Design of Sliding Mode Stabilizer for Wind Turbine Generator using Dynamic Compensation Observer Technique

ISSARACHAI NGAMROO
Sirindhorn International Institute of Technology
Thammasat University, Bangkok
THAILAND

Abstract: - A sliding mode stabilizer design for enhancement of the dynamic stability of a wind turbine generator supplying an infinite bus is presented. The dynamic compensation observer technique is applied to design the hyperplane in a sliding mode control. The salient feature of this technique is that only output information can be used as a feedback signal of a controller. In addition, this technique provides a systematic approach to obtain the control design specification, a robustness against system parameter variations as well as various loading conditions, and an enhancement of system dynamic performance against disturbances. Simulation study reveals that the proposed controller is not only able to damp power oscillations due to wind gust disturbances, but also very robust against various loading conditions and variations of system parameters.

Key-Words: - Power system dynamic stability, Sliding mode control, Wind turbine generator, Dynamic compensation observer

1 Introduction

The development of wind energy technology is growing rapidly throughout the world. As the oil price keeps increasing and the power demand also continues to rise, wind energy seems to become a new aspect to solve these problems. Wind energy provides many advantages to power system as it is a clean energy and it does not require a major redesign of the existing power system. However, the fluctuating nature of wind as well as the new generator types that are used in wind turbines, are the challenges that power system engineers encounters [1].

One of the areas that has been paid attentions by many researchers is the dynamic stability of the wind turbine generators [2-3]. The important aspect of the dynamic performance of the wind turbine generator is its effect on the dynamic stability of the system to which it is connected. The wind power generation unit should not contribute the dynamic instability of the power system. To improve system dynamic stability, many control theories have been adopted to design of power system stabilizers for a wind turbine system such as a proportional integral controller [4], a variable-structure stabilizer [5], etc. Among these schemes, a sliding mode control is one of advanced design methods. It offers the satisfactory attenuation performance against disturbance as well as the high robustness to a power system under various operating conditions and system uncertainties [6-7]. Nevertheless, the sliding mode controller is constructed by a state feedback control scheme. Accordingly, it is not easy to implement in practical system. Besides, the difficulty of switching gain setting in a hyperplane design is also an inevitable problem. To overcome these problems, a new technique for sliding mode control design is highly expected.

With the system where only output information of system is available, a hyperplane design based on a dynamic compensation observer [7] is applied to design a sliding mode stabilizer for a wind turbine generator in this paper. The main feature of this technique is that only output information can be used as a feedback signal of a controller. Additionally, the system robustness against various uncertainties such as system parameter variations, generating and loading conditions etc., can be guaranteed. Simulation studies in a single wind turbine generator infinite bus system insist that the proposed sliding mode control is not merely capable of damping low frequency oscillations, but also very robust to system uncertainties.

2 System Modeling

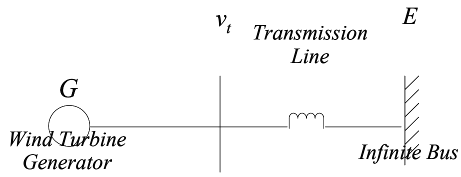

Fig.1 Single wind turbine generator infinite bus.

