low control accuracy and efficiency, which result in low pitch control effectiveness and relatively high power fluctuations during strong wind gusts.

Regarding the two types of pitch systems, various pitch angle controllers have been designed and presented in the literature. A proportional-integral-plus pitch controller was designed to improve the pitch angle tracking accuracy for a hydraulic pitch system (Zhang, 2013). However, the performance of the designed controller was not assessed using experimental results. A proportional-integral (PI) pitch controller was designed by Wang *et al*. (2011) based on the graphical approach. However, this PI pitch controller generally has low robustness against external loading disturbances and hence has low control efficiency. A hybrid pitch angle controller was proposed by Duong *et al*. (2015) to improve the power quality and transient stability of a fixed-speed wind turbine equipped with a squirrel-cage induction generator. The hybrid pitch angle controller was designed based on the PI control algorithm and fuzzy logic. Dynamic simulation results showed that the hybrid controller could be effective in smoothing output power fluctuations and improving grid voltage stability. An adaptive nonlinear sliding mode pitch angle controller and an integrated pitch control strategy were proposed by Yin *et al*. (2014a; 2014b) to smooth output power and load fluctuations for large-scale wind turbines. In the study of Belghazi and Cherkaoui (2012), a pitch angle controller was designed by using the genetic algorithm to reduce blade loads and improve generator power quality for wind turbines. However, the controller may not be implemented in real time because the genetic algorithm is computationally complex in general. A pitch angle controller was proposed by Abdelkafi and Krichen (2011) to ensure a balance between wind power production and energy consumption for a wind farm. The overall controller mainly consisted of a local controller for protecting the wind turbines against mechanical failure and a global controller capable of balancing the total power production and demand. An on-line training radial basis function network-based pitch angle controller was designed in the study of Lin *et al*. (2011) to regulate the aerodynamic power in wind turbines. The back-propagation learning algorithm was used to regulate the pitch angle controller. A pitch angle controller using a generalized predictive controller was proposed (Senjvu *et al*., 2006) to reduce output power fluctuations for a wind farm in all operating regions. However, the pitch angle controller was not experimentally evaluated. Other pitch angle controllers mainly include the fuzzy logic controller (Duong *et al*., 2014), neural network-based adaptive pitch controller (Poultangari *et al*., 2012), and preview-based feedforward pitch controller (Kragh and Hansen, 2012). However, these pitch angle controllers usually require very complex mathematical derivations and considerable calculation efforts and hence are not suitable for integration into a digital controller for real-time implementation.

In this paper, a rotary valve-controlled pitch system is proposed to better smooth the generator power fluctuations for a wind turbine. A pragmatic loading compensation approach is also proposed for the pitch system to attenuate the effects of external pitch loads, allowing significant improvements in the pitch control accuracy and robustness. Unlike the above-mentioned two types of pitch systems, a hydraulic motor is used in the proposed pitch system to pitch the turbine blade through a pitch gear and hence improve the final control accuracy and torque/weight ratio. The proposed pitch system does not require additional sensors, thereby reducing the overall system cost and complexity. Comparative experimental results are presented to validate the performances and effectiveness of the proposed pitch system and the loading compensation approach for generator power smoothing, control accuracy, and robustness.

2 System design

As illustrated in Fig. 1, the proposed pitch system is an integrated electrohydraulic servo system and mainly comprises an electric motor, a rotary valve, a hydraulic motor, and a pitch gear. The digital electric motor can be used to activate the rotary valve by transmitting the pitch command to the rotational displacement of this valve. The rotary valve uses a four-way spool type construction and has a critical center. The valve also has a linear gain in the vicinity of the neutral point and usually necessitates a slight overlap to offset the effects of radial clearance. The right end of the valve spool is reshaped into the screw to mesh with the nut that is rigidly connected to the input shaft of the hydraulic motor. Hence, this screw-and-nut combination can reasonably serve as the inherent feedback connection between the valve and the hydraulic motor. The hydraulic motor has fixed displacement and can be reasonably sized to deal with relatively large pitch loads and external disturbances. The pitch gear is used to match the relatively low blade pitch rate with the shaft speed of the hydraulic motor. However, this pitch gear is not necessarily required because the rotating speed of the hydraulic motor can be made relatively low to match the low pitch rate. Thus, the hydraulic motor can be directly coupled to the blade shaft to adjust the pitch angle and hence to significantly improve the reliability and availability of the pitch system. The gear issues can also be eliminated by the careful and regular maintenance of the pitch system. Furthermore, this pitch system can also directly provide sufficient hydraulic oil flow to maintain a lubrication oil film on the pitch gear teeth and the rolling elements of bearings to minimize surface pitting and wear (abrasion, adhesion, and scuffing).

