FAULT TOLERANT CONTROL DESIGN OF FLOATING OFFSHORE WIND TURBINES

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ABSTRACT. This work is concerned with active vibration mitigation in wind turbines (WT) but not through the use of specifically tailored devices. Instead, a general control scheme is designed for torque and pitch controllers based on a super-twisting algorithm, which uses additional feedback of the fore-aft and side-to-side acceleration signals at the top of the WT tower to mitigate the vibrational behavior. In general, proposed methods to improve damping through pitch and torque control suffer from increased blade pitch actuator usage. However, in this work the blade pitch angle is smoothed leading to a decrease of the pitch actuator effort, among other benefits evidenced through numerical experiments. The most frequent faults induce vibrations in the corresponding WT subsystems. In fact, vibration monitoring has been recently used for fault diagnosis Thus, by means of vibration mitigation, different faulty conditions can be alleviated leading to a passive fault tolerant control. In this work, coupled non-linear aero-hydro-servo-elastic simulations of a floating offshore wind turbine are carried out for one of the most common pitch actuator faults.

KEYWORDS: wind turbine, vibration mitigation, super-twisting algorithm

1 INTRODUCTION

The future of wind energy industry passes through the use of larger and more flexible wind turbines (WT) in remote locations, which are increasingly offshore to benefit stronger and more uniform wind conditions. Cost of operation and maintenance of offshore WT is among 15-35% of the total cost. From this, 80% comes from unplanned maintenance [3]. Thus, a promising way to contribute to the increasing requirements and challenges passes through applying low-cost advanced fault tolerant control (FTC) schemes.

The objective of FTC is to design appropriate controllers such that the resulting closed-loop system can tolerate abnormal operations of specific control components and retain overall system stability with acceptable system performance. In general, the FTC approaches can be classified into active or passive. In active schemes, the controller is reconfigured whenever a fault is detected. In passive schemes, the controller's structure is fixed. In this work, we concentrate in passive FTC.

In previous works (see [4], [5], and [6]), it has been proposed the use of classical sliding mode control (SMC) for WT control. Such approaches deal efficiently with the power regulation objective and provide the advantage of robustness against system uncertainties and perturbations, such as measurement noise.

Although classical SMC has shown good performance in an uncountable number of applications, its well-known drawback has been the discontinuous behavior of the computed control inputs that may derive into a high-frequency oscillation known as chattering (see [7]). Among great variety of chattering suppression methods, so-called high-order sliding mode control has been intensively studied within the last decade (see, for example, [8]) and has been applied in a wide variety of fields (see, for instance, [9], [10], [11], and [12]). The twisting and super-twisting control algorithms are intended for designing the second-order sliding mode. While the twisting algorithm needs an additional differentiator (preserving the structural requirement for the common first-order sliding mode), the super-twisting algorithm (STA) does not need it. The remarkable properties of the STA are: a) accurately regulating and tracking accomplished with finite-time convergence; b) as the control input is a continuous state function, there is a reduction of mechanical stresses (see [13]) and chattering; c) time derivative of the output is not needed; d) robustness with respect to various internal and external disturbances and model uncertainties; e) relatively simple control laws that can be designed based on nonlinear models. These properties explain high level of research activity related to stability analysis, estimation of the convergence time, and estimation of the admissible range of disturbances (see, among others, [14], [8], [15], and [16]). In this work, new torque and pitch controllers are proposed based on the STA by introducing the acceleration signals at top tower as a feedback perturbation signal, with the purpose of reducing vibrations.

The most frequent WT faults induce vibrations in the corresponding WT subsystems [1]. In fact, vibration monitoring has been recently used for fault diagnosis [2]. Thus, by means of vibration mitigation, different faulty conditions can be alleviated leading to a passive FTC. The problem of alleviating vibrations in WT systems is relatively new, being an efficient straightforward method the use of vibration control devices under passive, active or semi-active schemes (e.g., [17]). In this regard, the main contribution of this work is to propose new control techniques which provide active vibration mitigation in WT. The proposed controllers are based in the super-twisting algorithm (STA) by using feedback of the generator shaft speed as well as the *fore-aft* and *side-to-side* acceleration signals of the WT tower.