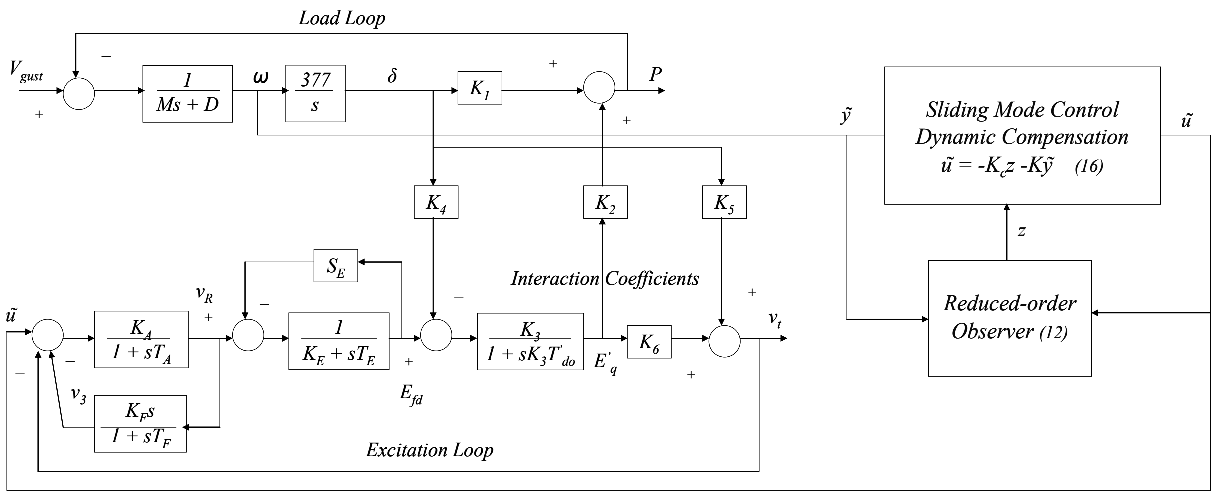

Fig.2 Block diagram of a linearized power system model with a dynamic compensation controller.

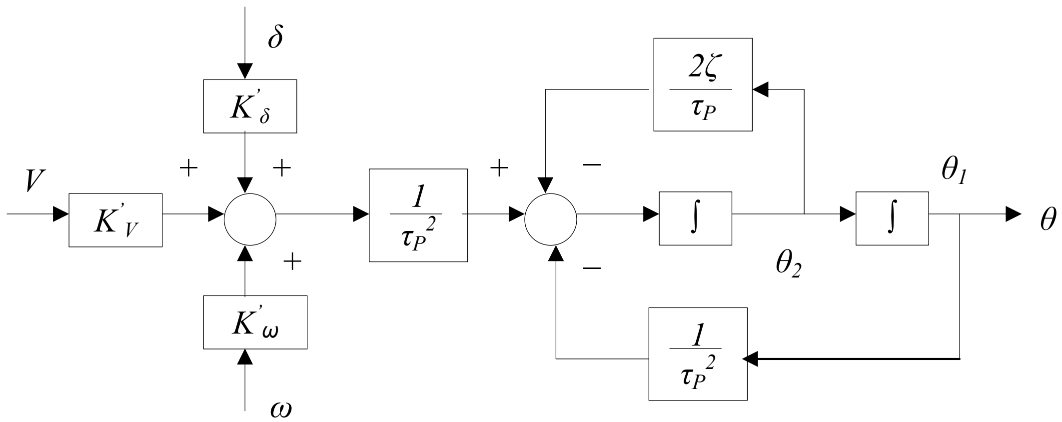

Fig.3 Block diagram of a pitch change regulation.

A wind turbine generator connected to an infinite bus [5] as shown in Fig.1 is used as a studied system. The linearized model of power system can be depicted in Fig.2. The block diagram of a pitch change regulation is shown in Fig. 3. Nomenclatures are given by

δ - Rotor angle
ω - Electrical speed
T_m - Mechanical input torque
x_e - Transmission line impedance
D - Damping constant of the generator
M - Inertia constant of the generator
T_A - Time constant of the amplifier
E_{fd} - Field voltage
E_q' - Transient EMF in q -axis
K_F - Stabilizer gain
K_E - Exciter gain
T_E - Time constant of the exciter
K_A - Amplifier gain
T_F - Time constant of the stabilizer loop
T_{do}' - Open circuit field time constant
V - Wind speed
θ - Blade pitch angle
$K_V = \frac{dT_m}{dV}$ - Coefficient of torque from wind speed
$K_\theta = \frac{dT_m}{d\theta}$ - Coefficient of torque from pitch angle
$K_\omega = \frac{dT_m}{d\omega}$ - Coefficient of torque from rotational speed
$K_\omega', K_v', K_\delta'$ - Pitch angle regulation constants
ζ - Damping ratio
τ_P - Actuator time constant
$K_1 - K_6$ - System constants

The state equation of a linearized system in Fig. 2 can be expressed as

$$\dot{x} = \begin{bmatrix} A_1 & A_3 \\ A_2 & A_4 \end{bmatrix} x + Bu \qquad (1)$$

where

$$x^T = \begin{bmatrix} \Delta\delta & \Delta\omega & \Delta E_q^{'} & \Delta E_{fd} & \Delta v_R & \Delta v_3 & \Delta\theta_1 & \Delta\theta_2 \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & 0 & 0 & 0 & \frac{K_A}{T_A} & \frac{K_A K_F}{T_A T_F} & 0 & 0 \end{bmatrix}$$