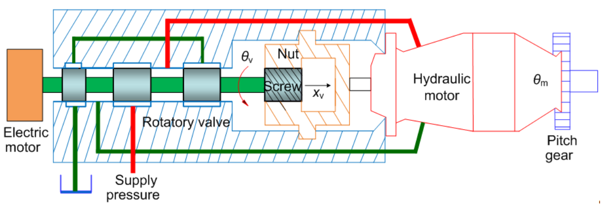

Fig. 1 Schematic representation of the proposed pitch system (the variables will be explained in the text)

When the pitch system is energized, the digital electric motor proportionally converts the pitch command into the displacement of the rotary valve to port the oil flow from the pressure supply to the hydraulic motor. Hence, the hydraulic motor can be activated and can change the blade pitch angle through the pitch gear. Because the shaft of the hydraulic motor is directly coupled to the valve spool through the screw-and-nut combination, the pitch motions of the hydraulic motor can be directly fed back through this combination and can then be subtracted from the valve motions to nullify the position of the rotary valve. Therefore, the overall pitch control actions create an inherent hydro-mechanical closed loop between the hydraulic motor and the rotary valve. The desired pitch angle can also be accurately maintained in proportion to the pitch command by carefully designing the screw-and-nut combination.

3 Dynamic modeling

The rotational displacement of the rotary valve can be described as

$$\theta_v = K_e \cdot u, \tag{1}$$

where K_e is the amplification gain related to the electric motor and pitch gear, and θ_v and u denote the angular displacement of the valve spool and the pitch command, respectively.

The sliding displacement of the valve spool can be obtained by using the screw-and-nut combination. Thus,

$$x_v = \frac{t_p}{2\pi}(\theta_v - \theta_m), \tag{2}$$

where x_v, t_p, and θ_m denote the linear displacement of the valve spool, the helical pitch of the screw-and-nut combination, and the rotational displacement of the hydraulic motor, respectively.

The flow equation of the rotary valve is

$$Q_L = K_q x_v - K_c P_L, \tag{3}$$

where Q_L and P_L denote the hydraulic oil flow rate and the load pressure difference of the hydraulic motor, respectively, and K_q and K_c denote the valve flow gain and the valve flow pressure coefficient, respectively.

The continuity equation of the hydraulic motor chambers can be formulated as follows (Merritt, 1967):

$$Q_{\mathrm{L}} = D_{\mathrm{m}} s \cdot \theta_{\mathrm{m}} + C_{\mathrm{t}} P_{\mathrm{L}} + \frac{V_{\mathrm{t}}}{4\beta_{\mathrm{e}}} s P_{\mathrm{L}}, \tag{4}$$

where s denotes the Laplace operator, β_{e} denotes the effective bulk modulus, and V_{t}, C_{t}, and D_{m} denote the total volume, the total leakage coefficient, and the volumetric displacement of the hydraulic motor, respectively.

The pitch torque balance equation of the hydraulic motor is

$$P_{\mathrm{L}} D_{\mathrm{m}} = J_{\mathrm{t}} \cdot s^2 \cdot \theta_{\mathrm{m}} + K_{\mathrm{g}} \cdot \theta_{\mathrm{m}} + T_{\mathrm{L}}, \tag{5}$$

where T_{L} denotes the pitch loads, J_{t} and K_{g} denote the total inertia and torsional spring gradient of the hydraulic motor, pitch gear, and blade, respectively.