Coupled non-linear aero-hydro-servo-elastic simulations of a floating offshore wind turbine (OWT) are carried out for an hydraulic pitch actuator fault. In particular, the tension leg platform (TLP) floating type of WT is used [18].

2 WIND TURBINE DESCRIPTION

The OWTs are installed far off the coast and the water depths can be varying from shallow to deep. The National Renewable Energy Laboratory (NREL) has classified 0-30m as shallow water, 30-60m as transitional waters and greater than 60m as deepwater for installing OWTs. Most of the floating wind turbines are applied for water depths of approximately 60 m to 900 m. Even at transitional depths of 30 m to 60 m, floating structures could provide a viable alternative to conventional fixed structures (monopile, tripod, and jacket). Several support platform configurations are possible for floating offshore wind turbines, particularly considering the variety of the mooring systems, tanks, and ballast options that are used in the offshore oil and gas industries. In this work, a tension leg platform OWT is used. A complete description of the wind turbine model can be found in [18]. Coupled non-linear aero-hydro-

Table 1: Gross properties of the wind turbine [19].

Reference wind turbine	
Rated power	5MW
Number of blades	3
Rotor diameter	126m
Hub Height	90m
Cut-In, Rated, Cut-Out Wind Speed	3m/s, 11.4m/s, 25m/s
Rated generator speed	1173.7rpm
Nominal torque	40681.5kNm
Gearbox ratio	97

servo-elastic simulations are performed with FAST [19]. The main properties of this turbine are listed in Table 1.

In this work, new controllers are proposed and its performance is compared with respect to the baseline torque and pitch controllers described in the technical report [19] by the U.S. Department of Energy's National Renewable Energy Laboratory (NREL). These baseline controllers are used as a reference by research teams throughout the world to quantify the benefits of advanced land- and sea-based torque and pitch controllers.

3 PROBLEM STATEMENT

To make easier the control system design, most control strategies for WT uncouple the control problem into two different single input single ouput (SISO) control loops: the torque and the pitch controllers (see, for example, [3], [20], [21], [22]). Although the uncoupled assumption (used also in this work), these controllers work collaboratively in the WT overall closed loop system (see, for instance, [20]). In this paper, scalar STA (see [23]) is used to design new torque and pitch controllers. A comprehensive analysis of the STA is conducted, for instance, in [8].

The most frequent WT faults induce vibrations in the corresponding WT subsystems [1]. In fact, vibration monitoring has been recently used for fault diagnosis [2], [24]. Thus, by means of vibration mitigation different faulty conditions can be alleviated leading to a passive FTC strategy. Therefore, in this work, an extra control objective for the proposed controllers is vibration mitigation. In particular, the torque control objectives are to regulate the electrical power and mitigate vibrations in the *side-to-side* direction and the pitch control objectives are to regulate the generator speed and mitigate vibrations in the *fore-aft* direction. Note that both controllers work together to obtain an electrical power regulated to the rated electrical power and, at the same time, a generator speed regulated to its nominal value.

3.1 Controllers design

On one hand, we propose the scalar STA-based torque controller

$$\tau_c(t) = -\alpha_1 \sqrt{|P_e - \mathrm{Pe}_n|}\mathrm{sign}(P_e - \mathrm{Pe}_n) + y, \tag{1}$$

$$\dot{y} = -\alpha_2 \mathrm{sign}(P_e - \mathrm{Pe}_n) + \alpha_3 a_{ss}(t),$$

where τ_c is the generator torque reference, $P_e(t)$ is the generated power, Pe_n is the nominal power of the WT, $\alpha_1, \alpha_2, \alpha_3 > 0$ and $a_{ss}(t)$ is the *side-to-side* acceleration measured at the tower top. Note that we introduce the acceleration as a perturbation signal to give the controller the ability to face with vibrations (and faulty conditions). A stability analysis for this controller is given in the next subsection.

