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PERFORMANCE COMPARISON OF TWO TURBINE BLADE PITCH CONTROLLER DESIGN METHODS BASED ON EQUILIBRIUM AND FROZEN WAKE ASSUMPTIONS

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ABSTRACT

This paper focuses on the design methods of two different gain-scheduled collective blade pitch controllers for a 5 MW horizontal axis wind turbine using equilibrium and frozen wake assumptions. The design and performance tests of the controllers are carried out using the MS Bladed Wind Turbine Simulation Model.

Keywords: Turbine Blade Pitch Controller Designs and Simulations, MS Bladed Wind Turbine Simulation Model, Equilibrium and Frozen Wake Assumptions

INTRODUCTION

Control systems of modern Horizontal Axis Wind Turbines, or HAWTs, consist of different layers; the highest, middle and the lowest level controls. The highest level control, also known as supervisory control, determines when to start or stop turbines depending on wind speed. Turbines start producing electricity at a wind speed that is referred to as cut-in wind speed. At another wind speed, referred to as cut-out wind speed, turbines are stopped in order to prevent them from damages. Middle-level control is about the turbine's own control and is referred to as operational control. This control level includes the generator torque, blade pitch and yaw controllers. The generator torque control is used to obtain the optimum turbine efficiency. The blade pitch control is utilized to generate the rated electrical power. Yaw control is employed to direct the nacelle into the wind. The lowest level control, however, includes an internal generator control, actuator control etc.[Johnson, Pao, Balas, & Fingersh, 2006; Sahin & Yavrucuk, 2017b].

Modern HAWTs with the above control properties can operate at variable rotor speeds and blade pitch angles. They have three basic operational regions: Region 1, Region 2 and Region 3. Below the cut-in wind speed is Region 1, where turbines do not generate any electrical energy because wind speeds are not sufficient to produce electrical power even for turbines' own systems. Electrical power generation starts at the cut-in wind speed and ends up at the cut-out wind speed. Between these two wind speeds, there are two operational regions, Region 2 and 3. The region

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between the cut-in and rated wind speeds is Region 2, whereas the region between the rated and cut-out wind speeds is Region 3[Johnson et al., 2006; Merabet, Thongam, & Gu, 2011]. Region 2 and 3 are also known as partial and full load or below and above rated regions, respectively. Some articles such as the reference of [Oudah, Mohd, & Hameed, 2014] considers the region above the cut–out wind speed as an extra region, Region 4. In this region, turbines are normally not allowed to operate due to the extreme turbine loadings. However, most modern turbines are operated even in Region 4 with one of the offline-shaped strategies such as ramp shaping, stepwise shaping etc.[Fischer & Shan, 2013]. In addition, between Region 1 and 2, as well as Region 2 and 3, there are transition regions, which are narrow and are known as Region 1.5 and 2.5, respectively. The torque controls for these regions are referred to as Region 1.5 and 2.5 controllers, respectively[Sahin, 2018].

The main objective in Region 2 is to maximize the energy capture of turbines because wind speeds in this region are low to produce the rated power. The maximum energy is generated by varying the rotor speed and keeping blade pitch angle at the optimum value against the changing wind speed. This is realized by a generator torque controller that allows turbines to operate at the optimum tip speed ratio, TSR and therefore with the maximum power coefficient, C_{pmax}. The main aim in Region 3, on the other hand, is to limit the power output of turbines to their rated values. Power limitation is generally achieved by keeping the generator torque constant at its rated value and adjusting the blade pitch angles to regulate the rotor speed. This is obtained by a blade pitch controller[Johnson et al., 2006; Merabet et al., 2011; Stol & Fingersh, 2004].

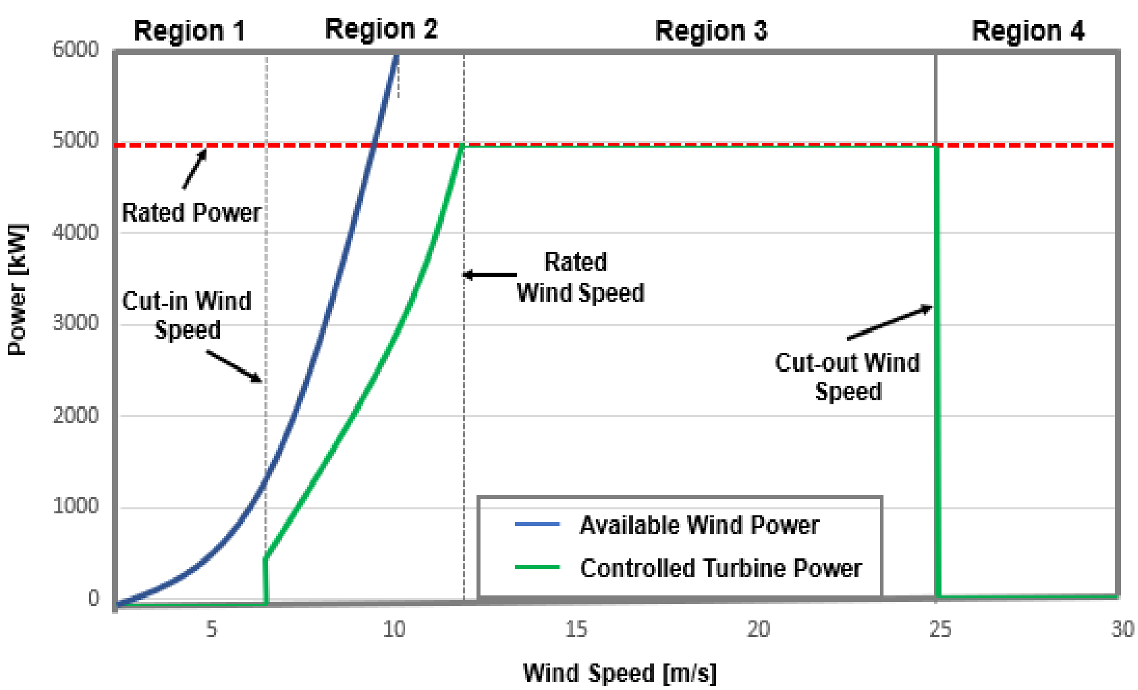

Figure 1: Illustration of Wind Turbine Operational Regions [Aho et al., 2012]

Figure 1 depicts all these regions and their boundaries for a 5 MW turbine. The dashed red line shows the turbine rated power. The blue curve indicates the available power in the wind. The green, however, represents the controlled power curve of the 5 MW turbine. As seen in the figure, not all the power available in the wind is extracted into the electrical power due to the Betz limit and losses in turbine mechanical and electrical components.