$$A_1 = \begin{bmatrix} 0 & 377 & 0 & 0 \\ -\frac{K_1}{M} & \frac{K_\omega - D}{M} & -\frac{K_2}{M} & 0 \\ -\frac{K_4}{T_{do}^{'}} & 0 & -\frac{1}{K_3 T_{do}^{'}} & \frac{1}{T_{do}^{'}} \\ 0 & 0 & 0 & -\frac{S_E + K_E}{T_E} \end{bmatrix}$$

$$A_2 = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{K_\theta}{M} & 0 \\ 0 & 0 & 0 & 0 \\ \frac{1}{T_E} & 0 & 0 & 0 \end{bmatrix}$$

$$A_3 = \begin{bmatrix} -\frac{K_A K_5}{T_A} & 0 & -\frac{K_A K_6}{T_A} & 0 \\ -\frac{K_A K_5 K_F}{T_A T_F} & 0 & -\frac{K_A K_6 K_F}{T_A T_F} & 0 \\ 0 & 0 & 0 & 0 \\ \frac{K_\delta^{'}}{\tau_P^{\,2}} & \frac{K_\omega^{'}}{\tau_P^{\,2}} & 0 & 0 \end{bmatrix}$$

$$A_4 = \begin{bmatrix} -\frac{1}{T_A} & -\frac{K_A}{T_A} & 0 & 0 \\ -\frac{K_F}{T_A T_F} & -\frac{T_A + K_A K_F}{T_A T_F} & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -\frac{1}{\tau_P^{\,2}} & -\frac{2\zeta}{\tau_P^{\,2}} \end{bmatrix}$$

The output y is

$$\begin{bmatrix} \Delta\delta \\ \Delta\omega \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \Delta\delta \\ \Delta\omega \\ \Delta E_q^{'} \\ \Delta E_{fd} \\ \Delta v_R \\ \Delta v_3 \\ \Delta\theta_1 \\ \Delta\theta_2 \end{bmatrix} \qquad (2)$$

3 Design of Sliding Mode Stabilizer using Dynamic Compensation Observer

The design of a sliding mode stabilizer based on a dynamic compensation observer [7] is divided into 2 main parts; the system matrix partition and the hyperplane gains design. As demonstrated in Fig. 2, the speed deviation $\Delta\omega$ which is an output signal from the system is used as a feedback signal of the sliding mode stabilizer and a reduced-order observer, to generate a control signal $\tilde{u}$. This signal is used to stabilize the system.

3.1 The partition of system matrix

The state equations (1) and (2) are expressed as

$$\dot{x} = Ax + Bu \qquad (3)$$
$$y = Cx \qquad (4)$$

where, $A \in R^{n\times n}$ is a system matrix, $B \in R^{n\times m}$ is an input matrix, $C \in R^{p\times n}$ is an output matrix, $x \in R^n$ is the state variables, $u \in R^m$ is a control signal and $y \in R^p$ is an output signal with $m \le p < n$.

From (3), the system matrix A is split into the following

$$A = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \qquad (5)$$

where $A_{11} \in R^{(n-m)\times(n-m)}$. This sub-block can be divided into

$$A_{11} = \left[\begin{array}{cc|c} A_{11}^0 & A_{12}^0 & A_{12}^m \\ 0 & A_{22}^0 & \\ \hline 0 & A_{21}^0 & A_{22}^m \end{array}\right] \qquad (6)$$

Define r to be a dimension of A_{11}^0 and $r > 0$. Then, $A_{11}^0 \in R^{r\times r}$, $A_{22}^0 \in R^{(n-p-r)\times(n-p-r)}$ and $A_{21}^0 \in R^{(p-m)\times(n-p-r)}$. Divide the submatrices A_{12} and A_{12}^m as

$$A_{12} = \begin{bmatrix} A_{121} \\ A_{122} \end{bmatrix} \text{ and } A_{12}^m = \begin{bmatrix} A_{121}^m \\ A_{122}^m \end{bmatrix} \quad (7)$$

where $A_{122} \in R^{(n-m-r)\times m}$, $A_{122}^m \in R^{(n-p-r)\times(p-m)}$ and split A_{122} and A_{21} into

$$A_{122} = \begin{bmatrix} A_{1221}^{(n-p-r)} \\ A_{1222}^{(p-m)} \end{bmatrix}, A_{21} = \begin{bmatrix} A_{211} & A_{212} & A_{213} \end{bmatrix} \quad (8)$$

where $A_{211} \in R^{(n-m+1:n)\times(1:r)}$, $A_{212} \in R^{(n-m+1:n)\times(r+1:n-p)}$, $A_{213} \in R^{(n-m+1:n)\times(n-p+1:n-m)}$ and $A_{22} \in R^{(n-m+1:n)\times(n-m+1:n)}$. Finally, define

$$\tilde{A}_{11} = \begin{bmatrix} A_{22}^0 & A_{122}^m \\ A_{21}^0 & A_{22}^m \end{bmatrix} \text{ and } \tilde{C}_1 = [0_{(p-m)\times(n-p-r)} \quad I_{(p-m)}] \quad (9)$$

