Control-oriented Modelling and State Estimation
of Tidal Turbines with Pitch Control

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Abstract: This contribution presents the dynamic modelling of horizontal axis tidal turbines (HATT) for the purpose of online mechanical load estimation, simulation and control. The turbine model is used as internal representation in nonlinear filter algorithms to provide estimates for the states and critical loads. Advanced sigma-point Kalman filters (SPKF) have been chosen to cope with the inherent nonlinearity due to hydro-elastic coupling. Illustrative simulation results emphasize the effective filter performance and its potential for predictive maintenance. The estimated turbine states are used successfully to reconstruct accurately the mechanical loads in a sophisticated manner.

1 Introduction
Offshore hydro power offers a great potential for clean and sustainable renewable energy generation. It can be harvested, for instance, by tidal turbines with horizontal rotational axis [1]. Unlike offshore wind turbines, the inflow stream of these turbines is in general more predictable, though unknown deterministic excitations by surface waves are likewise critical for maintenance. Hence, substantiated concepts for plannable maintenance are highly desirable to achieve a reliable power production which are unfortunately not available today. This contribution addresses the first step to plannable maintenance, namely the modeling of turbine dynamics and loads from a control engineer’s perspective and the state estimation based on available sensor measurements. Therefore, the remainder of this paper is structured as follows: Section 2 elaborates on the semi-physical white-box modeling of horizontal axis tidal turbines (HATT). Section 3 presents the proposed and implemented reference turbine as a standard parametrization. Section 4 gives a concise introduction to the theory of nonlinear state estimation using sigma-point Kalman filters (SPKF) and the choice of the design parameters. Section 5 presents the illustrative simulation results. Finally, Section 6 summarizes the main outcomes and provides an outlook on relevant future work.

2 Modeling of Tidal Turbines
This section deals with the derivation of the control-oriented dynamic model used for state estimation. The control-oriented modelling is not intended to deduce a detailed fluid mechanics representation, but rather a useful design model for the control engineer. Thus, this section presents a simplified HATT model and a parameter set for the so-called reference tidal turbine.
2.1 Dynamic Model Equations

To obtain the control-oriented model that covers the simplified fluid and mechanical dynamics, Newton’s principle of conservation of angular and linear momentum is applied to the drive-train and nacelle of the tidal turbine. A similar approach has been discussed for wind turbine application in [2]. A technical drawing of a two-bladed tidal turbine is shown in Fig. 1.

![Technical drawing of the two-bladed tidal turbine](image)

Fig. 1. Technical drawing of the two-bladed tidal turbine

A perfect alignment with the tidal current’s direction is assumed (which is practically ensured by active yaw control such that the rotor always faces the current perpendicularly). With this assumption, the differential equations governing the turbine’s drive-train dynamics are derived from conservation of angular momentum as follows:

\[
\Theta_r (\dot{\phi}_g + \Delta \dot{\phi}) + \Theta_g i_{gb}^2 \ddot{\phi}_g = \frac{\rho}{2} \pi R^3 C_M(\lambda, \beta)(v_h - \dot{x}_T)^2 - i_{gb} M_g
\]

\[
\Theta_r (\dot{\phi}_g + \Delta \dot{\phi} + 2 \zeta_{dt} \omega_{dt} \Delta \phi + \omega_{dt}^2 \Delta \phi) = \frac{\rho}{2} \pi R^3 C_M(\lambda, \beta)(v_h - \dot{x}_T)^2
\]

wherein \( \Theta_r \) is the rotor inertia, \( \Theta_g \) is the generator inertia (considered on the high-speed side of the gearbox), \( i_{gb} \) is the gearbox ratio, \( \rho \) is the water mass density and \( R \) is the blade tip radius. \( \omega_{dt} \) is the drive-train eigenfrequency and \( \zeta_{dt} \) the corresponding modal damping. \( C_M(\cdot) \) is called the hydrodynamic torque coefficient and \( \lambda \) denotes the so-called tip-speed-ratio (TSR) defined by

\[
\lambda = \frac{\phi R}{v_h - \dot{x}_T}
\]

which is a dimensionless rotor speed. Moreover, \( \phi = \phi_g + \Delta \phi \) is the rotor angular speed. \( M_g(t) \) is the controlled generator torque and \( \beta(t) \) is the (controlled) collective blade pitch angle. Both quantities are control inputs to the tidal turbine system. The main disturbance input is the rotor-effective tidal velocity \( v_h(t) \) which is obviously time-dependent due to deterministic and stochastic shares.
From conservation of linear momentum applied to the nacelle, the 2nd order differential equation for the nacelle’s fore-aft motion (in axial direction)

\[ m_T(\ddot{x}_T + 2\zeta_T\omega_T\dot{x}_T + \omega_T^2x_T) = \frac{6}{2}\pi R^2 C_T(\lambda, \beta)(v_h - \dot{x}_T)^2 \] (4)

is derived where \( \omega_T \) is the first natural frequency of the tower-rotor-foundation system, \( \zeta_T \) is the modal damping and \( C_T(\cdot) \) is the hydrodynamic thrust coefficient. Consider Eq. (2) again. Therein, the moment coefficient \( C_M(\cdot) \) relates to the hydrodynamic power coefficient as follows

\[ C_P(\lambda, \beta) = \lambda C_M(\lambda, \beta) \] (5)

which can be perceived as the dimensionless power equation of the HATT. The power coefficient basically describes the conversion of the tidal stream’s power into mechanical power at the rotor. This power coefficient has a well-known upper physical limit \( C_P^* = 16/27 \) (known as the Lancaster-Betz limit) which is the theoretical maximum [4]. In addition, the coefficient is also defined by

\[ C_P := \frac{P_a}{2\pi R^2(v_h - \dot{x}_T)^2} \] (6)