In this study, two different gain-scheduled Proportional and Integral (PI) strategy based collective blade pitch controller design methods are investigated for the NREL 5 MW turbine using equilibrium and frozen wake assumptions. The design and simulations are carried out using the MS (Mustafa SAHIN) Bladed Wind Turbine Simulation Model or MS Bladed Model. The study is organized as follows: the next section defines the MS Bladed Model briefly. Afterwards, the subject of blade pitch control is discussed. Methodology section focuses on the details of the turbine model linearization, blade pitch controller designs and simulations. Finally, the conclusions part is drawn.

MS BLADED WIND TURBINE SIMULATION MODEL

The MS Bladed Model, which is based on Blade Element Momentum (BEM) Theory, is developed for HAWT simulations. The aerodynamic calculations in the model are similar to those of Prop Code[Wilson & Lissaman, 1974], Wt_Perf[Platt & Buhl, 2012] and Aerodyn[Moriarty & Hansen, 2005]. By the MS Bladed Model, performance predictions of HAWT rotors may be examined. Turbine controller designs, new controller algorithm developments and their simulations may be realized. Turbine behaviour under normal/extreme turbulent winds may be investigated in time. Since the model includes various coordinate systems and aerodynamic corrections, it has the capabilities of nacelle yawing and individual or collective blade pitching. Further, it allows to define pre-cone and nacelle tilt angles. More information about the model may be found in the references of [Sahin, 2018; Sahin & Yavrucuk, 2017a, 2017c]. The main parts of the turbine system model are a turbine rotor, a gearbox and a simple variable torque electrical generator. It considers all the turbine parts such as blade, rotor shaft etc. as rigid structures and is constructed on the following relations.

$$J_t\dot{\Omega} = \tau_{aero} - \tau_{gen} \tag{1}$$

$$J_t = J_r + N_{gear}^2 J_{gen} \tag{2}$$

where, τ_{aero} is the rotor aerodynamic torque, τ_{gen}, generator electromagnetic torque, J_t, total inertia of the turbine system, Ω, rotor speed, N_{gear}, gearbox ratio and J_{gen}, generator inertia.

BLADE PITCH CONTROL OF TURBINES

Collective blade pitch control allows all the turbine blades to move together with the same amount of pitch angles to regulate turbine power output. For variable pitch turbines, it is achieved by pitch to stall or pitch to feather method, which respectively puts the turbine blades into stall or feather condition. In the first method, blades are operated under high angle of attacks (AOAs) that correspond to low aerodynamic lift and high drag coefficients. In the second method, however, blades are operated with low AOAs and therefore with low lift and drag coefficients. Although both methods utilize different aerodynamic phenomena, they share almost the same control block diagram (Figure 2). They both use the rotor speed or generally the generator speed as feedback to calculate the demanded pitch angle for pitch actuators. The actuators are limited by constructional constraints such as rate limits, min and max etc. For modern large-scale turbines, pitch to feather control is commonly used. Thus, this study focuses on this method with two different designs based on equilibrium and frozen wake assumptions. In the equilibrium wake assumption, there is nothing changed in the model, whereas in the frozen wake assumption, some variable throughout the blade spans are kept fixed during linearization.

In this study, the rotor speed is fed back to the blade pitch controller, rather than that of the generator. Therefore, the pitch command is determined based on the rotor speed error between the rated and measured rotor speeds. The pitch angle range is limited between -0.875 and 90 degree-pitch angles. A rate limiter of 8 deg/s is also added to the pitch control system.

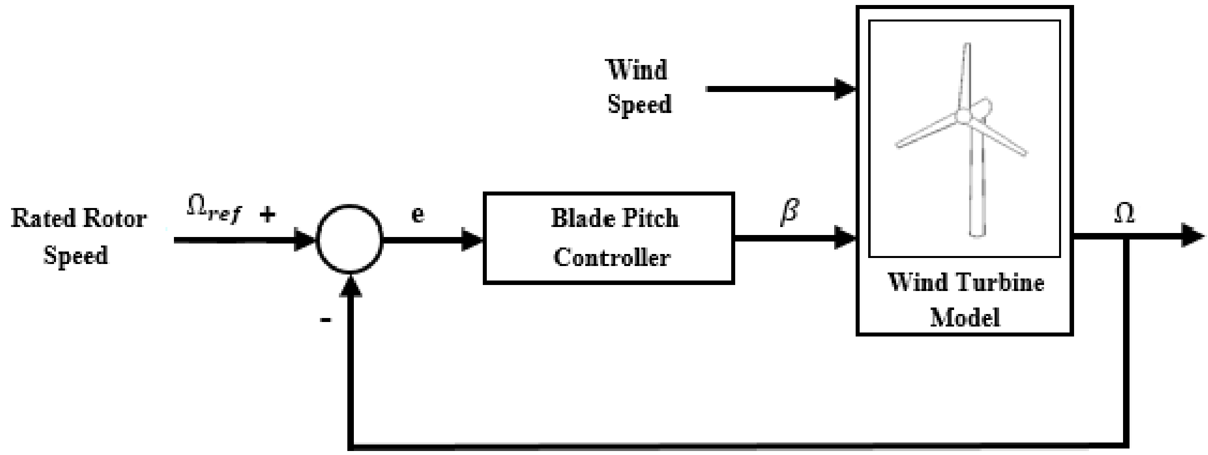

Figure 2: Block Diagram for Blade Pitch Controller [Sahin, 2018]

METHODOLOGY

In this part, two gain-scheduled collective blade pitch controllers are designed to regulate the rotor speed of the NREL 5 MW turbine and therefore to produce the rated electrical power. Proportional and Integral (PI) strategy based control is adopted to achieve the purpose. In the above rated region, a gain scheduling design is used since one linear PI-based controller designed for one equilibrium point demonstrates poor performance at other equilibrium points. Followings are the details about the designs and simulations.

Wind Turbine System Linearization

The MS Bladed Model consider the dynamic turbine system as in equation (1). The system is a nonlinear system due to both aerodynamic and generator torques. However, for the above rated region, or Region 3, the nonlinearity is caused by the aerodynamic torque only since generator torque is kept constant at its rated value. The system consists of only one state i.e rotor speed, Ω. It is the change of azimuth angle, Λ, of the turbine blades in time. The control inputs to the MS Bladed Model are only the pitch angles, whereas the freestream wind is a disturbance input to the model.