On the other hand, we propose to modify the baseline gain-scheduling pitch controller in the form

$$\beta_c(t) = K_p(\theta)(\hat{\omega}_g(t) - \omega_{g,n}) + K_i(\theta)z, \tag{2}$$

$$\dot{z} = \mathrm{sign}(\hat{\omega}_g(t) - \omega_{g,n}) + \alpha_4 a_{fa}(t),$$

where $\beta_c(t)$ is the reference pitch angle, $\hat{\omega}_g(t)$ is the filtered generator speed, $\omega_{g,n}$ is the nominal generator speed, $\alpha_4 > 0$ and $a_{fa}(t)$ is the *fore-aft* acceleration measured at the tower top. Note that the acceleration is introduced, similarly to the torque controller, as a perturbation signal. For the proposed pitch controller, as it is a gain-scheduling proportional integral control, the controller gains are *heuristically* tuned following the same procedure as in [19]. Finally, the pitch angle actuators generally present hard constraints on their amplitude and their speed response. Because of this, a pitch limit saturation to a maximum of 45° and a pitch rate saturation of 8°/s are implemented (see [19]) to avoid pitch actuator damage.

The block diagram in Figure 1 shows the connections between the WT and the proposed torque and pitch controllers.

3.2 Torque control stability analysis

For a perfectly rigid low-speed shaft, a single-mass model for a wind turbine can be considered ([25, 26, 27, 28]),

$$J_t \dot{\omega}_g = T_a - \tau_c, \tag{3}$$

where J_t is the turbine total inertia (Kg m^2), τ_c is the generator torque (Nm), and T_a is the aerodynamic torque (Nm) described as

$$T_a = \frac{1}{2}\rho\pi R^2 \frac{C_p(\lambda,\beta)}{\omega_r} u^3, \tag{4}$$

where ρ is the air density (kg/m^3), R is the rotor radius (m), ω_r is the rotor speed (rad/s), u is the wind speed (m/s), and $C_p(\lambda,\beta)$ is the power coefficient (bounded by the Betz limit). Note that, due to physical constraints, the aerodynamic torque is bounded. Thus, it is realistic to assume that $0 < T_a \leq \gamma, \ \forall t \geq 0$.

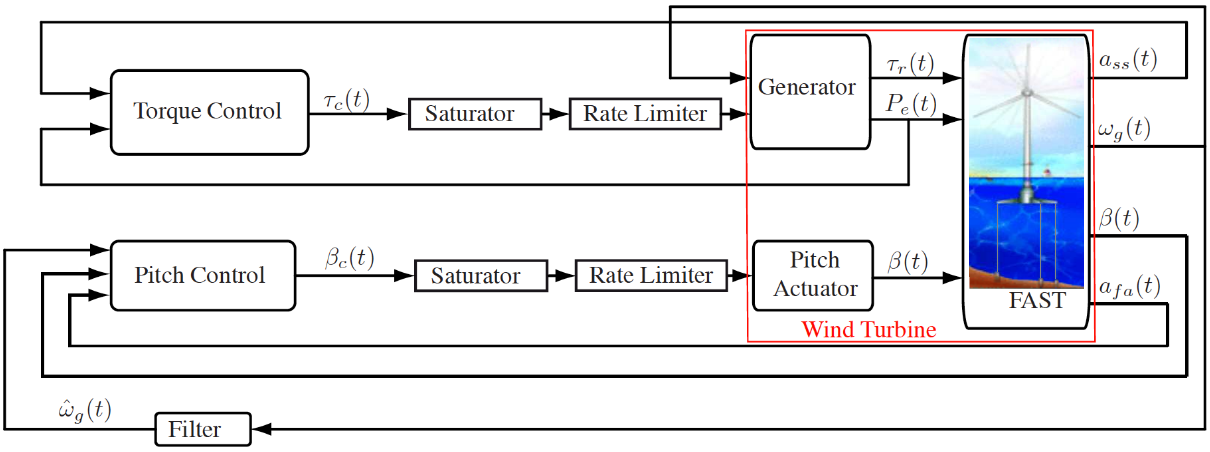

Figure 1: Block diagram of the closed loop system.