Therefore, the resulting pitch angle can be derived as

$$\beta = \frac{\theta_{\mathrm{m}}}{i_{\mathrm{g}}}, \tag{6}$$

where β and i_{g} denote the pitch angle and the pitch gear ratio, respectively.

The above Eqs. (1)–(5) define the basic characteristics of this rotary valve-controlled hydraulic motor combination and can be combined to yield the block diagram representation of this pitch system as shown in Fig. 2 (Jelali and Kroll, 2003).

4 Loading compensation

The final pitch angle β can be reasonably derived based on Fig. 2. Thus,

$$\beta = \left[\frac{K_{\mathrm{e}} K_{\mathrm{q}} t_{\mathrm{p}}}{2\pi \cdot i_{\mathrm{g}}} D_{\mathrm{m}} \cdot u - \left(\frac{V_{\mathrm{t}}}{4\beta_{\mathrm{e}}} s + K_{\mathrm{c}} + C_{\mathrm{t}}\right) \cdot \frac{T_{\mathrm{L}}}{i_{\mathrm{g}}}\right] \Bigg/ \left[\frac{J_{\mathrm{t}} \cdot V_{\mathrm{t}}}{4\beta_{\mathrm{e}}} s^3 + J_{\mathrm{t}} \cdot \left(K_{\mathrm{c}} + C_{\mathrm{t}}\right) s^2 + \left(\frac{K_{\mathrm{g}} \cdot V_{\mathrm{t}}}{4\beta_{\mathrm{e}}} + D_{\mathrm{m}}^2\right) s + K_{\mathrm{g}} \cdot \left(K_{\mathrm{c}} + C_{\mathrm{t}}\right) + \frac{t_{\mathrm{p}}}{2\pi} D_{\mathrm{m}} K_{\mathrm{q}}\right], \tag{7}$$

where K_{e} denotes a constant gain.

Eq. (7) represents the dynamic response of the pitch angle to both the pitch command and the pitch loads. The first term in the numerator can be identified as the no load condition, and the second term gives the pitch angle drop due to the pitch loads. As illustrated in this equation, the resulting pitch angle β not only depends on the control command u but also highly relies on the external pitch loads T_{L}. Because the pitch loads are generally time varying and arbitrary, their adverse effects could severely deteriorate the final pitch angle control accuracy, leading to undesirable control errors and even severe faults on the pitch system.

Therefore, an adequate loading compensation is necessary for enhancing the pitch angle control accuracy and improving the overall pitch control performances. As such, a loading compensation function is designed and involved in the pitch system to eliminate the adverse effects caused by the pitch loads T_{L}.

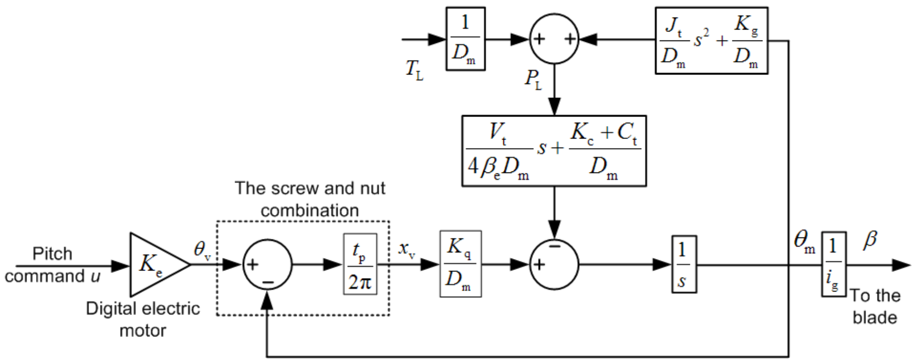

Fig. 2 Block diagram of the proposed pitch system

Considering the load pressure difference P_L as the measurable variable and adding the loading compensation function to its output, the compensation function has the following form:

$$F(s)=\frac{2\pi}{K_q t_p}\cdot\left(\frac{V_t}{4\beta_e}s+K_c+C_t\right), \tag{8}$$

where $F(s)$ denotes the loading compensation function.