In order to design a linear controller, the turbine model is to be linearized around an equilibrium point. A perturbation technique is a commonly-used approach for linearization. For the above rated region, an equilibrium point for a turbine may be defined as the point at which wind speed and blade pitch angle, the aerodynamic torque reaches at the rated generator torque at the rated rotor speed. Therefore, the rotor aerodynamic torque is a continuous function and depends on three variables; rotor speed, Ω, blade pitch angle, β, and lastly wind speed, *U*. Thus, following the reference of [A. Wright, 2004], the aerodynamic torque may be expanded by a Taylor series as,

$$\tau_{aero}(U,\Omega,\beta) = \tau_{aero}(U_e,\Omega_e,\beta_e) + \frac{\partial \tau_{aero}}{\partial U}(U-U_e) + \frac{\partial \tau_{aero}}{\partial \Omega}(\Omega-\Omega_e) + \frac{\partial \tau_{aero}}{\partial \beta}(\beta-\beta_e) + HOTs \qquad (3)$$

where U_e, Ω_e, β_e are respectively the values of wind speed, shaft speed, and blade pitch angle at an equilibrium point. $U - U_e = \Delta U,\ \Omega - \Omega_e = \Delta\Omega$ and $\beta - \beta_e = \Delta\beta$ are the perturbations or small deviations from the equilibrium point. For linear controller designs, the first order Taylor series expansion is good enough for the approximation of nonlinear system. Therefore, Higher Order Terms (HOTs) in equation (3) are neglected. If the series expansion is applied to the whole turbine system in equation (1), considering the generator torque on the LSS of gearbox, τ_{gen}, the following is obtained.

$$J_t(\Omega \dot{-} \Omega_e) = \tau_{aero}(\Omega_e, \beta_e, U_e) + \gamma\Delta\Omega + +\eta\Delta\beta + \mu\Delta U - \tau_{gen} \quad (4)$$

For Region 3, the rated generator torque on LSS of the gearbox is determined as 4180074.35 Nm[J. Jonkman, Butterfield, Musial, & Scott, 2009]. In addition, there is no rotor acceleration at steady-state because the aerodynamic and generator torques cancel each other at equilibrium. Thus, for any turbine operation at steady-state,

$$J_t\dot{\Omega} = \gamma\Delta\Omega + \eta\Delta\beta + \mu\Delta U \quad (5)$$

where, γ, η, μ represents the partial derivatives of $\frac{\partial\tau_{aero}}{\partial\Omega}$, $\frac{\partial\tau_{aero}}{\partial\beta}$ and $\frac{\partial\tau_{aero}}{\partial U}$, respectively. Further,

$$\dot{\Omega} = \frac{\gamma}{J_t}\Delta\Omega + \frac{\eta}{J_t}\Delta\beta + \frac{\mu}{J_t}\Delta U \quad (6)$$

If the followings are defined

$$\frac{\gamma}{J_t} = A \quad (7)$$

$$\frac{\eta}{J_t} = B \quad (8)$$

$$\frac{\mu}{J_t} = B_d \quad (9)$$

open loop turbine system turns out to be as

$$\dot{\Omega} = A\Delta\Omega + B\Delta\beta + B_d\Delta U \quad (10)$$

where A is the system gain, B is the input gain, and finally B_d is the disturbance gain. They are the ratios of the partial derivative of aerodynamic torque with respect to rotor speed, blade pitch angle and wind speed to the total inertia of the turbine system, respectively. As the goal in this part is to design a PI-based pitch control system, following the reference of [A. D. Wright & Fingersh, 2008], the perturbation of pitch angle, $\Delta\beta$, is related directly to that of rotor speed, $\Delta\Omega$,

$$\Delta\beta(t) = K_p\Delta\Omega(t) + K_i\int \Delta\,\Omega(t)dt \quad (11)$$

where K_p and K_i represent the proportional and integral gains, respectively. Thus, the controller output is as follows, $\beta = \Delta\beta + \beta_e$. Putting the equation (11) into the equation (10) constructs the closed-loop turbine system with PI strategy as in equation (12).

$$\dot{\Omega} = A\Delta\Omega + B\left(K_p\Delta\Omega(t) + K_i\int \Delta\,\Omega(t)dt\right) + B_d\Delta U \quad (12)$$

Taking the Laplace Transform of equation (12) as follows,

$$s\Omega(s) = A\Delta\Omega(s) + B\left(K_p\Delta\Omega(s) + \frac{K_i}{s}\Delta\Omega(s)\right) + B_d\Delta U(s) \tag{13}$$

and then carrying out some algebraic manipulations, the closed loop transfer function between the rotor speed, $\Delta\Omega(s)$ and the wind, $\Delta U(s)$ is obtained as,

$$G_{CL}(s) = \frac{\Delta\Omega(s)}{\Delta U(s)} = \frac{B_d s}{s^2 + \left(-A - BK_p\right)s + (-BK_i)} \tag{14}$$

The denominator of the transfer function or the characteristic equation of the closed loop system gives the information about the system stability considering the controller gains, K_p and K_i. Therefore, by designing K_p and K_i, the desired performance is easily obtained from the system. In order to have a closed loop turbine system stable, both roots of the characteristic equation are to be at least negative. Thus, the terms in parenthesis of the characteristic equation are to be larger than zero.

$$-A - BK_p > 0 \tag{15}$$

$$-BK_i > 0 \tag{16}$$

However, in order to achieve the desired response from the system, proper gains should be selected according to the design requirements. For a second order system, design requirements are determined by the desired natural frequency, w_n and damping ratio, ζ. The following subpart deals with the selection of natural frequency, damping ratio and the calculation of these gains.

<u>Design and Performance of Collective Blade Pitch Controller</u>

Here, the first PI-based control method, which consider the equilibrium wake, is discussed. Initially, for a selected equilibrium point, the effect of damping ratio on the turbine rotor speed response is examined to determine a suitable damping ratio. However, the desired natural frequency is kept the same as in the literature[Hansen et al., 2005]. The damping ratio of 0.8 is determined to give the best performance result in terms of settling time at the selected equilibrium point. Later, the blade pitch controller giving the best performance is also tested at other equilibrium points. However, the test simulations have demonstrated poor performance results. Thus, a gain-scheduled PI-based controller is designed for the NREL 5 MW turbine to have almost the same performance at every equilibrium point in the above rated region. The followings are the details of this design process.

In the first method, in order to design a blade pitch controller, the values of partial derivatives ɣ and η are required to be first determined to calculate the values of *A* and *B* gains at the selected equilibrium point. These gains are explicitly seen in the characteristic equation in (14). Here, these gain values are determined using the MS Bladed Model. The nonlinear aerodynamic model is linearized around a desired equilibrium point. Model linearization is carried out using the central difference theorem considering the equilibrium wake assumption. When the characteristic equation in (14) is considered as a standard second order system in Laplace form as

$$s^2 + 2w_n\zeta s + w_n^2 \tag{17}$$

Then, equations (18) and (19) become the relations among the natural frequency, damping ratio, system and input gains as well as controller gains.

$$2w_n\zeta = -A - BK_p \tag{18}$$

$$w_n^2 = -BK_i \tag{19}$$

Therefore, K_p and K_i gains are obtained by equations (20) and (21), respectively if the desired natural frequency and damping ratio are already known.

$$K_p = \frac{-2w_n\zeta}{B} - \frac{A}{B} \tag{20}$$

$$K_i = \frac{-w_n^2}{B} \tag{21}$$

According to the explanations above, Table 1 is prepared here for the discussions of controller design and analysis thought this subchapter. In Table 1, EP represents the equilibrium point.