The generator-converter system can be approximated by a first-order differential equation, see [29], which is given by:

$$\dot{\tau}_r(t) + \alpha_{gc}\tau_r(t) = \alpha_{gc}\tau_c(t), \tag{5}$$

where τ_r and τ_c are the real generator torque and its reference (given by the controller), respectively. In the numerical simulations, $\alpha_{gc} = 50$, see [19]. Moreover, the power produced by the generator, $P_e(t)$, may be given by (see [29]):

$$P_e(t) = \eta_g\omega_g(t)\tau_r(t), \tag{6}$$

where η_g is the efficiency of the generator and ω_g is the generator speed. In the numerical experiments, $\eta_g = 0.98$ is used, see [29].

The STA-based torque control objective is to regulate the electrical power. Thus, we define the error:

$$e(t) = P_e(t) - \mathrm{Pe}_n,$$

and the control objective is that it converges to zero as time goes on. It is obvious that

$$\dot{e}(t) = \dot{P}_e(t) = \eta_g\left[\dot{\omega}_g(t)\tau_r(t) + \omega_g(t)\dot{\tau}_r(t)\right].$$

Using (5) and (3), from the generator-converter model and WT model respectively, the error dynamics can be written as

$$\dot{e}(t) = \eta_g\left[J_t^{-1}\left(T_a - \tau_c\right)\tau_r(t) + \alpha_{gc}\omega_g(t)\left(\tau_c(t) - \tau_r(t)\right)\right],$$

and, assuming that $\tau_c(t) - \tau_r(t) \approx 0$, it can be simplified to

$$\dot{e}(t) = \eta_g J_t^{-1} T_a\tau_c(t) - \eta_g J_t^{-1}\tau_c^2.$$

Finally, linearizing the previous dynamics around $\tau_c(t) = 0$, the error dynamics yield

$$\dot{e}(t) = \eta_g J_t^{-1} T_a \tau_c(t),$$

and, as $\eta_g J_t^{-1} T_a$ is positive and bounded, to prove the local stability of this system is equivalent to study the local stability conditions of the system

$$\dot{e}(t) = \tau_c(t).$$

This system, after substituting (1) gives the closed loop error dynamics,

$$\dot{e}(t) = -\alpha_1 \sqrt{|e|}\mathrm{sign}(e) + y, \tag{7}$$

$$\dot{y} = -\alpha_2 \mathrm{sign}(e) + \alpha_3 a_{ss}(t). \tag{8}$$

Since we consider that the *side-to-side* acceleration, $a_{ss}(t)$, is a perturbation signal (giving the controller the ability to face with vibrations), system (7)-(8) is stable as has been proven in [16]. This finally concludes the stability of the proposed torque control.

4 SIMULATION RESULTS

This section presents the performance evaluation of the proposed STA controllers with respect to the baseline control system in [19] that is used as a frame of reference. Simulations were conducted for a realistic wind speed sequence with mean speed of 14 m/s, and over 800 s of run time. This wind speed sequence is illustrated in Fig. 2 with the waves elevation. The rated and cutout wind speeds are 11.4 m/s and 25 m/s, respectively. Thus, the wind profile lies in the above rated work region.

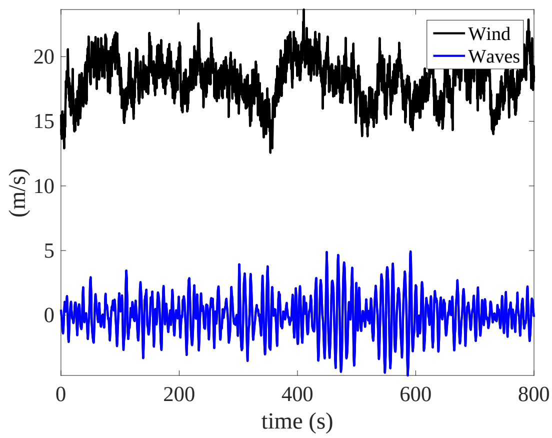

Figure 2: Wind speed (m/s) and wave elevation (m).

