Comparison of computational modelling and field testing of a small wind turbine operating in unsteady flows

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Abstract. Small horizontal-axis wind turbines are likely to operate in a broad range of operating flow conditions, often in highly turbulent flow, due, in part, to their varied site placements. This paper compares the computational simulations of the performance of a 5 kW horizontal-axis wind turbine to detailed field measurements, with a particular focus on the impact of unsteady operating conditions on the drivetrain performance and generator output. Results indicate that the current Blade Element Momentum Theory based aerodynamic models under-predict the effect of high turbine yaw on the rotor torque, leading to a difference between predicted and measured shaft speed and power production. Furthermore, the results show discrepancies between the predicted instantaneous turbine yaw performance and measurements.

1. Introduction

Small wind turbines are primarily used in off-grid applications, either as a stand-alone system or in a hybrid configuration, such as a solar-wind or diesel-wind system. They are defined by the IEC 61400.2 – 2013 standard as having a swept rotor area of less than 200 m², which corresponds to a power production of up to 50 kW and a rotor diameter of less than 16 m [1]. A common feature of small wind turbines is that they are located near the end user. In many cases, a small wind turbine is installed at a pre-existing site to fulfil a requirement for power, rather than at a site with an ideal wind profile, resulting in this class of turbine operating in a vast array of wind regimes. As such, unsteady and turbulent wind conditions, where both the wind speed and direction rapidly change, often simultaneously, present one of the greatest challenges to the control, design, and performance of small wind turbines [2, 3].

Small wind turbines utilise passive yaw systems, where the turbine is free to follow changes in wind direction. A passive yaw control is favoured over more complex sensor-based control systems due in part to the cost of both the wind measurement equipment and the yaw control system. Additionally, small wind turbines are subjected to significantly higher yaw rates than their larger counterparts, with measured yaw rates exceeding 160°s⁻¹ [4], providing complexity to alternative yaw control strategies. In a passive yaw system, the net restoring moment is the sum of the rotor moment, $M_R$, generated by variation between each of the blades performance, and the moment produced by the tail fin, $M_T$. This means that there is no controlled limit on yaw errors, $\theta$, or yaw rates, $\dot{\theta}$, other than the inertial characteristics of the turbine, $I_{yaw}$, and the net yaw moment. The resulting angular acceleration of the turbine, $\ddot{\theta}$, is given by equation (1):
\[ \frac{M_R - M_T}{l_{\text{yaw}}} = \dot{\theta} \] (1)

The use of a passive yaw control strategy, however, may potentially lead to significant gyroscopic loading, often being the most severe loading on the blade root and rotor shaft [5]. For a turbine operating with a yaw error, there can be significant losses in the turbine’s power generation due to the \( \cos^2 \theta \) [6, 7] relationship between power and turbine alignment. However, the turbulent environment adds additional complications, with high turbulence at low wind speeds leading to a potential increase in energy production [8].

There are a number of theoretical models to predict the performance for each of the components of a wind turbine. However, many of these models are used either in isolation or are derived from steady-state assumptions. Furthermore, the accuracy of these models has not been thoroughly evaluated for a turbine of this class when operating in unsteady flows, due principally to the lack of detailed experimental data. There have been limited studies on instrumented small wind turbines [9], and of these limited studies, furling turbines were used, whereas this study focuses on a fixed tail fin configuration. Here a comprehensive computational model of a small horizontal-axis wind turbine has been developed utilising current modelling theory, and the accuracy of this model has been evaluated against experimental data acquired from a 5 kW horizontal-axis wind turbine operating in an unsteady flow.

