Proceedings of the
44th IEEE Conference on Decision and Control, and
the European Control Conference 2005
Seville, Spain, December 12-15, 2005

Towards Pitch-Scheduled Drive Train Damping in Variable-Speed, Horizontal-Axis Large Wind Turbines

Amit Dixit and Shashikanth Suryanarayanan

Abstract— **Given the prohibitive costs of replacement of damaged gearboxes, damping drive train oscillations is of immense significance in large wind turbine control design. The Drive Train Damper (DTD) is an important constituent of the power production control routine in variable-speed, pitch-regulated large wind turbines. The DTD is used to mitigate fatigue loading of drive train components. This paper motivates the need for scheduling the parameters of the damper based on blade pitch angle. It is shown that due to the strong coupling between blade edgewise motion and the drive train torsion, the fundamental frequency associated with the drive train's torsional motion can vary significantly across the range of blade pitch angles observed in practice. If left unaccommodated, this variation in fundamental frequency is shown to adversely affect the performance of the DTD.**

I. INTRODUCTION

Over the last couple of decades, development of environment-friendly alternate power production technologies has attracted considerable attention. Wind power has, thus far, proved to be the most promising "green power" option. Today, several wind farms produce power at a Cost-of-Energy (CoE) comparable to that of coal and natural gas based power plants [1].

For economic reasons, much of the work on wind turbine development has focused on large wind turbines. Typical large wind turbines of today (Fig. 1) are massive 3-bladed, horizontal-axis structures with enormous blade spans (70-100m in diameter), tall towers (60-100m in height) and power ratings in the 1-5 MW range. Most of these machines support variable-speed operation i.e., the rotor speed is not rigidly coupled to the grid frequency. Variable speed action enables realization of higher power capture efficiencies.

Amongst the technologies that enable realization of these machines, advanced control plays a pivotal role. Modern large wind turbines are endowed with sophisticated control systems which are organized to support several modes of operation such as start-up. shut-down, power production etc. In this paper, we look at a component of the control routine used in the power production mode of variable-speed large wind turbines known as the "Drive Train Damper (DTD)".

The drive-train in horizontal-axis large wind turbines consists of a low speed shaft coupled to the rotor, a step-up gear box and a high speed shaft that is coupled to the generator. Damping drive train oscillations becomes critical given the prohibitive costs of replacement of failed gearboxes. Drive train damping solutions proposed in the past have revolved around the idea of utilizing filtered versions of generator speed information to damp oscillations of the resonant mode of the drive train [2], [3]. In the absence of precise estimates of the fundamental frequency of interest, fixed parameter solutions tend to be conservative. In this paper, we propose a modeling approach to predict the frequency associated with that of the drive train's fundamental torsional mode. Specifically, we assert that the fundamental frequency in question varies with the blade pitch angle. This result gains significance since drive train oscillations/fatigue loading are most significant in high wind speed operating conditions during which the pitch control loop is active [4].

Fig. 1. Horizontal-Axis Large Wind Turbine (Courtesy: GE Energy)

The paper is organized as follows. Section II discusses the state-of-the-art of drive train damping solutions and motivates the need for an adaptive DTD structure. This section also includes a tutorial that illustrates the effect of coupling in mechanical systems. Section III details the effect of coupling between blade edge and drive train torsional oscillations that results in variation of the fundamental frequency associated with the drive train torsional mode. Section IV discusses simulation results performed on FAST -

A. Dixit and S. Suryanarayanan are with the Department of Mechanical Engineering, Indian Institute of Technology Bombay, Powai, Mumbai 400076, India. Email: : `adixit@me.iitb.ac.in`, `shashisn@iitb.ac.in`

a horizontal-axis wind turbine simulator developed at NREL[1] that corroborate analytical predictions of DTD performance. The paper concludes with a summary and a discussion on future problems of interest.

The primary contribution of the paper is the result that the fundamental frequency associated with the torsional mode of vibration of the drive train in horizontal-axis wind turbines varies with pitch angle. It is shown that since the coupling between the blade edge and drive train torsion modes of oscillation is pronounced, the fundamental frequency can vary significantly. These results are used to motivate the need to incorporate pitch scheduling into the DTD architecture.