Here, performance indices are given to present a comparison between STA and baseline controllers:

$$J_{fa}(t) = \int_0^t |a_{fa}(\tau)|\, d\tau,\ [m/s]$$

$$J_{ss}(t) = \int_0^t |a_{ss}(\tau)|\, d\tau,\ [m/s]$$

$$J_P(t) = \int_0^t |P_e(\tau) - \mathrm{Pe}_n|\, d\tau,\ [J]$$

where $a_{fa}(t)$ and $a_{ss}(t)$ are the *fore-aft* and the *side-to-side* accelerations, respectively, at the tower top.

4.1 Hydraulic leakage of pitch actuator

One of the most common pitch actuator faults is the hydraulic leakage. This fault changes the dynamics of the pitch actuator. A detailed description can be found in [30], [29], and [31].

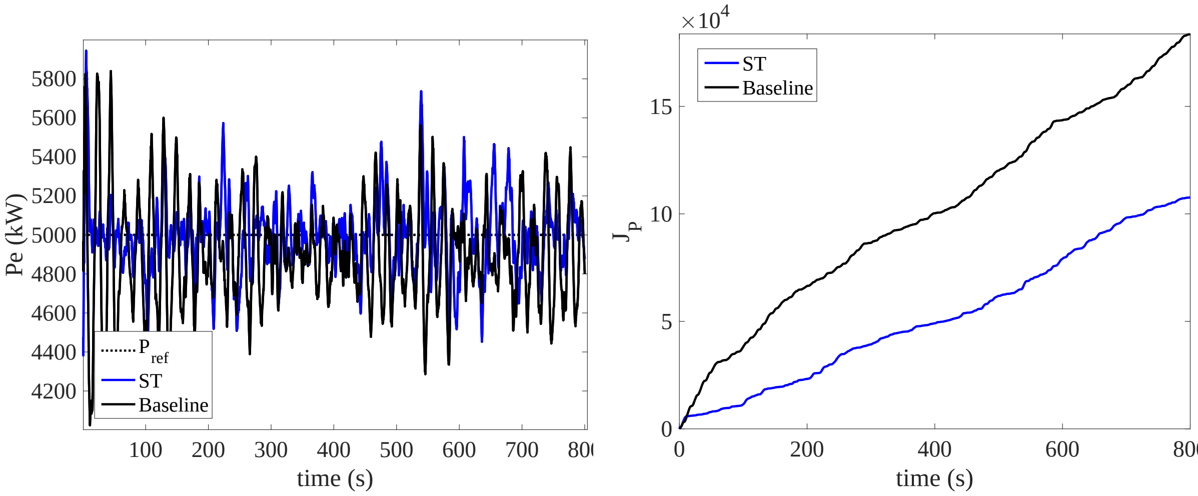

Figure 3: Electrical power (left) and J_P index (right).

Figure 3 presents the electrical power (left) and J_P index (right) for the proposed STA controllers and compared to the baseline ones. Results show that the proposed controllers improve the power generation quality. Due to the rate-limiter action and the complexity of the WT model used for simulation (FAST), the finite-time convergence behavior of the STA torque controller is not evidenced in the results, as can be seen in Figure 3 (left). The J_P performance index is improved, that is the error in the regulation of the electrical power is reduced. In a 800 seconds simulation, the accumulated error is almost halved with respect to the baseline strategy as can be seen in Figure 3 (right).