Table 1: Selected Equilibrium Points for Controller Design and Simulations

Equilibrium Points	Wind Speed (m/s), U_e	Rotor Speed (rpm), Ω_e	Pitch Angle (deg), β_e	Rotor Torque (Nm), τ_e
EP1	18	12.1	14.9525	4180074.35
EP2	16	12.1	10.5521	4180074.35
EP3	13	12.1	6.7206	4180074.35
EP4	11.5	12.1	2.2792	4180074.35
EP5	12.6607	12.1	5.9676	4180074.35
EP6	23	12.1	20.9964	4180074.35

EP1 is taken into account, first. One PI-based blade pitch controller is designed for this equilibrium point. During a controller design for a turbine, the reference of [Hansen et al., 2005] has suggested utilizing a natural frequency, w_n of 0.6 and a damping ratio, ζ of 0.6-0.7 to have a satisfactory response. However, by keeping the natural frequency as w_n of 0.6, Wright and Fingersh have selected a damping ratio of 1 for the NREL 600 KW CART turbine after some trials during the pitch controller design[A. D. Wright & Fingersh, 2008]. Therefore, these values, particularly the damping ratio, depends on turbines. Thus, a similar approach in the reference of [A. D. Wright & Fingersh, 2008] is adopted here to find the best damping ratio. Later on, the same damping ratio is kept being used for both controller designs. Through the linearization of MS Bladed Model, A, B and B_d gains at EP1 are obtained as follows.

$$A = -0.2401$$

$$B = -1.1672$$

$$B_d = 0.0275$$

Using the above system gain, A and input gain, B and natural frequency, w_n and damping ratio, ζ respectively as 0.6 and 0.7 in equations (20) and (21) results in a proportional gain, K_p of 0.5140 and an integral gain, K_i of 0.3084. By this designed controller, the rotor speed response of controlled turbine (dashed blue) to a step increasing wind input from 17 to 18 m/s at 50s of the simulation is shown in Figure 3. Rotor speed response overshoots slightly the steady-state at around 58s and settles down eventually at around 65s. The rotor speed response to a disturbance wind input looks quite satisfactory in terms of settling time because it is only a duration of approximately 15 seconds. However, there may be another damping ratio that gives a better rotor speed response than the one obtained. To decide an appropriate damping ratio, the value of damping ratio of the closed turbine system is decreased and increased. All these responses are also given in Figure 3.

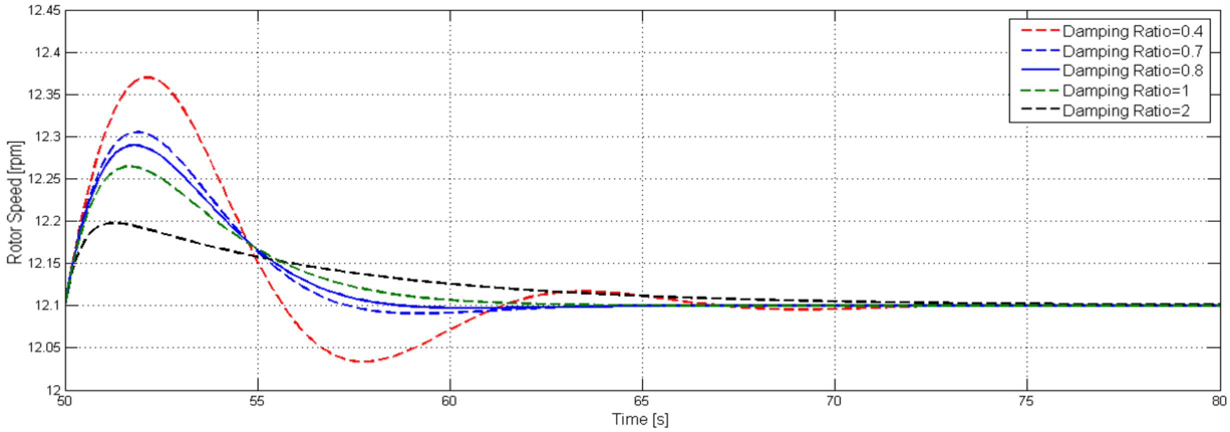

Figure 3: The Best Rotor Speed Response with the Damping Ratio of 0.8 at EP1

As seen from the figure, with the damping ratio of 0.4, the response is an undamped oscillation and takes a quite large time to settle down. Even in 15 seconds, it does not reach the steady-state and still oscillates. In terms of settling time, the same is also valid for the response with the damping ratio of 2. However, the damping ratios of 0.7 and 1 demonstrate closer settling times. The response with damping ratio of 1 gives a closer output to the steady-state around 61s. On the other hand, the response with damping ratio of 0.7 gives a closer value to the steady-state around 58s. It overshoots slightly and settles down approximately within 15 seconds. Thus, the damping ratio of 0.7 seems to be a better damping ratio. However, it also seems that selecting a damping ratio of 0.8 provides the best performance in terms of settling time.

Table 2 shows the corresponding proportional and integral gains of the closed-loop system as well as the system roots when the above-mentioned damping ratios and natural frequency are used for controller design. In Table 2, when the damping ratio is less than 1, the closed-loop system has complex conjugate roots, turning into an undamped system and therefore demonstrates an oscillatory response. This response typically occurs since the closed-loop system is turned into an underdamped system. However, when the damping ratio is increased to 1, the system operates with the repeating roots.

Table 2: Estimation of the Best Damping Ratio

Damping Ratio ζ	**Natural Frequency** w_n	**Proportional Gain** K_p	**Integral Gain** K_i	**System Root 1**	**System Root 2**
0.4	0.6	0.2055	0.3084	-0.24-0.5500i	-0.24+0.5500i
0.7	0.6	0.5140	0.3084	-0.42-0.4285i	-0.42+0.4285i
0.8	**0.6**	**0.6168**	**0.3084**	**-0.48-0.3600i**	**-0.48+0.3600i**
1	0.6	0.8224	0.3084	-0.6	-0.6
2	0.6	1.8505	0.3084	-2.2392	-0.1608

Thus, the system becomes a critically damped system. Increasing the damping ratio further into 2, the system has two different negative real roots which make the system overdamped. Therefore, the eventual response of the system is determined by the smaller root in magnitude.