II. PRELIMINARIES

A. Drive Train Damping

Large wind turbines of today are often equipped with four actuators that include three blade pitch motors and the generator (used as a torque control device) to control rotor speed, power produced and mechanical loads on the turbine structure. Fig. 2 shows the typical power production mode control architecture used in large wind turbine control. The torque control loop is active, primarily, in the lower wind speed region of operation during which the control objective is to maximize energy capture. On the other hand, the pitch control loop is active in higher wind speed regions of operation where the goal is to minimize fatigue damage of the turbine structure. See [5], [4] for more details.

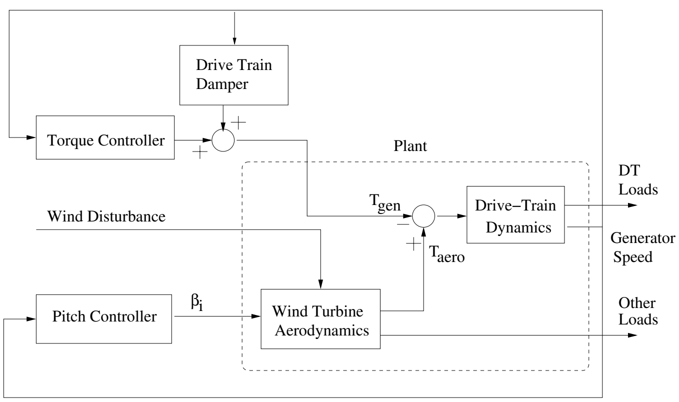

Fig. 2. Typical Control Architecture

In this paper, we concern ourselves with the drive train damper. As illustrated in Fig. 2, the drive train's torsional mode is excited by the rotor aerodynamic torque input (T_{aero}) produced by the interaction of the turbine's blades with a turbulent wind field incident upon the rotor plane. In the past, drive train dynamics have been modeled [2] using a mass-spring model as described below.

$$J\ddot{\theta}(t) + K\theta(t) = T_{aero} - T_{gen} \tag{1}$$

where, J and K are lumped parameters representing the equivalent rotational inertia and torsional stiffness of the drive train. The generator torque demand is denoted as T_{gen}. Natural damping is often neglected in the model structure since the drive train components are largely made out of steel with no explicit interconnecting damping elements.

Drive train damping solutions revolve around utilizing the generator as a damping device. The essential idea is to introduce damping action by making a component of T_{gen} proportional to a filtered version of generator speed ω. Bossanyi [3] has proposed a DTD structure of the form:

$$B(s) = \frac{Ks(1+\tau s)}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{2}$$

where ω_n corresponds to the frequency of the resonant mode associated with the torsional oscillations of the drive train. Such a structure is motivated by the following requirements:

1) Large gain close to the resonant frequency ω_n to provide adequate damping.
2) Small gains at low frequencies so that unwarranted power fluctuations are not introduced.
3) DTD component of torque demand should have a phase lag of 90^o to that of the forcing input. It turns out that for the dynamics considered in Eqn. 1, the filter should add close to zero phase at the resonant frequency ω_n.

Fig. 3 shows the Bode plot of a candidate filter that uses the above structure.

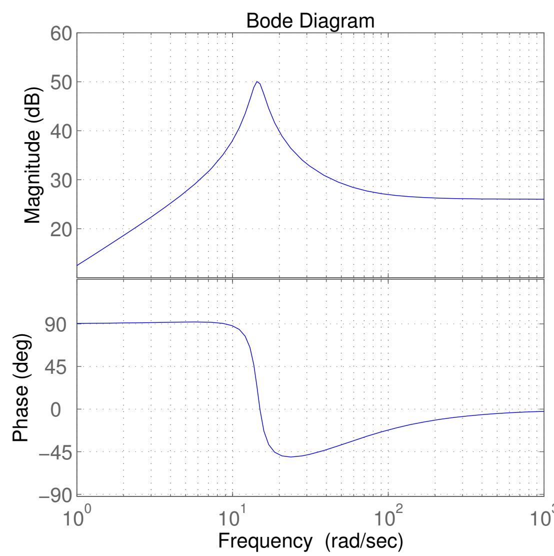

Fig. 3. Bode Plot of Candidate Filter

If the resonant frequency ω_n is precisely known, the filter parameter ζ may be chosen small so that the requirements above are met. However, since an estimate of the resonant frequency has been found to be difficult to obtain in practice, the damper parameter ζ is chosen fairly large to realize robust performance (damper peak has larger spread for larger values of ζ).