Considering this compensation function, the block diagram of the pitch system described in Fig. 2 can be reformatted to the one shown in Fig. 3.

Hence, the compensated pitch angle can be derived based on Fig. 3:

$$\beta=\frac{K_e K_q t_p}{2\pi D_m i_g s+K_q t_p i_g}u. \tag{9}$$

As illustrated in Eq. (9), the compensated pitch angle can be accurately controlled based on the pitch command and will not be affected by the variations of the pitch loads. Therefore, this loading compensation design can easily lead to better controllability of the final pitch angle and hence can significantly improve the overall pitch control accuracy (Bishop and Dorf, 2004).

In practice, the loading compensation can be made implementable by reasonably rearranging the block diagram in Fig. 3 and obtaining its equivalent block diagram in Fig. 4. Therefore, the loading compensation can be readily achieved by adding the compensation signal to the input pitch command by using the software in a computer without necessitating additional hardware.

The loading compensation function $F_1(s)$ in Fig. 4 can be equivalently derived from Fig. 3. Thus,

$$F_1(s)=\frac{F(s)}{K_e}=\frac{2\pi}{K_e K_q t_p}\cdot\left(\frac{V_t}{4\beta_e}s+K_c+C_t\right). \tag{10}$$

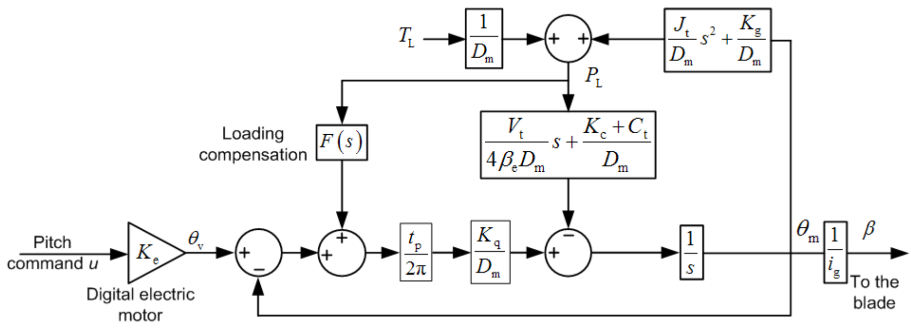

Fig. 3 Block diagram of the proposed pitch system with loading compensation

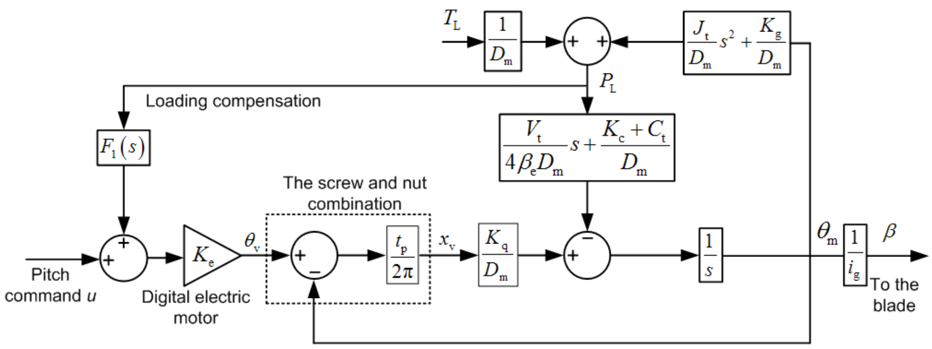

Fig. 4 Pragmatic loading compensation of the proposed pitch system

5 Experimental results and discussion

The proposed pitch system and loading compensation approach have been experimentally evaluated in terms of power smoothing and control accuracy.