2. Turbine details and field measurements
The Small Wind Turbine Research Group at the University of Newcastle, Australia has access to an upwind, two-bladed, horizontal-axis, 5 kW Aerogensisis wind turbine, see figure 1. This turbine has been manufactured as a Class III turbine to the IEC 61400.2 – 2013 Standard and has a rotor diameter of 5 m, a cut-in speed of 3.5 ms\(^{-1}\), and a rated power production of 5 kW at 10.5 ms\(^{-1}\). The turbine operates optimally at a tip speed ratio of 8, which corresponds to a rotor speed of 320 rpm. The rotor drives an induction generator via a two stage speed-increasing gearbox which a gear ratio of 1:8, allowing for the required aerodynamic rotational speed of the blades to be matched with the synchronous speed of the generator. The test turbine’s generator is configured in a stand-alone self-excited configuration, as is typical for an off-grid turbine. Additionally, the turbine incorporates a Maximum Power Point Tracking (MPPT) control strategy, with the objective of maintaining the rotor at a pre-set tip speed ratio. Shaft speed control is achieved by a PI controller. The turbine is mounted on the top of a tapered 18 m octagonal monopole with a hinge at the halfway point of the tower to provide ground level access for maintenance and instrumentation. Full details of the turbine can be found at [10]. Figure 1 shows a picture of the turbine with labels indicating the location of the turbine’s data acquisition systems (excluding system 9) with details of each system given in table 1.

![Figure 1. The 5 kW Aerogenesis horizontal-axis wind turbine; data acquisition sensors as per table 1.](image-url)
Table 1. Data acquisition systems, showing acquisition sample rates.

| Part number | Acquisition system                          | Sample rate (Hz) |
|-------------|---------------------------------------------|------------------|
| 1           | Wind vane                                   | 5                |
| 2           | Cup anemometer                              | 5                |
| 3           | Optical turbine direction sensor            | 24               |
| 4           | Arduino based hub data logging system       | 500              |
| 5           | Twelve strain gauges on one of the blades   | 500              |
| 6           | Eight tower accelerometers                  | 500              |
| 7           | Tail fin moment sensor                      | 5                |
| 8           | Blade position sensor                       | 500              |
| 9           | Generator shaft encoder                     | 5                |
|             | Electrical Power                            | 5                |

In order to acquire detailed unsteady operating data, the turbine has been instrumented with a range of wireless and hard-wired data acquisition systems; ranging is sampling rate from 5 to 500 Hz. A central computer allows for overall control and common timestamping of each of the systems, allowing for alignment and post-processing to be undertaken. In total, nine separate acquisition systems have been fitted to the turbine (see table 1), with the majority of the systems been custom built using Arduino based technology, to meet the specific challenges of acquiring data on a rotating turbine at the acquisition speed necessary to capture unsteady phenomena. The process of instrumenting a turbine is far from trivial, and complete details of the measurement systems and challenges can be found in [11], however for the readers convenience, the acquisition systems directly relevant to this study will be summarised in the following section.

A wind vane and cup anemometer are located on opposing faces of the tower by a fixed three-meter boom, three metres below the rotor axis so that they operate outside the region of disturbed flow caused by the rotating blades. Wind shear corrections have been included to account for the difference between the anemometer and rotor plane as per IEC 61400.2 – 2013. The turbines direction is measured optically using a high-definition camera and post-processed using optical character recognition. This measurement technique was adopted due to the presence of strong electromagnetic interference at the site, which caused significant problems with magnetometer-based systems. Rotor position and rotational speed are determined via a reed switch, integrate into the Arduino based hub unit. This particular unit was set to acquire signals at 500 Hz providing excellent resolution to determine rotational speed. A shaft encoder in located on the end of the generator shaft to measure the rotational speed of the generator. Finally, the electrical power produced by the turbine is measured directly from the inverter.

The turbine is located in a highly turbulent position on campus at The University of Newcastle, Australia, making it ideally suited to measure the effects unsteady operating conditions on turbine performance. As with any built environment, the site has many surrounding structures, both manufactured and natural, which contribute to the localised wind turbulence. The test site has a measured turbulence intensities of 27% [12], which exceeds the required turbulence intensity of 18% stated in IEC 61400.2 – 2013 standard for a Class III turbine.

3. Computational model details
As with any dynamic system, a wind turbine is more than the sum of its individual parts, and as such, a complete dynamic model of the turbine has been developed incorporating the relevant theories for each of the individual components. This model is built and solved using Simulink and utilises a combination of user defined Matlab function, standard library elements, and the SimScape toolbox.
Alternative modelling packages such as Fatigue, Aerodynamics, Structures, and Turbulence (FAST) were considered, however as FAST does not include native models for the induction generator or MPPT capability, these models would have needed to be developed separately. The performance of the base model developed shows excellent agreement when benchmarked against a FAST model of the same wind turbine, using the same generator and control system model. Additionally, there are competing theories on tail fin modelling [4, 5, 13, 14] which have not been thoroughly evaluated against unsteady field measurements. As such, more control and future functionality of each model were obtained from developing it as a complete system.