In this paper, we provide a methodology to predict changes in ω_n with changes in blade pitch angle. The benefits of such a strategy gains significance since drive train oscillations/fatigue loading are most significant in high wind speed

[1]National Renewable Energy Labs, Boulder, USA

operating conditions during which the pitch control loop is active.

B. Coupling in Mechanical Systems

In this subsection, we discuss the effects of "coupling" between elements of an interconnected mechanical system and its effect on system behavior. Consider an n degree-of-freedom system whose dynamics are described by the following coupled set of differential equations.

$$[M]\ddot{x(t)} + [C]\dot{x(t)} + [K]x(t) \quad = \quad \{f(t)\} \tag{3}$$

where $x(t)$, $\dot{x}(t)$ and $\ddot{x}(t)$ respectively denote $n \times 1$ displacement, velocity and acceleration vectors, while $[M]$, $[K]$ and $[C]$ denote $n \times n$ mass, damping and stiffness matrices respectively. The forcing function is denoted by $\{f(t)\}$. We will refer to the off-diagonal elements in the mass, damping and stiffness matrices respectively as the coupled mass, coupled damping and coupled stiffness terms.

An important set of parameters which governs the behavior of the system described in Eqn. 3 is the set of fundamental (natural) frequencies associated with the system. With the aid of a pedagogical example, we reiterate the well-known result that the natural frequencies of a "coupled" system can be significantly different from those of its constituent "uncoupled" systems. Consider the dynamics of a 2 degree-of-freedom system represented in Fig. 4. The pendulum system consists of a mass m attached to one end of a massless link of length L. The other end of the link is hinged to the mass M. For small displacements, the dynamics may be approximated as:

$$\begin{bmatrix} m_1 & m_c \\ m_c & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}(t) \\ \ddot{\theta}(t) \end{bmatrix} + \begin{bmatrix} k_1 & 0 \\ 0 & k_2 \end{bmatrix} \begin{bmatrix} x(t) \\ \theta(t) \end{bmatrix} = \begin{bmatrix} F(t) \\ \tau(t) \end{bmatrix}$$

where $m_1 = M + m$, $m_2 = mL^2$,$m_c = mL$, $k_1 = k$ and $k_2 = mgL$. The natural frequencies of this system are given by

$$\omega_{1,2} = \frac{m_1k_2 + m_2k_1 \pm \sqrt{(m_1k_2 - m_2k_1)^2 + 4m_c^2k_1k_2}}{2(m_1m_2 - m_c^2)}$$

which are different from the "uncoupled" natural frequencies $\omega_{uc1} = \sqrt{\frac{k1}{m1}}$ and $\omega_{uc2} = \sqrt{\frac{k2}{m2}}$

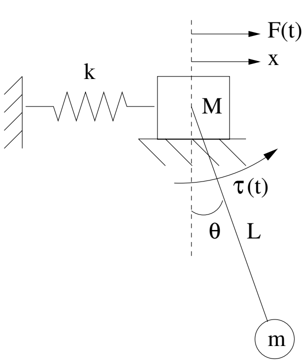

Fig. 4. A pendulum system

In general, the coupled mass, coupled stiffness or coupled damping terms are functions of the system geometry. In the pendulum system considered above, if the plane of rotation of the simple pendulum were at an angle β to the plane of motion of the mass M, it can be shown that the expression for the coupled mass changes to $m_c = mLcos\beta$, while all the other quantities remain unchanged. Changes in the coupled mass/stiffness terms result in changes in the natural frequencies associated with the the system. As shown in the following section, such a situation arises in the study of the dynamics of horizontal-axis wind turbines where the "coupled mass" of interest is a function of the blade pitch angle. For different pitch angles, the natural frequencies of the turbine system and in particular the frequency corresponding to drive train's torsional oscillations can vary significantly. As shown later in Section IV, such a change in the natural frequncy can have a significant impact on the performance of the DTD.