For all the cases in Table 2, the turbine system with the controller is stable due to negative real parts in the system roots. Therefore, it eventually reaches at the steady-state.

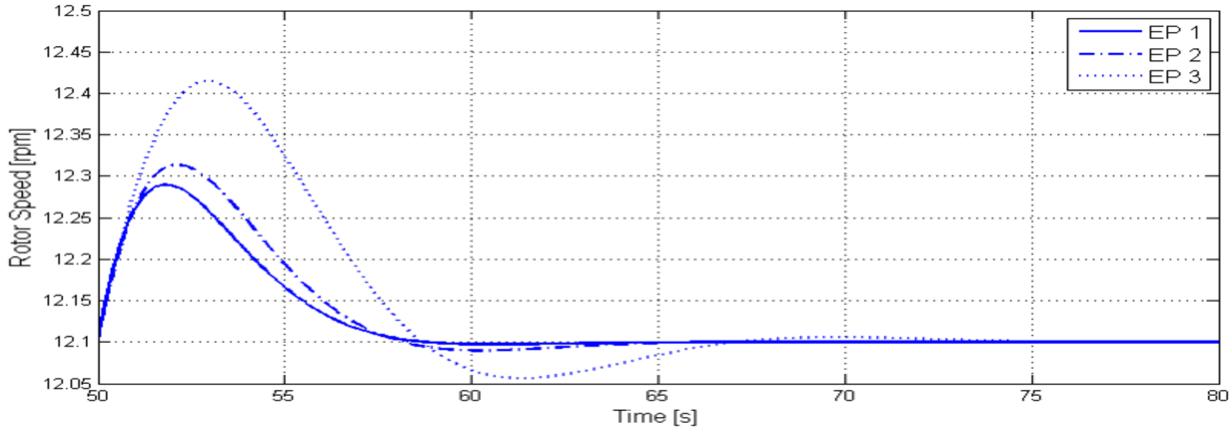

Figure 4: Controller Performance Deterioration at other Equilibrium Points

Until now, for EP1, a PI-based pitch controller with different gains are considered. It is seen in Figure 3 that the best performance is obtained with the damping ratio of 0.8. Therefore, from now on, the damping ratio of 0.8 is kept being used as the desired damping ratio, while the same natural frequency, 0.6, is kept being utilized as before. However, when the controller with K_p of 0.6168 and K_i of 0.3084 is tested at other equilibrium points, EP2 and EP3 (Table 1), the controller performance deteriorates, particularly at EP3, a very close equilibrium point to the rated equilibrium point. Figure 4 shows the deterioration which occurs due to the variation in control input gain, B with the changes in pitch angle and wind speed[A. D. Wright & Fingersh, 2008]. The value of input gain, B is directly related to η, the partial derivative of aerodynamic torque, τ_{aero} with respect to the blade pitch angle, β at the rated torque and rotor speed.

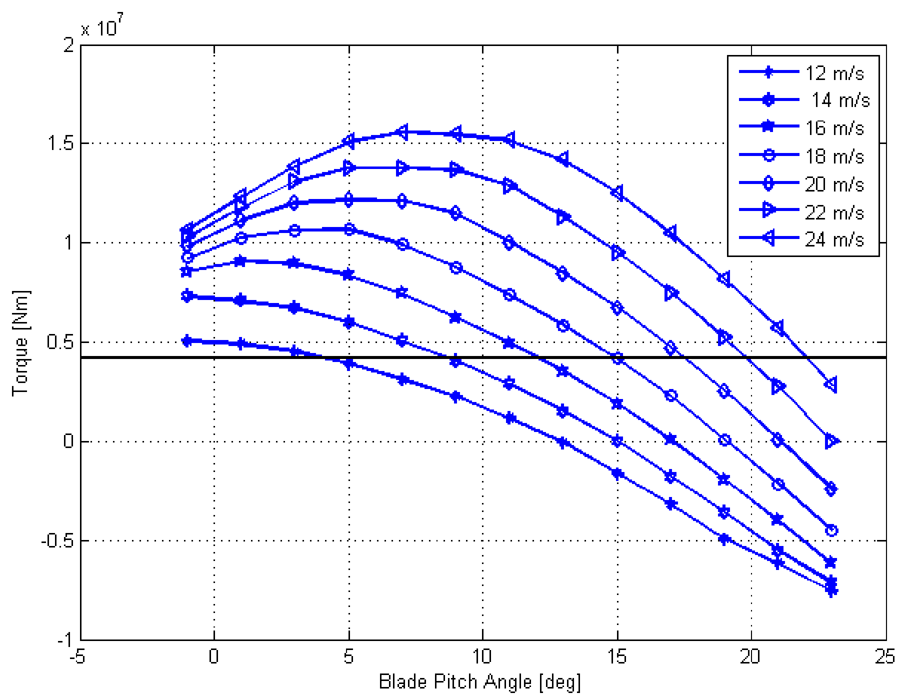

Figure 5: Aerodynamic Torque versus Pitch Angle at Various Wind Speeds

Figure 5 is obtained using the MS Bladed Model and shows the aerodynamic torque versus pitch angle at different wind speeds. The solid black line is the rated torque on the LSS of the gearbox. The crossing points of this line with the blue torque curves are the equilibrium points for the open loop turbine system. As seen in the figure, the value of input gain, B, is directly related to the partial derivative of $\frac{\partial \tau_{aero}}{\partial \beta}$, whose value in magnitude increases when the pitch angle increases and vice versa. Therefore, the input gain, B differs at every equilibrium point. For this reason, a controller designed at any equilibrium point does not give the same performance at other equilibrium points. Thus, in order to get similar performance at every equilibrium point, the controller gains are required to be scheduled relying on the blade pitch angle. As seen in Figure 5, the pitch angles correspond to different wind speeds at different equilibrium points. Thus, a gain-scheduled PI-based blade pitch controller is designed and implemented, here.