Eq. (6) also provides a definition for the instant hydrodynamic power \( P_a \) available in the tidal current. Looking at the LCoE, the generator power

\[ P_R(t) = \eta_R i_{gb}(t) M_R(t) \] (7)

is one quantity of utmost relevance. In Eq. (7), \( \eta_R \) is denoted as the combined gearbox-generator efficiency. The power must be controlled such that the energy

\[ E(T) = \int_0^T P_R(t)dt \] (8)

is maximized subject to fluctuating hydro inflow conditions \( v_h(t) \) for an operational period between \( T \in [25, 30] \) years. This control objective is identical for tidal turbine control (TTC) and for wind turbine control (WTC). That is why, principles of WTC are applicable to tidal turbines as well (cf. [5]).

### 2.2 Mechanical Turbine Loads

There are four critical turbine loads which are decisive for life-time assessment and fatigue load discussion. First, the rotor thrust force

\[ F_T(\lambda, \beta, v_h, \dot{x}_T) = \frac{6}{2}\pi R^2 C_T(\lambda, \beta)(v_h - \dot{x}_T)^2 \] (9)

is relevant for the design of the tripod construction including the foundation in the seabed. It depends strongly on the tidal velocity and also the oscillations of the nacelle. Furthermore, the reaction force at the tripod reads

\[ F_R(\dot{x}_T, x_T) = m_T(2\zeta_T\omega_0\dot{x}_T + \omega_0^2 x_T) \] (10)

which depends simply on the nacelle velocity \( \dot{x}_T \) and the position \( x_T \). Moreover, the hydrodynamic torque

\[ M_h(\lambda, \beta, v_h, \dot{x}_T) = \frac{6}{2}\pi R^3 C_M(\lambda, \beta)(v_h - \dot{x}_T)^2 \] (11)

influences the rotor and blade loads. Finally, the mechanical drive-train torque
\[ M_{dt}(\Delta \phi, \Delta \varphi) = \Theta_r \left( 2 \zeta_{dt} \omega_{dt} \Delta \phi + \omega_{dt}^2 \Delta \varphi \right) \] (12)

need to be assessed for monitoring of drive-train fatigue loads. As can be seen in Eqs. (9-12), all mechanical states as well as the disturbance input \( v_h(t) \) have a significant influence on the instantaneous loads. Unfortunately, in a real tidal turbine these states and disturbances are not accurately measured or often cannot be measured (due to sensor costs or lack of sensors). Hence, an efficient estimation algorithm is desired in order to supply the best possible estimates to the controller (cf. Section 4).

3 The Reference Tidal Turbine

For the purpose of simulation studies and to assess the potential benefits of tidal turbines, a reference horizontal axis turbine is defined in the following. A basic turbine design proposed by Faudot & Dahlhaug in [7] is taken up and extended such that it can be used for estimator design and simulation. The power rating of this tidal turbine is proposed as 1 MW concept with two rotor blades.

3.1 Main Model Parameters

The turbine is considered as variable-speed variable-pitch machine where the rotor speed varies in order to improve the hydrodynamic performance. A geared drive-train is chosen to match the nominal rotor speed \( n_{r,0} = 15.9 \) rpm with the nominal speed \( n_{g,0} = 1176 \) rpm of asynchronous induction generator. Furthermore, a stiff-stiff design for the tripod-tower is chosen in order to mitigate the forced excitations from rotor-frequency (1p) loads. Hence, the first natural frequency of the structure \( f_0 = 0.8 \)Hz lies well above the 1p and 3p frequency excitation (harmonics of the rotor speed).

<table>
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<th>Description</th>
<th>Symbol</th>
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<tr>
<td>Drive-train first eigenfrequency</td>
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<td>kg</td>
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<td>Water depth</td>
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Table 1: Model parameters and nominal values for the reference tidal turbine
The eigenfrequency then writes $\omega_0 = 2\pi f_0$. For a cost-effective over-all design, a two-bladed concept is preferred over the three-bladed concept since the additional blade increases the maintenance costs by 50%. Table 1 provides the complete overview of mechanical model and system parameters.