The pair $(\tilde{A}_{11}, A_{122})$ is controllable and the triple $(\tilde{A}_{11}, A_{122}, \tilde{C}_1)$ is stabilizable with respect to output feedback. As a result, the original systems (3) and (4) are reduced to the fictitious systems as

$$\dot{\tilde{x}} = \tilde{A}_{11}\tilde{x} + A_{122}\tilde{u} \quad (10)$$

$$\tilde{y} = \tilde{C}_1\tilde{x} \quad (11)$$

The systems (10) and (11) are used to design sliding mode control. The signal $\tilde{y}$ is an actual output feedback from the system which will be used as an input for the controller and the observer as depicted in Fig.2.

3.2 Design of compensator variables and hyperplane gains

The design of dynamic compensator variables and hyperplane gains can be explained as follows:

1) Specify a new damping ratio (ζ) and a new real part of the eigenvalue corresponding to the power oscillation mode.
2) Define a gain matrix $L^0 \in R^{(n-p-r)\times(p-m)}$ so that $A_{22}^0 + L^0 A_{21}^0$ is stable. Select $L^0 = p_1$ where p_1 is a $(n-p-r)$ vector representing the desired poles of $A_{22}^0 + L^0 A_{21}^0$. The values of p_1 is usually real negative values. Define p_2 as a $(n-m-r)$ vector representing the desired poles of $\tilde{A}_{11} - A_{122}[K_1 \;\; K_2]$. The values of p_2 may be real negative numbers. Adjust p_2 to obtain $[K_1 \;\; K_2]$ by using a pole placement method.
3) Define the reduced-order observer [8] for the system in (10) and (11) as

$$\dot{z} = Hz + D_1\tilde{y} + D_2\tilde{u} \quad (12)$$

The compensator variables in (12) are obtained from

$$H = A_{22}^0 + L^0 A_{21}^0 \quad (13)$$

$$D_1 = A_{122}^m + L^0 A_{22}^m - (A_{22}^0 + L^0 A_{21}^0)L^0 \quad (14)$$

$$D_2 = A_{1221} + L^0 A_{1222} \quad (15)$$

The state feedback law is provided by

$$\tilde{u} = -K_1 z - (K_2 - K_1 L^0)\tilde{y} = -K_C z - K\tilde{y} \quad (16)$$

where the hyperplane gains K and K_C are given by

$$K = K_2 - K_1 L^0 \quad (17)$$

$$K_C = K_1 \quad (18)$$

4 Simulation Study

Based on system parameters [5], the operating point selected for the dynamic compensation design is $(P,Q) = (1.0, -0.3)$ pu. The system matrix A and the input matrix B are given as

$$A_1 = \begin{bmatrix} 0 & 377 & 0 & 0 \\ -0.1031 & -0.1733 & -0.1220 & 0 \\ -2.4454 & 0 & -2.7795 & 0.5149 \\ 0 & 0 & 0 & -1.2615 \end{bmatrix},$$

$$A_2 = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0.0062 & 0 \\ 0 & 0 & 0 & 0 \\ 0.7692 & 0 & 0 & 0 \end{bmatrix},$$

$$A_3 = \begin{bmatrix} -269.1 & 0 & -6954.4 & 0 \\ -8.1 & 0 & -208.6 & 0 \\ 0 & 0 & 0 & 0 \\ -0.2 & -930.7 & 0 & 0 \end{bmatrix},$$

$$A_4 = \begin{bmatrix} -50 & -20000 & 0 & 0 \\ -1 & -601 & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -44 & -651 \end{bmatrix}$$

$$B^T = \begin{bmatrix} 0 & 0 & 0 & 0 & 20000 & 600 & 0 & 0 \end{bmatrix} \quad (19)$$