C. Fatigue Data Analysis

The general approach to fatigue life prediction of strucutral components subjected to a random stresses is to relate the life of the element to laboratory fatigue experiments of simple specimens subjected to constant amplitude fluctuating stresses. The so-called S-N data for a specimen relates the magnitude of the fluctating stress with the number such stress cycles to failure. A typical S-N curve for steel is shown in Fig. 5.

To estimate the fatigue-life of a specimen subjected to a combination of constant amplitude stress-reversals, a damage rule is assumed. A commonly used damage rule (due to Palmgren and Miner) states that if there are k different stress magnitudes in a spectrum, $S_i (1 \leq i \leq k)$, each contributing $n_i(S_i)$ cycles, then if $N_i(S_i)$ is the number of cycles to failure of a constant stress reversal S_i (read from the S-N curve), failure occurs when

$$\sum_{i=1}^{k} \frac{n_i}{N_i} = C \tag{4}$$

For design purposes, C is often assumed to be equal to 1.

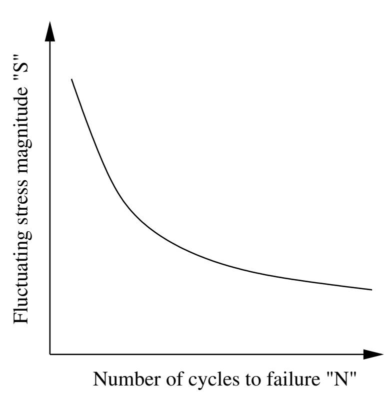

Fig. 5. Typical S-N curve for steel-like materials

Rainflow counting [6], [7] is a method used in the analysis of fatigue data to translate a spectrum of varying stress into sets of reversing stress cycles of constant magnitude. Each such set (often called a "bin") is characterized by the magnitude of the fluctuating stress. For estimating the number of cycles corresponding to each bin, the varying stress spectrum is first converted to a series of tensile and compressive peaks. By observing the pattern the peaks, an estimate of the number of cycles corresponding to each bin is made.

Consider two time series' of stress variation S_1 and S_2 in a structural element where S_2 is a "damped" version of S_1 i.e., the magnitude of stress variation in S_2 is lesser than in S_1. It is useful to note that in comparison to S_2, S_1 will contribute more rainflow counts to "bins" corresponding to larger magnitudes of stress variation. However, a rainflow count excercise on S_2 will yield larger counts for "bins" associated with small magnitudes of stress variation since large stress amplitude variations are moved to bins with smaller stress amplitudes. As will be seen, this fact will be useful in interpreting the load mitigating performance of drive train dampers.

III. COUPLING BETWEEN BLADE EDGE MOTION AND DRIVE TRAIN TORSION

The dynamics of horizontal-axis wind turbines can be expressed using a 6 degree-of-freedom model (Fig. 6). The motion of the tower along the direction of wind is termed "tower fore-aft", while the motion of the tower perpendicular to the wind direction is termed "tower side-to-side". The motion of blade in the plane of rotaion is termed "blade edge" while the out-of-plane motion of the blades is termed "blade flap". The generator azimuth degree-of-freedom accounts for the variable speed nature of the turbine, and essentially constitutes a rigid body rotational mode. The oscillations of the drive-train are governed by the drive-train torsional degree-of-freedom. It should be noted that this torsional motion of the drive train is strongly coupled to blade edge-wise motion.