5.1 Experimental setups

An experimental setup has been designed and built to evaluate the performances of the proposed pitch system. As illustrated in Fig. 5, the experimental setup mainly consists of a wind turbine emulator, a host computer, and the proposed pitch system.

The wind turbine emulator is used to replicate the dynamic characteristics of a 1.5 MW pitch-controlled wind turbine. The emulator mainly comprises a wind rotor, a target computer, and a loading system. The wind rotor is driven by a torque-controlled electric motor and is used to simulate the effects of actual pitch actions and pitch loads. The rotor blades are designed to have the same mass and moment of inertia as that of the 1.5 MW wind turbine. The loading system mainly consists of a set of valve-controlled single-rod hydraulic cylinders and is used to exert the pitch loads to the blades. The target computer is used to control the loading system and model other necessary subsystems of this 1.5 MW wind turbine by using the GH Bladed software (Bossanyi, 2000; Yin *et al.*, 2015).

The host computer is used to control the proposed pitch system and display the necessary operating parameters, such as the generator power, the pitch angle, and the wind speed for condition monitoring.

A PI reference pitch angle controller is implemented in the host computer to generate the pitch command u.

This reference pitch angle controller can be represented as

$$u = k_{\mathrm{p1}}\left(P_{\mathrm{gR}} - P_{\mathrm{g}}\right) + k_{\mathrm{i1}}\int_0^t \left(P_{\mathrm{gR}} - P_{\mathrm{g}}\right)\mathrm{d}t, \qquad (11)$$

where P_{g} and P_{gR} denote the actual generator power and the rated generator power, respectively, and k_{p1} and k_{i1} denote the proportional and integral gains, respectively.

The proposed rotary valve-actuated pitch (RVP) system has also been compared with a servo valve-controlled pitch (SVP) system under the same operating conditions to verify its control performances. As shown in Fig. 6, the SVP system has been implemented in the same experimental setup and mainly uses a servo valve-controlled hydraulic motor to generate the pitch actions. The servo valve is used instead of the rotary valve to control the hydraulic motor and hence to change the pitch angle through the pitch gear. A PI pitch angle controller has also been designed in the host computer to control the SVP system.

This PI controller can be described as

$$u_{\mathrm{c}} = k_{\mathrm{p2}}\cdot(u-\beta) + k_{\mathrm{i2}}\cdot\int_0^t (u-\beta)\mathrm{d}t, \qquad (12)$$