From section 3.1, $n = 8, m = 1, p = 2$ and $r = 5$ are obtained. The eigenvalues corresponding to the electromechanical oscillation mode are $0.0735 \pm 1.91i$ which are unstable. In order to acquire system stability, the new real part and the new damping ratio of these eigenvalues are specified at -0.9 and 0.57, respectively. As a result, the matrices $\tilde{A}_{11}$ and A_{122} are calculated as

$$\tilde{A}_{11} = \begin{bmatrix} -601 & 0 \\ 0 & 0 \end{bmatrix} \text{and } A_{122} = \begin{bmatrix} 0 \\ 1 \end{bmatrix} \quad (20)$$

To reach the desired specifications above, the value p_1 is adjusted to be -75 and $[K_1 \quad K_2]$ are determined as $\begin{bmatrix} -0.0005 & -50 \end{bmatrix}$, respectively. The hyperplane gains are calculated as

$$K = -50.0375 \text{ and } K_C = -0.0005 \quad (21)$$

According to equation (11), the actual output signal $\tilde{y}$ is as follow

$$\Delta\omega = \begin{bmatrix} 0 & 1 \end{bmatrix} \begin{bmatrix} \Delta\delta \\ \Delta\omega \end{bmatrix} \quad (22)$$

Consequently, the eigenvalues and damping ratios of the system with and without stabilizer are given in Table 1. The designed specifications are met.

Table 1 System eigenvalues

Without stabilizer	With sliding mode stabilizer
$0.0735 \pm 1.91i$ ($\zeta = -0.038$)	$-0.941 \pm 1.34i$ ($\zeta = 0.57$)

In order to examine the performance of the dynamic compensation controller, the system shown in Fig.1 was disturbed by the pseudo-typical wind gust of the form

$$V_{gust} = G(1 - \cos\frac{2\pi t}{T}) \quad (23)$$

where $0 < t < T$, $T = 4$ and $G = 7.5\%$. Fig. 4 shows the wind gust which is simulated to the system under three various conditions. Besides, the effect of wind turbine system the designed sliding mode stabilizer is compared to that of the Variable Structure Stabilizer (VSS) [5].

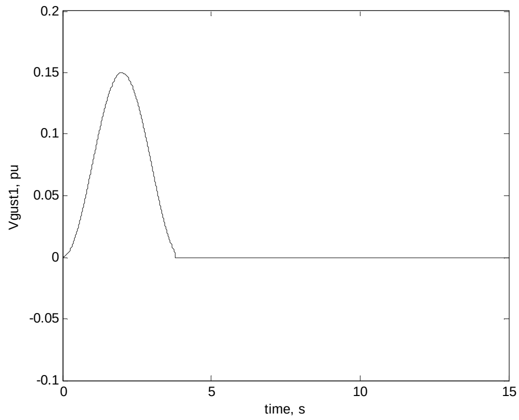

Fig.4 Wind gust disturbance.

4.1 At the normal operating point

The power system operates at $P = 1.0$ pu, $Q = -0.3$ pu, the field time constant $T'_{do} = 1.942$ sec,and the transmission line $X_e = 0.3$ pu. The speed deviation responses of the system with and without controllers are illustrated in Fig.5. The response without controller is unstable whereas the response of the system with both VSS and sliding mode stabilizer is well damped.

4.2 With the changes in reactive power and field time constant

In this situation, reactive power and field time constant are reduced to be $Q = -0.15$ pu, $T'_{do} = 1$ sec, respectively. The responses of generator speed deviation are shown in Fig.6. It can be seen that the amplitude of the response with the sliding mode stabilizer is lower than that with the VSS. The VSS is sensitive to these parameter variations.

4.3 At light loading with a decreased field time constant

Under this condition, real power, reactive power, and field time constant are set at $P = 0.1$ pu $Q = -0.6$ pu

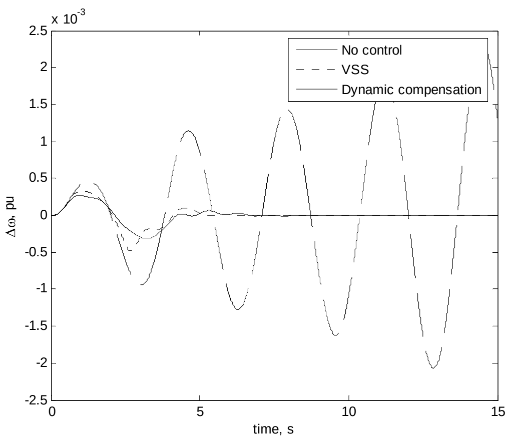

Fig.5 Speed deviation response at operating point.