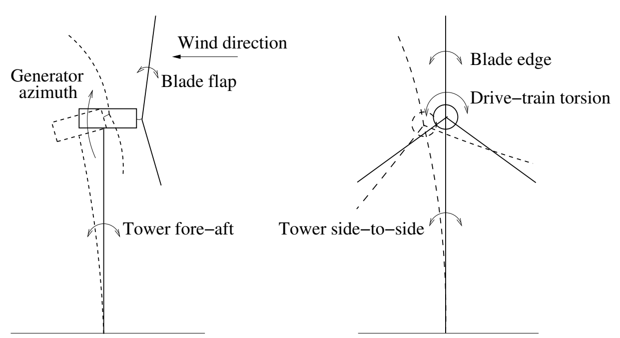

Fig. 6. 6-DOF model of a horizontal axis variable speed wind turbine

The structural components of wind turbines can be viewed as distributed parameter systems. The response of such a system to an external excitation may be written as a weighted sum of an infinite number of basis functions (called "modes"). Associated with each mode is a mode shape - a non-dimensional function which describes the spatial variation of the amplitude of vibration of the mode. The response of distributed parameter systems can often be approximated by superimposing responses of a finite number of 'dominant' modes of the system (the "Assumed Modes" method). The number of modes to be considered for a good approximation of the response is found by ensuring that the norm of the difference between the exact response and the approximated response for a predefined input is less than a critical value. See [8] for more details.

We now derive an expression for how the blade edge and drive-train torsion degrees-o-freedom are coupled. We use the "Assumed Modes" method to model blade deflections. In the interests of simplicity, only the first dominant mode of the blade edge-wise deflection is used. Let $\mu(r)$ denote the mode shape function associated with the dominant blade edgewise deflection mode at a distance r from the axis of the turbine. Then, by definition, the blade edge-wise deflection is given by $x_e(r) = \mu(r)q$, where q denotes the generalized displacement of the first blade edge-wise mode.

Fig. 7 shows a blade section at radius r. It can be seen that due to pitching of the blades, the edge-wise motion of the blade section is not entirely in the plane of rotation. The in-plane component of the blade edge-wise deflection is given by

$$x_p(r) = (\mu(r)q)cos\beta \tag{5}$$

The coupled mass between drive-train and the blade edge-wise degrees of freedom is the torque exerted on the drive-train due to produce unit acceleration along the blade-edge degree-of-freedom. The inertial force associated with a blade section at radius r (in-plane) is

$$f(r) = [\mu(r)m(r)dr]cos\beta \tag{6}$$

where $m(r)$ is the mass per unit length of the section and dr denotes the thickness of the section. This force exerts a torque on the drive-train. The total torque on the drive-train due to the in-plane acceleration of the entire blade is computed by summing up the torques exerted by all such blade sections. By definition, this torque equals the coupled mass. Hence, the expression for the coupled mass may be expressed as:

$$m_c = \int_0^R \{r\mu(r)m(r)dr\}cos\beta \tag{7}$$

As is clear from Eqn. 7, the coupled mass is a function of the blade pitch angle.

Fig. 8 and Fig. 9 represent the power spectral density functions of the simulated drivetrain torque for a typical 1.5 MW wind turbine under two different operating conditions, denoted by OC1 and OC2. In turbines of this rating, the pitch angle can vary from a low of $0-2^o$ below rated wind speed to around 25^o in high wind speed conditions. It can be seen that the drive-train frequency differs by approximately 4% between the two operating conditions.

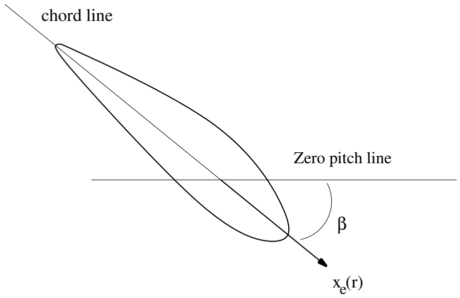

Fig. 7.

The change in the drive-train natural frequency with a change in the operating conditions is not accounted for in the current fixed-gain drive-train damper structures. This situation provides an opportunity to investigate methods of increasing the effectiveness of the drive train damper by scheduling its gains based on pitch angle. This idea is further discussed in the next section.