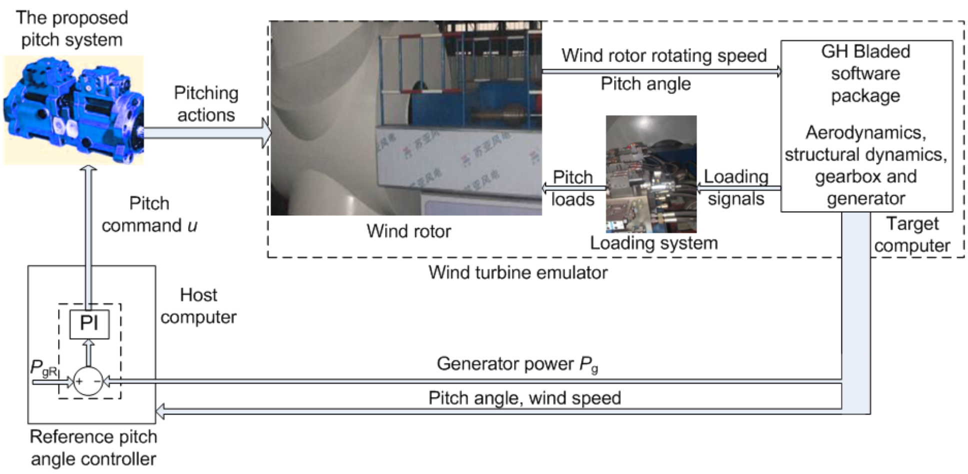

Fig. 5 Experimental setup of the proposed pitch system

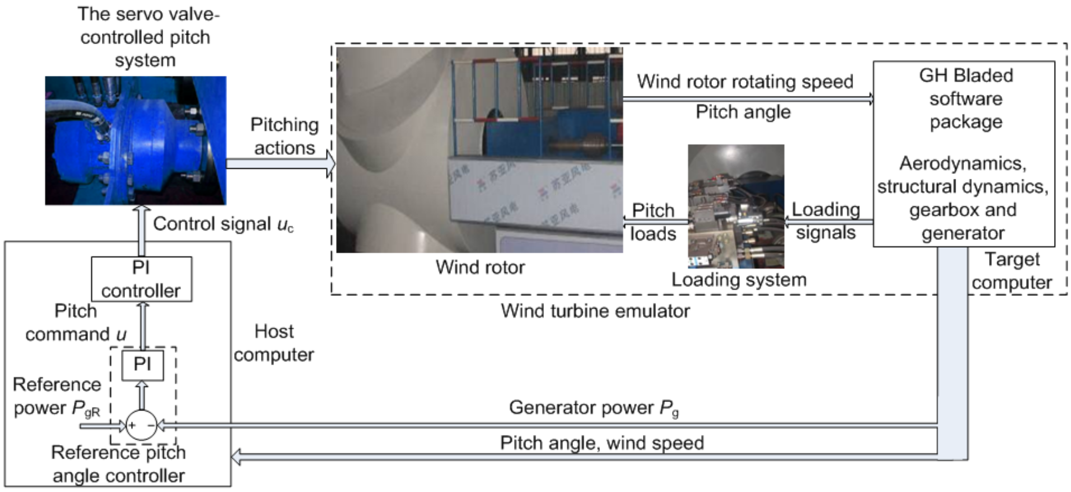

Fig. 6 Experimental setup of the SVP system

where u_c denotes the pitch control signal for the SVP system, and k_{p2} and k_{i2} denote the proportional and integral gains, respectively.

The proportional and integral gains of the two PI pitch angle controllers can be automatically tuned by using the Ziegler–Nichols tuning method (Astrom and Hagglund, 1995) until the two pitch systems have almost the same pitch control performances at the start of the experiments under the same operating conditions. The same reference pitch angle controller should be utilized for the two pitch systems for fair comparisons.

5.2 Power smoothing under the rated wind speed

As illustrated in Fig. 7a, a 65-s realistic wind speed time series was used to evaluate the performances of the two pitch systems. The wind speed ranges from 6.5 m/s to 10.8 m/s and has the averaged value of 8.6 m/s.

As shown in Fig. 7b, the pitch angles generated from the proposed RVP system have relatively large values than those from the SVP system. The RVP system exhibits more pitch control than the SVP system to better regulate the pitch angle and generator power under the rated wind speed.

As shown in Fig. 7c, the generator power varies significantly from 0.9 MW to 1.12 MW when the SVP system is applied, whereas the generator power can be well maintained below 1.1 MW and the power fluctuations can be significantly reduced when the proposed RVP system is used. Therefore, the proposed RVP system can be used to significantly smooth the generator power fluctuations and hence

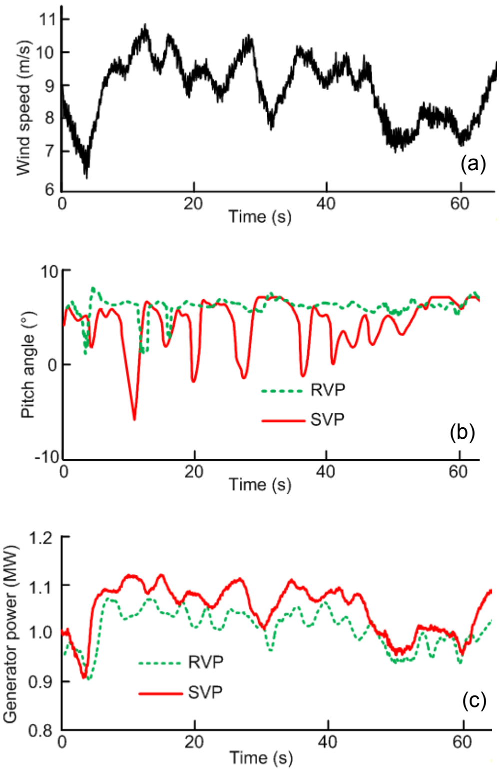

Fig. 7 Wind speed time series under the rated wind speed of 12 m/s (a), pitch angle variations under the rated wind speed (b), and generator power variations under the rated wind speed (c)