Figure 4 (left) displays the generator speed. It is observed that higher oscillations are obtained for the baseline controllers. The proposed STA does not induce increased mechanical stress as there are no

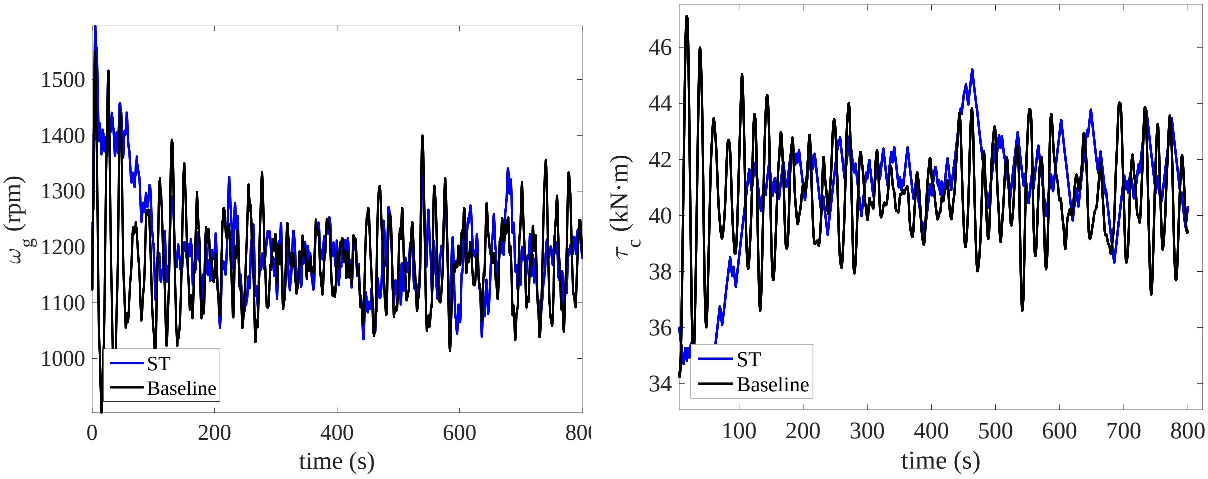

Figure 4: Generator speed (left) and torque control (right).

strong torque variations, as can be seen in Figure 4 (right). The torque generator remains smooth and tracks more efficiently the wind fluctuations than in standard control. Indeed, and as expected, this leads to a reduction of the accelerations in the tower, as can be seen in Figure 5, where the time histories and the performance indices J_{fa} and J_{ss} are displayed. It is noteworthy how the indices show that accelerations in the *fore-aft* direction have been significantly improved whereas accelerations in the *side-to-side* direction are comparable to the ones obtained with the baseline control.

Recall that, when designing the pitch angle control loop, it is of great importance to avoid a high activity of the pitch, since it could not only damage the pitch actuators but also give rise to unstable modes of operation, see, for instance, [20]. The pitch control, shown in Figure 6, is smoothed with the STA-based controllers. This lower pitch activity leads to lower mechanical stress (vibration mitigation) spreading the wind turbine lifetime and also resulting in softer output power.

Remark 1. The gains $\alpha_1 = 0.1$, $\alpha_2 = 200$, $\alpha_3 = 1$, and $\alpha_4 = 5$ are used in the simulations. They were selected in order to reduce the *fore-aft* motion. However, other gain values could be used, for example, to obtain also an improvement in the *side-to-side* direction.

5 CONCLUSIONS

This work focused on the design of a robust STA for efficient and reliable control of a large floating offshore wind turbine in the full load region, and in the presence of wind turbulences and a pitch actuator realistic fault scenario. Compared to the baseline controllers, the developed STA-controllers have been able to improve the overall performance of the wind turbine in this faulty condition, and to reduce the *fore-aft* and *side-to-side* accelerations with respect to the baseline control.