In the literature, the gain scheduling of PI methodology is realized by two similar means. These are based on the partial derivative of aerodynamic torque with respect to pitch angle, $\frac{\partial \tau_{aero}}{\partial \beta}$ [A. D. Wright & Fingersh, 2008] or rotor aerodynamic power with respect to pitch angle, $\frac{\partial P}{\partial \beta}$, known as pitch sensitivity[J. Jonkman et al., 2009]. For a gain scheduling purpose, both methods use a term referred to as gain correction factor, $GK(\beta)$. By simply multiplying the calculated controller gains by $GK(\beta)$, a superior performance may be achieved from the controller at any equilibrium point throughout the above rated region. The equation for $GK(\beta)$ is defined in the references of [J. Jonkman et al., 2009; A. D. Wright & Fingersh, 2008] as follows.

$$GK(\beta) = \frac{1}{\left(1 + \frac{\beta}{\beta_K}\right)} \tag{22}$$

where β is the blade pitch angle required for the turbine to produce the rated torque at any wind speed when the turbine operates at the rated rotor speed. The estimation of β_K is similar and is probably derived from the same idea in both approaches. According to Wright and Fingers, β_K is the blade pitch angle where the input gain, *B*, calculated at an equilibrium point close to the border of Region 2 into 3 has doubled in its value at another equilibrium point further in Region 3[A. D. Wright & Fingersh, 2008]. Their application of gain scheduling employs the FAST linearization considering the equilibrium wake assumption. However, according to the approach used by Jonkman et al.(J. Jonkman et al., 2009), β_K is defined as the blade pitch angle at which the pitch sensitivity, $\frac{\partial P}{\partial \beta}$ at zero pitch angle has doubled in its value further in Region 3. The partial derivative $\frac{\partial P}{\partial \beta}$ at zero pitch angle is obtained by a curve fitting approach to the pitch sensitivity values at various pitch angles. The best fit line is used to calculate the pitch sensitivity at zero blade pitch angle and is later utilized for obtaining the β_K value. The pitch sensitivity values are estimated considering the frozen wake assumption rather than the equilibrium wake assumption during the FAST Model linearization[J. Jonkman et al., 2009].

$$\frac{\partial P}{\partial \beta}(\beta = \beta_K) = 2\frac{\partial P}{\partial \beta}(\beta = 0) \tag{23}$$

Therefore, the blade pitch controller design is started here with the approach employed by Wright and Fingersh, initially. According to them, an operating point close to the entry of Region 2 into Region 3 is first selected[A. D. Wright & Fingersh, 2008]. This corresponds to the EP4 in Table 1. Later, β_K is calculated according to the approach they utilized. By the linearization of MS Bladed

Model, the following system gain A, input gain, B and disturbance gain, B_d are obtained for the EP4.

$$A = -0.0554$$

$$B = -0.2658$$

$$B_d = 0.0227$$

When the desired damping ratio and natural frequency are used respectively as 0.8 and 0.6, the proportional and integral gains are calculated through equations (20) and (21), respectively as

$$K_p = 3.4033$$

$$K_i = 1.3544$$

EP5 is the equilibrium point in Region 3 at which the input gain, B has doubled in its value. This equilibrium point is estimated by means of model linearization. Here, at the EP5, β_K has a blade pitch angle value of 5.9676 degrees. Therefore, the gain correction factor, $GK(\beta)$, is obtained using this β_K value. The β value in GK formula in may be obtained by different ways. Here, the adopted method is the interpolation of pitch angles with respect to wind speeds. For the gain scheduling purpose, the above proportional, K_p and integral, K_i gains are multiplied by the gain correction factor, GK.

Figure 6 shows the performance of the gain-scheduled PI-based controller at other three equilibrium points, where step increasing wind inputs such as from 12 m/s to 13 m/s and 17 m/s to 18 m/s and lastly 22 m/s to 23 m/s are applied to the controlled MS Bladed Model.

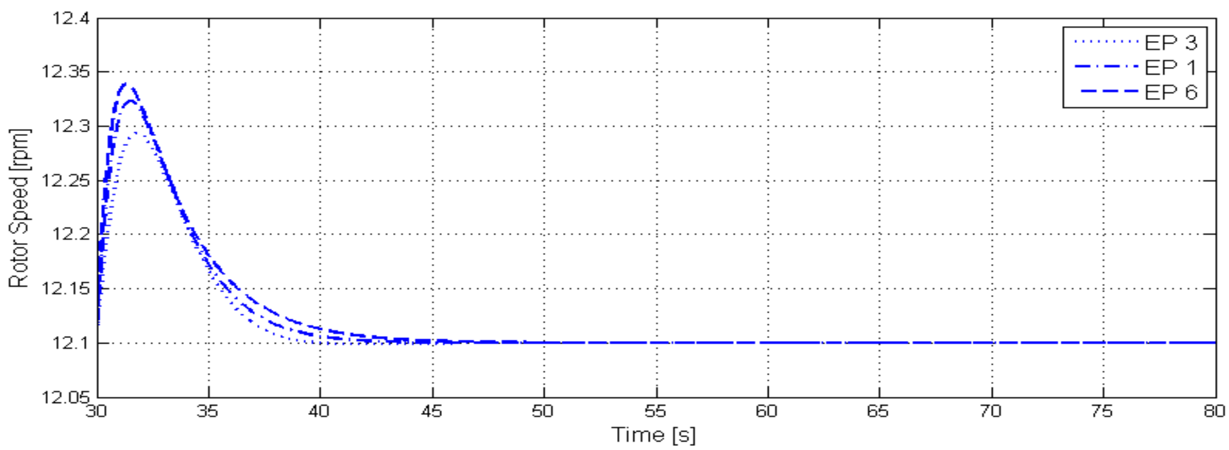

Figure 6: Gain Scheduled PI-based Pitch Controller, the Damping Ratio of 0.8

Which are, in fact, correspond to EP3, EP1 and a newly added equilibrium, EP6, respectively. As seen from the simulations, the gain-scheduled pitch controller demonstrates almost the same performance at three different equilibrium points. This is contrary to the previously demonstrated poor performance of one linear controller in Figure 4. The settling times of rotor speed responses in Figure 6 are around 22 seconds. They are quite satisfactory even though they have slightly different rise and decay rates.

Figure 7 shows the change of gain correction factor, GK, with respect to blade pitch angle, β, whereas Figure 8 shows the changes in proportional and integral gains based on GK with respect to blade pitch angle according to the approach of Wright and Fingersh[A. D. Wright & Fingersh, 2008].