As the theories covering the performance of each component of a turbine are lengthy in detail, they will not be reproduced here, and the reader is encouraged to review each theory separately. Alternatively, a thorough presentation of each model used can be found in [10]. As stated earlier, the performance of each component within the turbine relies on the interactions of the whole system; figure 2 provided a summary of the component interaction for the computational model.

![Figure 2](attachment:image.png)

**Figure 2.** The overall block diagram showing interactions between each of the subsystems of the computational model.

The computational model utilises Unsteady Blade Element Momentum theory to model the aerodynamics [15, 16], incorporating yaw misalignment [17], dynamic stall model [18], and skewed wake corrections [17]. Due to the length and complexity of the aerodynamics model, a MATLAB embedded function was used to integrate the aerodynamic model into the Simulink environment, rather than a block diagram representation. Additionally, this theory was used to determine the effective wake velocity ‘seen’ by the tail fin, due to the reduction in the free stream velocity, resulting from power extraction by the rotor. Unlike the classical Blade Element Theory, which iteratively...
solves for each blade element until equilibrium is reached, the Unsteady Blade Element Theory uses a time-marching approach with small time steps to allow the development of the time developed aerodynamic phenomena which may occur in turbulent wind conditions. A time step of 0.001 seconds was utilised for this simulation, with a sensitivity analysis performed to ensure an adequate aerodynamic time step, and to match the required minimum time step used by Simulink’s ordinarily differential equation solver. The aerodynamic torque produced by the rotor is input into a six-mass torsional model of the drivetrain. The modelling theory of gearboxes, particularly one of this size, is well understood, and the requisite theory can be found in numerous texts with a good review of the mathematical models provided by [19].

A torsional mass model combines the equations of rotational motion of each body within the system, and couples these together to form one final torsional inertia matrix describing the overall response of the drivetrain. By using the Simscape™ Driveline module of Simulink, the drivetrain may be modelled in a symbolic block diagram, significantly reducing the potential for modelling error, which may occur with higher order torsional models; a summary of this modelling technique can be found in [20]. The mass properties of the gears and driveshaft’s within the drivetrain were experimentally determined and can be found in [10]. The SimScape Power Systems model of Simulink has been used to model the self-excited induction generator. The parameters for this model were experimentally determined, and in addition to the fundamental generator parameters, the model also includes saturation characteristics and variations in resistance due to temperature effects [21]. In addition to the induction generator model, the existing control strategy of the turbine has been modelled, incorporating the fundamental behaviour of the inverter and proportional-integrator control system, which are utilised for MPPT. This model has not considered the structural degrees of freedom associated with the blades or tower.

The yaw performance of the turbine has been modelled as a combination of the delta wing tail fin, based on unsteady slender body theory [5] and the contributions from the rotor due to differences in the blades respective performance during the Azimuthal cycle of the rotor, and operating in yaw flow. As mention previously, this model will be used to assess the impact of various tail fin models, which will be undertaken in future work. For this study, only the unsteady slender body theory has been considered.

4. Simulations results and discussion
Steady-state benchmarking of the model shows excellent agreement between predictions and the nameplate performance. Figure 3 shows the simulation results at rated conditions of 10.5 ms⁻¹ wind speed at 0° yaw error. As can be seen, the turbine produces 5.03 kW at a shaft speed of 2520 rpm, whereas the rated steady-state performance of the turbine is 5 kW at a generator speed of 2566 rpm, resulting in a difference of just over 1% for generator speed. A maximum generator efficiency of 82% was obtained, which correlates well with induction generators of a similar class [22].
In addition to steady-state performance, the individual performance of the induction generator, drive train and control system models have been evaluated against experimental data. Figure 4 shows a representative sample of experimental data. In this particular simulation, measured rotor speed, from the reed switch on the hub unit, was fed into the model to remove the effects of the aerodynamic and yaw uncertainties, so only the drivetrain, control system, load, excitation and induction generator models were active. As can be seen in figure 4, there is an excellent correlation between measured and simulated electrical power, with an error of just under 10%, with the measured signal bounding the simulated power. The generator model is reasonably sensitive to the measured parameters, and the small overall error is most likely due to variations which occur during operation, and may include effects such as resistor temperature dependence and operational variations in components.