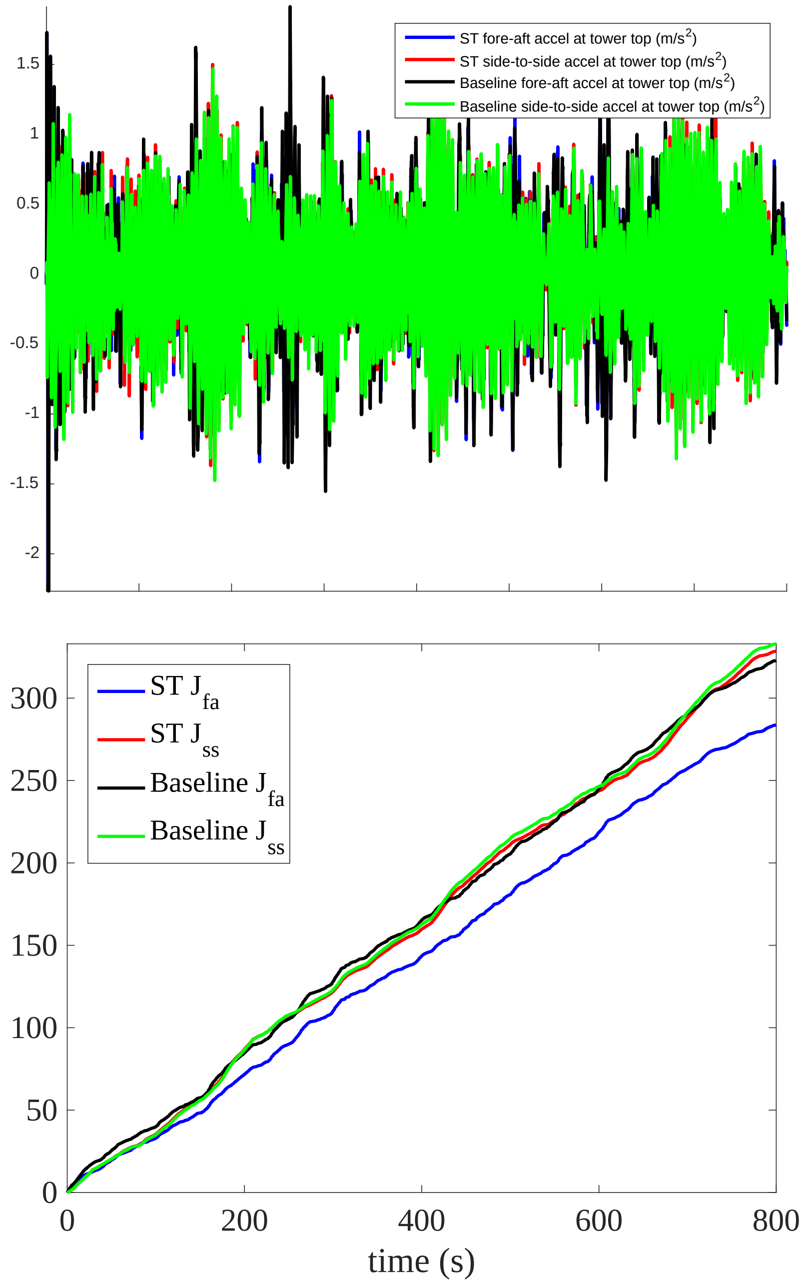

Figure 5: *Fore-aft* and *side-to-side* accelerations (top) and related J_{fa} and J_{ss} indices (bottom) at the tower top.

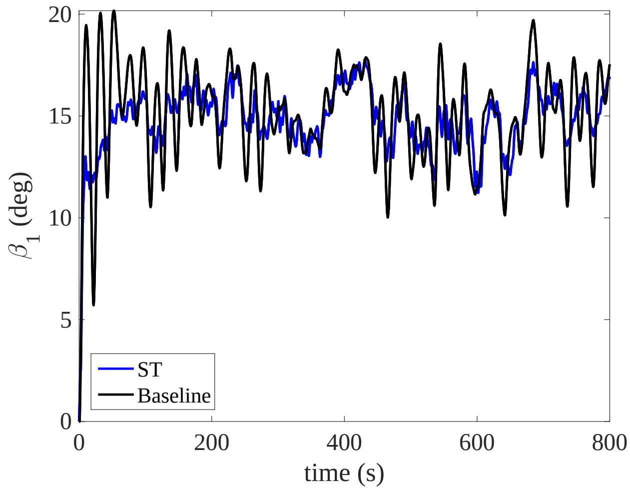

Figure 6: Pitch angle.

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