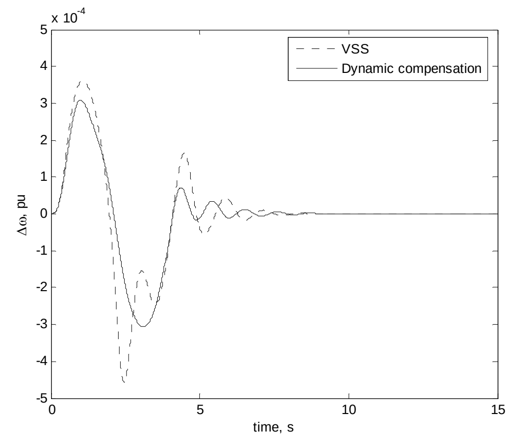

Fig.6 Speed deviation response with decreases in reactive power and field time constant.

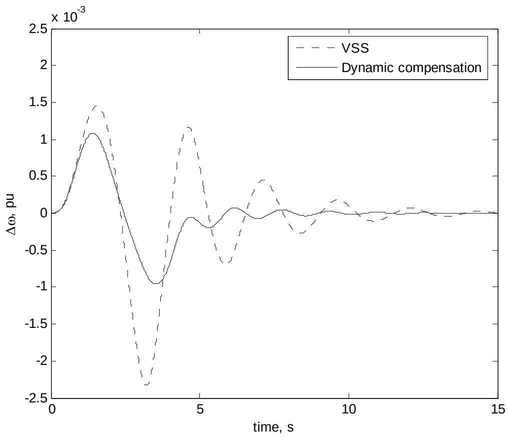

Fig.7 Speed deviation response at light loading.

and $T'_{do} = 1.2$ sec, respectively. As demonstrated in Fig.7, it is clear that the speed deviation response by VSS controller is poorly damped, the oscillation is much severe than that of the sliding mode stabilizer. The VSS controller is very sensitive to system parameter variations. On the other hand, the proposed sliding mode stabilizer is very robust to this operating condition. It is able to damp speed oscillation significantly and rapidly. These simulation results confirm that the robustness of the proposed sliding mode stabilizer is much superior to that of the VSS [5].

5 Conclusion

In this paper, the dynamic compensation observer technique has been applied to design a sliding mode stabilizer for the wind turbine system. The main advantage of this method is that only output information is required as a feedback signal of the controller. Moreover, this technique provides a systematic design to achieve the design specification, robustness against system parameter variations, and an enhancement of system dynamic performance. Simulation results against the variety of loading conditions and system parameters under the wind gust disturbance indicate that the sliding mode stabilizer provides higher effectiveness and superior robustness than the variable structure stabilizer.

In the future work, the effectiveness of the proposed stabilizer will be investigated under more severe, continuous and random wind gusts, various system parameter variations etc.

References:

[1] T. Ackermann, *Wind Power in Power Systems*, John Wiley & Sons, Ltd, 2005.

[2] E.N. Hinrichsen and P.J. Nolan, Dynamics and stability of wind turbine generators, *IEEE Trans. Power Apparatus System*, Vol. 101, 1982, pp. 2640-2648.

[3] C.C. Johnson and R.T. Smith, Dynamics of wind generators on electric utility networks, *IEEE Trans. on Aerospace Electron and System*, Vol.12, 1976, pp.483-492.

[4] T.S. Bhatti, A. A. F. Al-Ademi and N. K. Bansal, Load frequency control of isolated wind diesel hybrid power systems, *Energy Conversion and Management,* Vol. 39, No.9, 1997, pp.829-837.

[5] Y.L. Abdel-Magid, Z.M. Al-Hamouz, J.M. Bakhashwain, A variable-structure stabilizer for wind turbine generators, *Electric Power Systems Research*, Vol. 33, 1995, pp. 41-48.

[6] U. Itkis, *Control systems of variable structure*, Wiley, New York, 1976.

[7] C. Edward and S.K. Spurgeon, *Sliding Mode Control: Theory and Application*, Taylor and Francis, 1798-1998, ch.5.

[8] K. Ogata, *Modern Control Engineering*, Prentice -Hall, Inc., 4^{th} edition, 2002, ch.12.