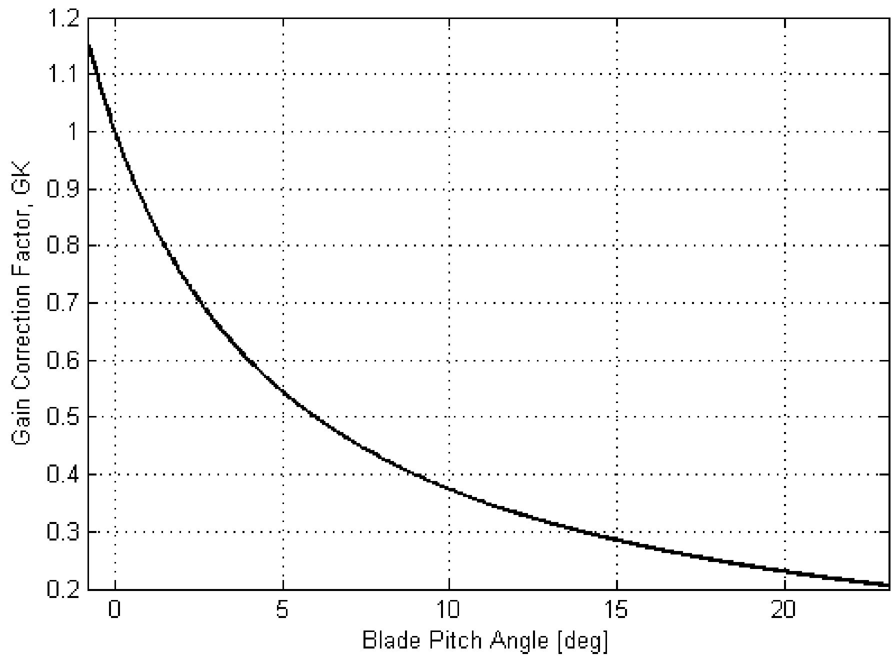

Figure 7: Gain Correction Factor versus Blade Pitch Angle

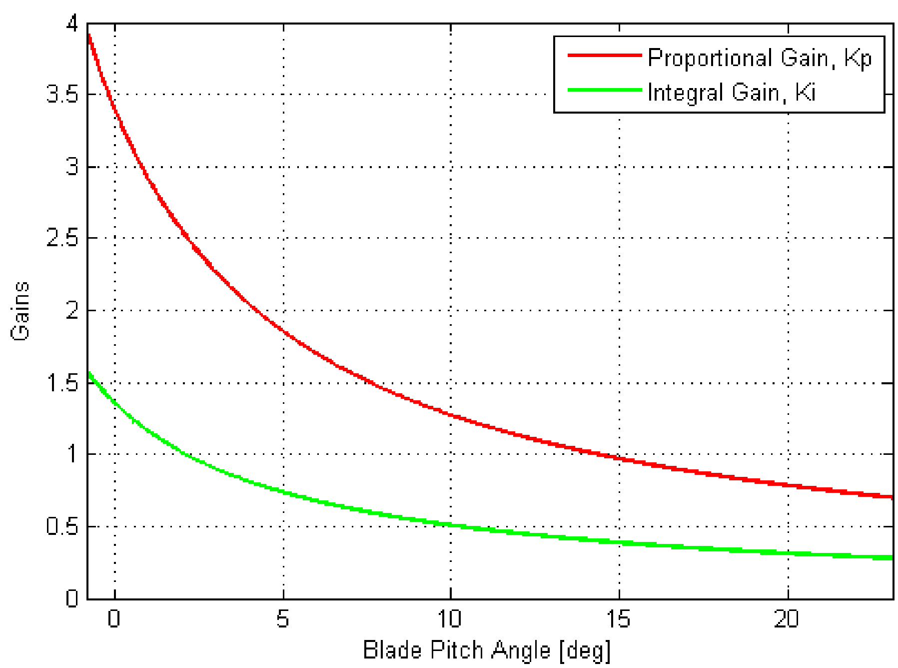

Figure 8: Proportional and Integral Gains versus Blade Pitch Angle

The second way of designing a gain scheduled blade pitch controller is to use the approach employed by Jonkman et al.[J. Jonkman et al., 2009]. According to them, the proportional and integral gains are found by equation (24) and (25). More information about obtaining these equations are available in the reference of [J. Jonkman et al., 2009]. However, these two equations are modified here in order not to include the gearbox ratio effect since the rotor speed is fed here to the controller, rather than the generator speed. During the blade pitch controller design or obtaining the gains, K_p and K_i with the approach of Jonkman et al., a frozen wake assumption is ought to be considered while obtaining the blade pitch sensitivity, $\frac{\partial P}{\partial \beta}$.

$$K_p = \frac{2J_t\Omega_e\zeta w_n}{-\frac{\partial P}{\partial \beta}(\beta = 0)}GK(\beta) \tag{24}$$

$$K_i = \frac{J_t\Omega_e w_n^2}{-\frac{\partial P}{\partial \beta}(\beta = 0)}GK(\beta) \tag{25}$$

This is due to the problem of a loss of control authority. Therefore, the frozen wake assumption is realized by fixing the elemental axial and tangential induced velocities, $-V_{bx_{i,j}}a_{i,j}$ and $V_{by_{i,j}}a'_{i,j}$ throughout each blade span during model linearization[J. M. Jonkman & Jonkman, 2016]. However, in the equilibrium wake assumption, there is nothing changed in the model during linearization. The same GK formula in (22) is utilized as in (24) and (25). The pitch angle, β is obtained as in the previous case. On the other hand, β_K is obtained as follows.