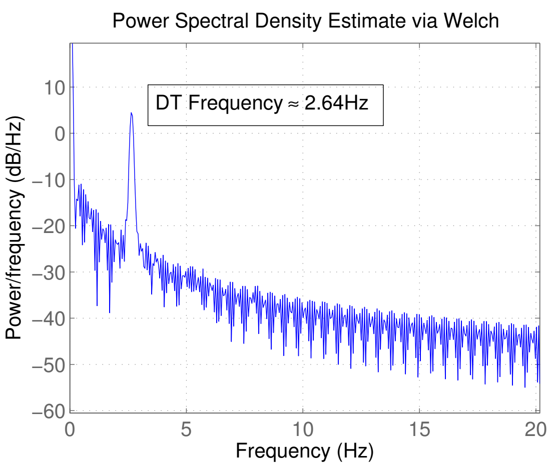

Fig. 8. OC1: mean wind speed = 12 m/s; mean pitch angle $\approx 8^0$

IV. SIMULATION RESULTS

In this section, we present simulation results to demonstrate a case where improved DTD performance is achieved by accommodating change in drive-train fundamental frequency due to pitching. We focus on the above-rated wind speed region, where the goal is to minimize the structural loads on the turbine. As mentioned earlier, the pitch control loop is active in this region. The 'pitch controller' regulates the generator speed to a desired value while the torque demanded by the 'torque controller' is kept constant (see Fig. 2). The drive train damper imposes an additional torque demand on top of that demanded by the 'torque controller'.

A nonlinear model representing the dynamics of a typical 1.5MW horizontal-axis wind turbine was constructed using the wind turbine simulation code FAST [9]. A PI controller acting on the generator speed was used to compute the blade pitch angles for regulating the generator speed.

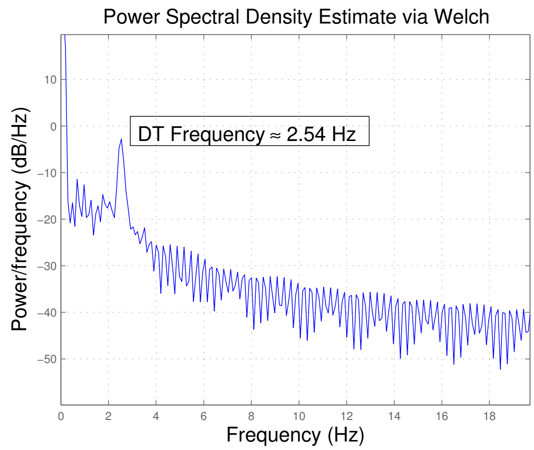

Fig. 9. OC2: mean wind speed = 18 m/s; mean pitch angle $\approx 20^0$

The model was subjected to two sets of wind inputs denoted by OC1 and OC2 with mean wind speeds of $12m/s$ and $18m/s$ respectively. A turbulence intensity of 10% (small in comparison to IEC2A wind class) was used for the simulations. Note that the mean pitch angles ($\beta_1 = 8^o, \beta_2 = 20^o$) required for speed regulation differ due to the difference in mean wind speeds. Based on the computational procedure discussed in Section III, a correspondence can be established between each of the mean pitch angles β_i and the natural frequency ω_{n_i} associated with the torsional oscillations in the drive-train.

To provide corroboration for the need to accommodate variation in the drive train natural frequency due to pitch action, two filters $B_i, (i = 1, 2)$ corresponding to mean pitch situation β_i were designed. The DTD structure used is identical to that proposed by Bossanyi [3] (see Eqn. 2).

$$B_i(s) = \frac{K_i s(1 + \tau_i s)}{s^2 + 2\zeta_i \omega_{n_i} s + \omega_{n_i}^2}$$

The load/oscillation mitigation performance of the dampers was evaluated by using the "Rainflow Counting" method (see Sec. II-C).

Rainflow counts of the drive-train torque for wind input condition OC1 with and without damper B1 included are presented in Table. I. Fig. 10 shows a time history of the drive-train torque with and without the use of a damper.