**Figure 3.** Steady-state benchmarking of the turbine model showing Electrical power; high-speed shaft speed and wind speed, simulation.

**Figure 4.** Comparison between the measured electrical power and the simulated electrical power.
In unsteady flows, however, the complete dynamic model has poor correlation with the measured results, as shown in figure 5. These results utilise site wind measurements to account for the previously mentioned site turbulence, and the dataset considered here is a representative sample of a larger data set consisting of many hours of operation. As can be seen, the model fails to capture the instantaneous yaw, shaft speed and generator output of the turbine suggesting the unsteady behaviour has a strong effect on the turbine’s overall performance.

![Figure 5](image)

**Figure 5.** Comparison of the dynamic model’s performance compared to experimental data. In Figure 5a), the blue cross indicates the measured turbine direction, the simulated turbine direction is the solid black line. The measured wind direction is the red dot. In all other series, the blue marker indicates measured data while the solid black line indicates simulated signals.

Experimental data was used as the initial conditions for the dynamic model. Also, the initial two seconds of simulated data were excluded to allow for time development of the aerodynamic code. During the time period between 2 to 5 seconds in figure 5, the turbine operates significantly out of the wind, with more than 40° yaw error. In contrast, the predicted yaw response is more damped, yielding approximately half this yaw error. The difference between predicted and measured results is significant when considering the $\cos^2\theta$ relationship between power and turbine direction; similar differences have been observed throughout this data set and others, where the turbine has been seen to yaw out of the wind by as much as 90°. Unfortunately, the existing modelling theory is unable to explain this measured phenomenon. One possible explanation for the disparity between predictions and measurements could be a loss of tail fin restoring moment due to the tail fin operating in the rotor’s wake, which has not be accounted for in the theory. Alternatively, a significantly greater
difference between the left and right side of the rotor plane during yawed flow may contribute to a larger net moment about the yaw axis. The complete models also over predict the shaft speed of the turbine, with a difference of approximately 13% between simulated and measured performance; this may result from the aerodynamics model performing more favourable during periods of higher yaw error, and producing a greater net torque. The simulated results also over predict the rotor acceleration when compared to measured results, which would indicate a more sudden predicted increase in torque once the yaw error reduces. This higher shaft speed leads to an over-prediction of the electrical power produced by the turbine.

When considering average turbine performance, there is a 13% variation between predicted mean shaft speed and measured shaft speed during the simulated period. However, as the control algorithm attempts to control the turbine via the cube of the input shaft speed, this results in a mean electrical power difference of more than 50% for the simulated signal.

5. Conclusions and recommendations
Turbulent site flow conditions present significant challenges to the operation of small wind turbines. In this study, a complete dynamic model of a 5 kW small horizontal-axis wind turbine was developed and compared to an instrumented turbine operating in turbulent flow, with a site turbulence intensity of 27%. Results show excellent agreement between steady-state simulations and the nameplate data with a generator speed difference of approximately 1%. Benchmarking of the drivetrain, generator, and control models also show excellent agreement with less than 10% difference between measured and simulated electrical power, when using measured rotor angular velocity. However, the overall model performance showed relatively poorer correlation, with significant variations in predicted turbine direction. Instantaneous differences were found between the drivetrain high-speeds shaft, resulting in a larger difference between measured and simulated electrical power.

From this study, it is clear that the predicted net yaw moment, due to the tail fin and rotor interactions, has not been accurately modelled for turbines operating in turbulent flows. Further investigation of the yaw moment during operation in turbulent flow is therefore required. Additionally, the instantaneous aerodynamic performance of the rotor during yaw in turbulent flow has not been adequately captured by existing models necessitating additional field-testing in turbulent flow to allow each model to be refined.

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