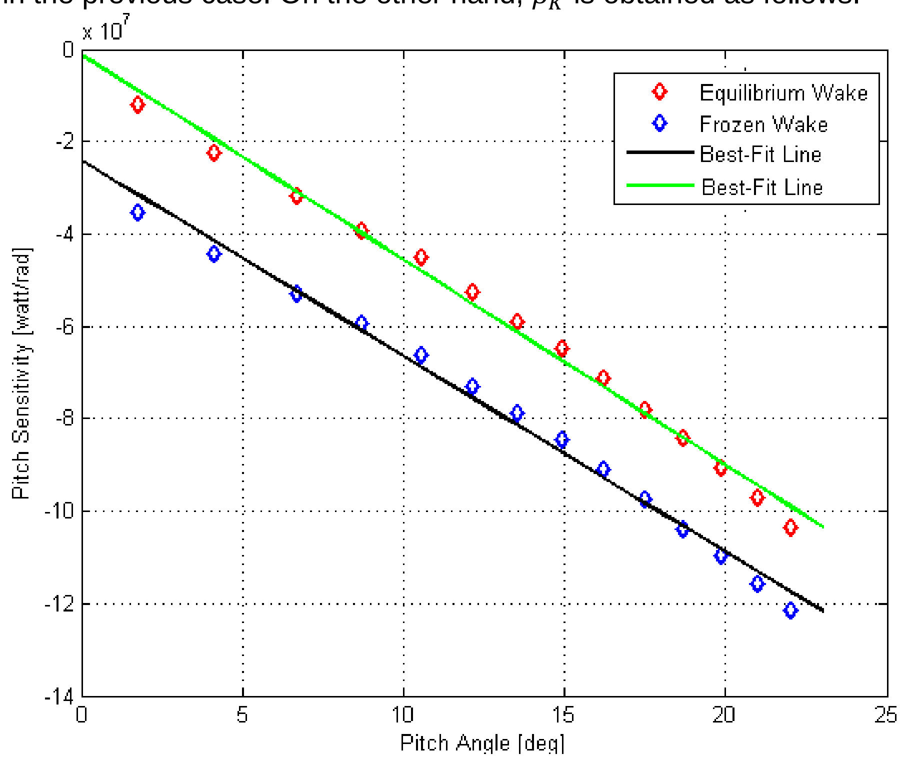

Figure 9: The Best-fit Line of Turbine Blade Pitch Sensitivity in Region 3

Figure 9 shows the pitch sensitivity versus blade pitch angle with the equilibrium wake and frozen wake assumptions. These are obtained using the MS Bladed Model and are given respectively by red and blue diamond symbols. Green and black lines are the best fit lines to these sensitivity values. Pitch sensitivity, $\frac{\partial P}{\partial \beta}$ at zero pitch angle is obtained as -24076000. Therefore, β_K, which corresponds to the double value of pitch sensitivity at zero pitch angle, is determined as 5.6775 degrees. Therefore, the gain-scheduled controller designed by the approach used by Jonkman et al.[J. Jonkman et al., 2009] with the same natural frequency, w_n of 0.6 and damping ratio, ζ of 0.8 gives the responses in Figure 10 at the same equilibrium points, EP3, 1 and 6.

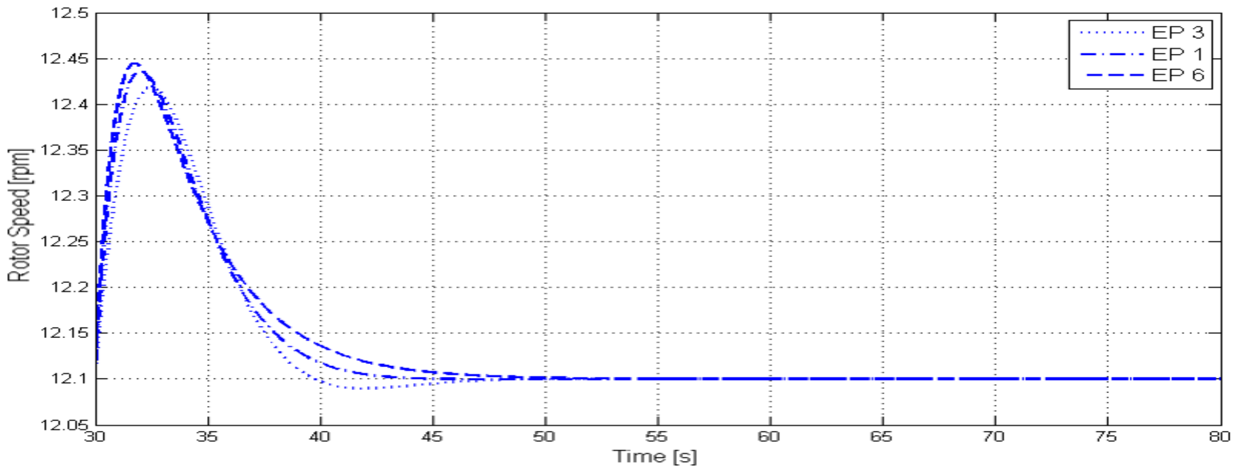

Figure 10: Gain Scheduled PI-based Pitch Controller, the Damping Ratio of 0.8

When the gains are scheduled according to the approach used by Jonkman et al.[J. Jonkman et al., 2009] using the gain correction factor, $GK(\beta)$ as in equations (24) and (25), all the responses settle similarly at around 22s after the same step increasing wind inputs are applied at 30s of the simulations. However, all the simulation results have larger peak responses than the ones in Figure 6, which are obtained by using the approach of Wright and Fingersh.

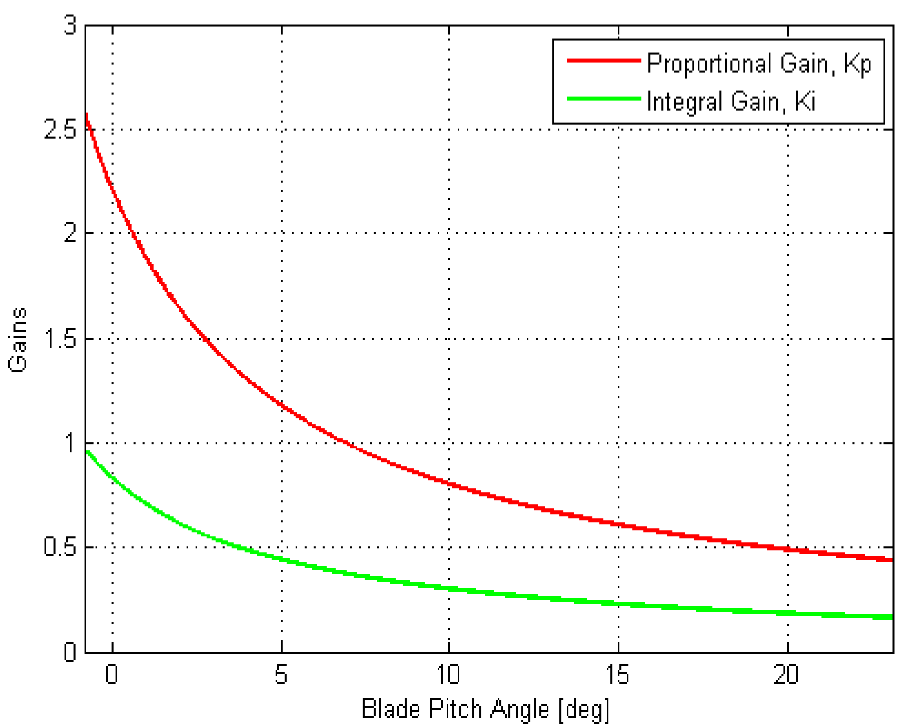

Figure 11: Proportional and Integral Gains versus Blade Pitch Angle

Figure 11 shows the proportional, K_p and integral, K_i gain values with respect to blade pitch angle. They are less than the ones obtained by the approach of Wright and Fingers at the same blade pitch angle, β.

CONCLUSIONS

This study has investigated the performance of two different PI strategy-based controller designs using two different methods which are based on equilibrium and frozen wake assumptions during turbine linearization. Two different methods are utilized to obtain the controller gains. In the first method, where equilibrium wake assumption is considered, the gain scheduled controller gains are obtained larger than those of the second method, which are obtained using pitch sensitivity and frozen wake assumptions. In terms of transient responses, the approach which uses pitch sensitivity has larger rotor speed peaks, slightly longer settling times at the same equilibrium points. Both methods give approximately the same performances. However, the first approach uses larger proportional and integral gains at the same operating or equilibrium points.

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