TABLE I

RAINFLOW CYCLE COUNTS; WIND CONDITION OC1

Torque [N.m]	No Damper	Damper B_1
300	18	75
910	110	19
1520	112	11
2130	19	2

Table. II lists the rainflow counts for the performance of the damper B1 and B2 both simulated for wind input

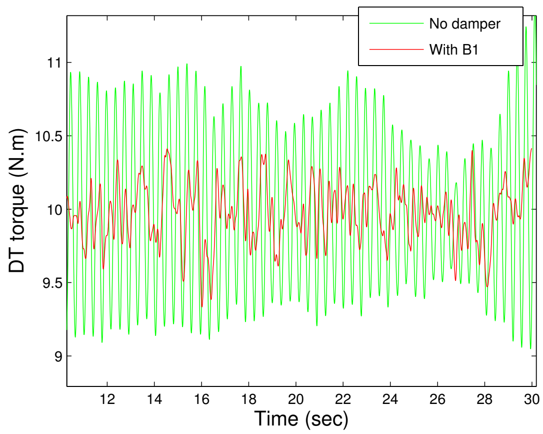

Fig. 10. Variation of Drive Train Torque with and without DTD

condition OC2. This comparison was made to highlight the ill-effects of a fixed-parameter damper (say B_1).

TABLE II
RAINFLOW CYCLE COUNTS; WIND CONDITION OC2

Torque [N.m]	No Damper	DamperB_1	DamperB_2
270	269	173	189
810	67	60	40
1340	4	3	1

The results in Tables I, II and Fig. 10, indicate the following:

- The inclusion of a damper significantly reduces drive-train oscillations
- The DTD that accommodates natural frequency variation displays better load mitigation performance

As discussed earlier, improved oscillation damping leads to a reduction in large amplitude oscillations and consequently increased smaller amplitude oscillations. It should be noted that the exponential nature of the S-N curve (see Sec. II-C) indicates that the number of cycles to failure reduces drastically with increasing stress cycle magnitude. Typically, the number cycles to failure reduce by around $\frac{1}{50}$ times when stress cycle amplitude increases 3 times. The implication is that an increased number of low amplitude stress cycles can be tolerated, if it is accompanied by a significant decrease in the number of cycles in high magnitude region.

V. SUMMARY AND FUTURE WORK

Active drive train damping solutions are deployed in large horizontal-axis wind turbines to mitigate fatigue damage due to drive-train oscillations. In this paper, we studied the effect of blade pitch on the natural frequency associated with drive train torsional oscillations and its impact on drive train damper design. It was shown that due to strong coupling between the blade edge-wise motion and drive train torsional oscillations, the drive train natural frequency varies significantly across the range of pitch angles encountered during operation of large wind turbines. Further, it was shown that this variation, if unaccommodated, leads to sub-optimal drive train damper performance.

The results discussed in this paper provides a platform for the investigation of strategies for scheduling the gains of the drive train damper based on the prevalent pitch angle. Given that in turbulent high wind speed conditions, signifcant variations in pitch angle is not uncommon, it is expected that stability considerations would play a significant role in the design of scheduling algorithms. Addressing such considerations is an issue of immediate interest.

REFERENCES

[1] Danish Wind Industry Association*http://www.windpower.org.*

[2] P. Novak et al., Modeling and Control of Variable-Speed Wind Turbine Drive-System Dynamics, *IEEE Control Systems Magazine*, pp. 28-38, August 1995.

[3] E. A. Bossanyi, *The Design of Closed-loop Controllers for Wind Turbines*, Wind Energy, vol. 3, 2000, pp. 149-163.

[4] T. Burton, D. Sharpe, N. Jenkins and E. A. Bossanyi, *Wind Energy Handbook*, John Wiley & Sons, Ltd, NY; 2001.

[5] S. Suryanarayanan and A. Dixit, *On the Dynamics of the Pitch Control Loop in Horizontal-Axis Large Wind Turbines*, Proc. of the American Control Conference, Portland, 2005, pp. 686-690.

[6] I. Rychlik, *A New Definition of the Rainflow Cycle Counting Method*, Intl. Journal of Fatigue, vol. 9, pp. 119-121, 1987.

[7] N. E. Dowling, *Fatigue Prediction for Complicated Stress-Strain Histories*, Journal of Materials, vol. 7, pp. 71-87, 1972.

[8] L. Meirovich, *Elements of Vibration Analysis*, McGraw-Hill , NY; 1986.

[9] FAST: An aero-elastic design code for horizontal axis wind turbines *http://wind.nrel.gov/designcodes/simulators/fast*