to improve the power quality under the rated wind speed as compared with the SVP system.

5.3 Power smoothing above the rated wind speed

As shown in Fig. 8a, a 65-s wind speed time series has also been used to test the control performances of the two pitch systems above the rated wind speed of 12 m/s. The wind speed changes significantly between 12 m/s and 18 m/s and has the mean value of 15.8 m/s.

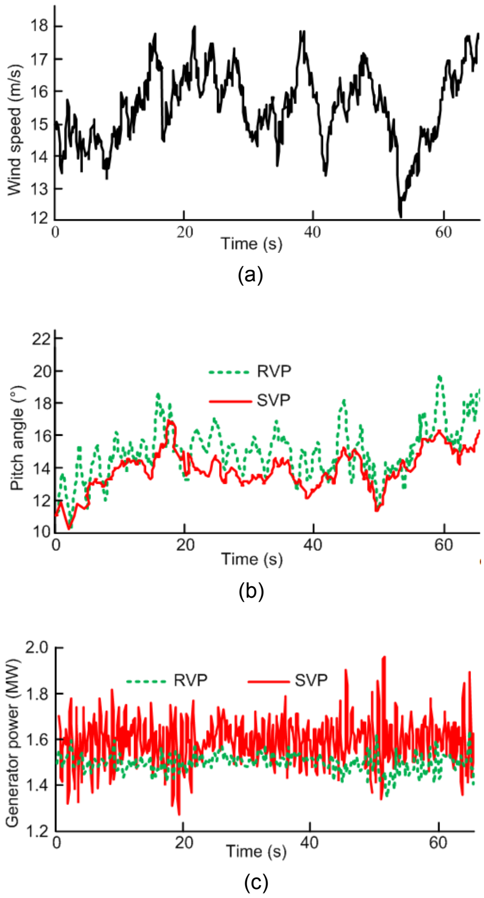

Fig. 8 Wind speed time series above the rated wind speed of 12 m/s (a), pitch angle variations above the rated wind speed (b), and generator power variations above the rated wind speed (c)

As shown in Fig. 8b, the SVP system generates relatively little pitch actions for controlling the generator power, whereas the proposed RVP system exerts more pitch control actions and the resulting pitch angle varies more significantly between 10° and 20° than the SVP system. Therefore, the proposed RVP system has a faster pitch control response and can generate more control actions for better regulating the generator power as compared with the SVP system.

As illustrated in Fig. 8c, the generator power can be well maintained around the rated value of 1.5 MW by using the proposed RVP system, whereas the power fluctuates significantly between 1.28 MW and 1.88 MW when the SVP system is applied. This comparative result clearly indicates that the proposed RVP system can be used to better maintain the rated generator power and hence improve the power stability above the rated wind speed as compared with the SVP system.

5.4 Comparative results of loading compensation

Figs. 9a and 9b explicitly illustrate the comparative results of the loading compensation approach for the proposed RVP system subject to external pitch loads. As illustrated in Fig. 9a, the proposed RVP system has a relatively fast step response and can achieve the steady state within 0.5 s. Before