3.2 Assessment of Hydrodynamic Coefficients

Besides the lumped parameters in Table 1, the dynamic response of the turbine is strongly influenced by the hydrodynamic coefficients of rotor torque $C_M(\lambda, \beta)$, rotor thrust $C_T(\lambda, \beta)$ and rotor power $C_P(\lambda, \beta)$. These coefficients represent the main non-linearity of the hydrokinetic system. Their derivation from known blade geometry and profile data using the blade element theory and profile theory (BEM) is presented exemplarily for wind turbines in [8]. The approach is similarly applicable to tidal turbines. For the reference tidal turbine, the blade geometry, proposed in [7], is shown in Fig. 2. It consists of a cylinder-type profile at the hub as well as S816, S825 and S826 profiles [9]. The detailed hydrodynamic curves are not displayed for the sake of due brevity.

![Fig. 2. Blade of the tidal turbine in 3D representation [7]](image)

4 Nonlinear state estimation

State estimation is a central topic for numerous technical and biological applications. Especially advanced state-feedback controllers need the full state information to be accessible in real-time. A major obstacle for the reliable utilization is the requirement not only for accurately known full dynamic state, but also for external disturbances and critical system parameters.

In this paper, the cubature Kalman filter (CKF) is used for the task of nonlinear state estimation. The CKF is one family member of the so-called sigma-point filter [6]. Due to the paper’s limited scope, details about the algorithm cannot be supplied here. Consult [3] for application of CKF to wind turbine control systems. The SPKF are efficient algorithms that recursively assess state estimates based on the available sensor information and an internal filter model. That is, a one-step forecast predicts the future dynamics and the sensor information is then used to correct this prediction. More advanced concepts also consider the adaptivity of the algorithm to cope with time-varying process and sensor noise [9]. However, this cannot be considered here.
Additionally, the choice of the free filter parameters is crucial for the estimator’s performance. Thus, performance criteria and an automated design approach is preferable [9].

5 Simulation results
For the simulation study, the reference HATT from Section 3 and the cubature Kalman filter (CKF) from Section 4 are used. The turbine is implemented in the high-fidelity simulator FAST [11] to test the filter performance under realistic conditions. The obtained simulation results for states, loads and disturbances serve as benchmark conditions for the CKF estimates.

5.1 Test Scenario Description
The turbulent water inflow in Fig. 3 is applied to the tidal turbine. The selected inflow scenario includes velocities below as well as above rated tidal speed \( v_0 \). Moreover, the effects of surface waves are considered with an amplitude of 0.5m/s and a typical wave period of 7s. Hence, the filter performance is tested also with critical excitation.

![Fig. 3. Turbulent water inflow with an average tidal velocity of 2.5m/s and a turbulence intensity of 9% including a harmonic excitation with a wave period of 7s in order to investigate water surface effects.](image)

It must be noted that the only measured quantities are the generator torque \( M_g \) and the pitch angle \( \beta \) as well as the measured generator speed \( n_g \) and nacelle acceleration \( \ddot{x}_T \). These are shown in Fig. 5 on the left. Especially, both controlled inputs show remarkably the intense effect of the wave excitation. Nevertheless, the tidal turbine controller ensures the decent power production despite strong fluctuations of the inflow’s hydrodynamic power (cf. Fig. 4).

![Fig. 4. Comparison of hydrodynamic power at the rotor side and produced electrical generator power](image)
5.2 Estimation Results

First, the state estimates are shown in Fig. 5 on the right. This includes also the tidal velocity $v_h$ which is a unknown disturbance input to the tidal energy system. Again, it is emphasized that the tidal velocity is not measured and is only estimated based on available standard sensor equipment. Still, all dynamic states are reconstructed very accurately (cf. Fig. 5) which is the prerequisite for load estimation.

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Fig. 5. Control inputs, sensor outputs and state estimates

Fig. 6. Estimates of mechanical loads and powers, tip-speed-ratio and power coefficient
The estimates for critical mechanical loads are reconstructed using the knowledge of the system states (Fig. 5) and the model Eqns. (9-12). Thus, the more accurate state estimates and model are, the more accurate the load prediction is in general.

This proves to be correct also for load estimation of tidal turbines, as Fig. 6 indicates. Moreover, it demonstrates that estimates for power coefficient, tip-speed-ratio and nacelle acceleration are assessable simultaneously. These estimated loads are essential for predictive maintenance concepts since the precise load forecast (and also a retrospect) is one due requirement to quantify the remaining life-time of critical system components.

6 Conclusions

The paper demonstrates impressively what is in principle possible to achieve with advanced state estimation techniques. Internal states, disturbance inputs and also mechanical loads are assessable in real-time with existing and already installed standard sensors. The model-based filter algorithms (which constitute software-based virtual sensors) play a key role in this process of prediction and correction. Thereby, the necessity for expensive hardware sensors is (at least partially) alleviated whenever there is sufficient a priori knowledge about the dynamic system under investigation available. Therefore, this paper has introduced a reduced-order model which serves the purpose of load estimation with limited computational power perfectly. Moreover, a 1MW reference tidal turbine has been proposed to study future tidal energy systems in greater detail.

Future work will bring the predictive maintenance of critical components into focus. This topic is presently penetrating various industrial sectors like process industry, renewable energy and also life science (to name only a few). Yet for successful application, extensive know-how about components, component wear and simplified fatigue models is needed. This knowledge must be supplemented by a thorough study of available concepts of machine learning and/or system identification.

References


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