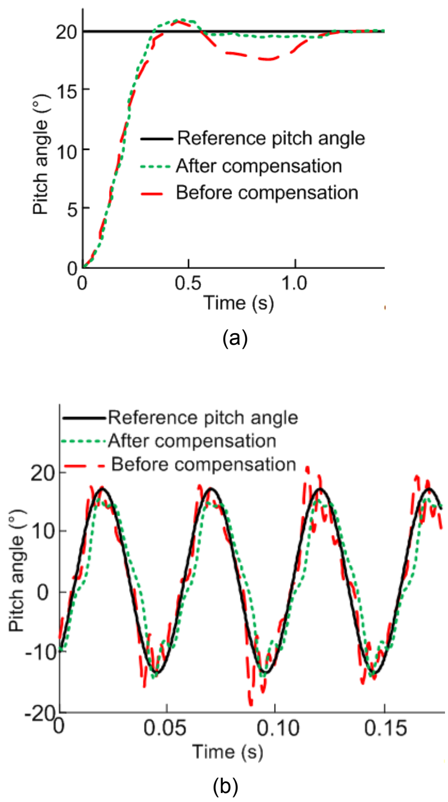

Fig. 9 Step response (a) and sinusoidal response (b) of the proposed RVP system

compensation, the pitch angle deviates significantly from its steady-state value due to the large pitch loads, whereas the resulting pitch angle can be well maintained at the steady-state value of 20° after compensation, demonstrating the strong robustness of the loading compensation against pitch loads.

As shown in Fig. 9b, the proposed RVP system can be well commanded to track the reference sinusoidal input with frequency of 20 Hz after the loading compensation, whereas the sinusoidal response of the RVP system without compensation could be significantly distorted. The added large pitch loads do not affect the pitch angle tracking performances of the RVP system when the loading compensation approach is applied. The reference sinusoidal pitch angle can also be tracked at steady state with sufficient precision in spite of large pitch loads, verifying the effectiveness and robustness of the loading compensation approach to external pitch loads.

6 Conclusions

A rotary valve-controlled pitch system has been presented in this paper to regulate the generator power and smooth the power fluctuations for a wind turbine. Dynamic modeling and the operating principle of the pitch system have been presented. A loading compensation approach has also been presented to guarantee a prescribed pitch angle tracking performance while achieving significant disturbance attenuation in the presence of large pitch loads. Furthermore, comparative experimental results have also been presented to demonstrate the effectiveness and the strong performance robustness of the proposed pitch system and compensation approach in power smoothing and disturbance rejection. All the experimental results have demonstrated that the proposed pitch system can significantly reduce the generator power fluctuations, in addition to improving the power stability and the performance robustness as compared with a servo valve-controlled pitch system under different wind speed conditions. The proposed pitch system is generally characterized by compact construction, high efficiency and power density, and light weight. Therefore, the proposed pitch system is obviously promising and attractive to large-scale wind turbine applications.

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中文概要

题　目： 旋转阀控马达变桨距的建模与载荷补偿的对比性实验研究

目　的： 风速的随机瞬变容易导致风电机组输出功率的大幅波动。本文拟采用数字旋转阀控马达变桨距技术，以有效地提高各种风况下的风电机组输出功率平滑控制的性能，并采用变桨载荷补偿的方法，以提高变桨距控制的精度和抗干扰能力。

创新点： 1. 提出数字式旋转阀控马达变桨距的控制技术，推导得出变桨距控制的模型；2. 提出变桨距载荷补偿的控制方法，以提高变桨距控制的精度和鲁棒性；3. 搭建实验台，并与伺服阀控马达变桨距进行功率平滑控制的对比性研究。

方　法： 1. 通过理论推导，构建旋转阀控马达变桨距的机理模型，得到输出桨距角与输入控制信号之间的定量关系；2. 通过对比性实验分析，验证旋转阀控马达变桨距在输出功率平滑控制方面的高效性。

结　论： 1. 相比于伺服控马达，数字式旋转阀控马达变桨距能更有效地平滑风电机组的输出功率，提高输出功率的稳定性和质量；2. 变桨载荷补偿的方法能更为有效地提高旋转阀控马达变桨距控制的精度、响应速度和鲁棒性。

关键词： 风电机组；变桨距系统；旋转阀；载荷补偿