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Research Article

Parking Strategies for Vertical Axis Wind Turbines

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Strategies for parking a vertical axis wind turbine at storm load are considered. It is proposed that if a directly driven permanent magnet synchronous generator is used, an elegant choice is to short-circuit the generator at storm, since this makes the turbine efficiently damped. Nondamped braking is found to be especially problematic for the case of two blades where torsional oscillations may imply thrust force oscillations within a range of frequencies.

1. Introduction

It has become increasingly important to broaden the search for potential technologies for renewable energy conversion, as part of the joint effort to cut greenhouse gas emissions. Within wind power, the established and through incentives now commercially viable technology of horizontal axis wind turbines (HAWTs) has attracted most of the attention during the last decades. Modern HAWTs rely on a quite impressive list of moveable parts for its function, for example, individually pitchable blades and usually a gearbox-expensive and maintenance demanding matters [1], which in part explains the incentives dependence. Another concept for harnessing wind power, the vertical axis wind turbine (VAWT), has the inherent potential to reduce the number of moving parts as pitch and yaw mechanisms are not needed [2]. The generator of VAWTs may conveniently be placed on ground level, facilitating use of bulky direct drive generators which may be optimized for low cost rather than low weight. Patented already in the 30's by Darrieus [3], the VAWT concept was studied quite intently during the 70's and 80's [4, 5], but since then the major resources have been directed towards HAWTs. VAWT activities in the mean time have mostly been concentrated to small-scale turbines where the technology has indeed been successfully commercialized [6, 7]. Renewed interest in larger-scale turbines has arisen lately, especially in the context of offshore applications where the low center of gravity may be advantageous [4]. For the large-scale VAWT prototypes that have been demonstrated so far (i.e., those constructed in the 70's and 80's), the aerodynamic efficiency is of the order of 35–40%, which is somewhat lower than for HAWT. The challenge is, therefore, to reach a point where manufacturing and maintenance costs of VAWTs, as compared to HAWTs, are at least 20% lower per swept area, in order to render the concept viable.

In this study, a straight-bladed VAWT, also called H-rotor, is considered. A 200 kW prototype H-rotor with direct drive has recently been manufactured based on research from Uppsala University [8–10]; this turbine is currently one of the largest operating VAWTs in the world, see Figure 1. A turbine of this type is considered with respect to strategies for safe parking during extreme wind loads. Storm load is of increasing concern at growing scale since the wind speed increases with the height and faults may have more hazardous implications for large turbines than for small turbines closer to the ground.

2. The H-Rotor at Storm Load

For the estimates performed in this study, blade loads are calculated from measured 180° lift and drag data for the NACA0018 profile [11]. The considered turbine is essentially at rest (or very slowly revolving) at high wind speeds, suggesting that static airfoil data can be used, where the angle of attack is just the azimuth (possibly plus some offset). It

Figure 1: A 200 kW prototype H-rotor in Falkenberg, Sweden. The turbine is based on research from Uppsala University [8–10] and is currently one of the largest operating VAWTs in the world.

Table 1: Data for a 200 kW H-rotor with direct drive.

Radius (R)	13 m
Blade area (S)	58 m^2
Shaft torsional stiffness (k_s)	$3 \cdot 10^6$ Nm
Turbine moment of inertia (I_t)	$3 \cdot 10^5$ kg m^2
Generator moment of inertia (I_g)	$2.4 \cdot 10^3$ kg m^2
Inner resistance of generator (R_i)	0.065 Ω
Noload voltage per angular velocity (β)	140 Vs

is also assumed that the incident wind flow on each blade is unaffected by the presence of the other blades. This is reasonable from the point of view that the power absorbed by the turbine is small compared to the power contained in the incident wind; on the other hand, not realistic for some specific positions where a blade is directly behind another. However, for the rough estimates made here, this simple model is expected to suffice. In addition, no correction for finite aspect ratio is included.

The solidity of the turbine is here defined as of the total blade area S to the total swept area. Deciding the solidity for large VAWTs is generally a trade-off between aerodynamic efficiency and large forces on the blades and the tower (during operation and storm). It is, therefore, reasonable to keep the solidity constant when varying the blade count.

The thrust force $F(\theta)$ and torque $\tau(\theta)$ as functions of the azimuth θ are calculated according to the simplifying assumptions discussed above

$$\begin{aligned} F(\theta) &= \frac{S\rho v^2}{2N} \sum_i C_D\left(\theta + \frac{2\pi i}{N} + \alpha\right), \\ \tau(\theta) &= \frac{SR\rho v^2}{2N} \sum_i C_T\left(\theta + \frac{2\pi i}{N}\right), \end{aligned} \tag{1}$$

where $\rho = 1.2\,\text{kg/m}^3$ is the air density, v the wind speed, $i = 1, \ldots, N$ labels the blades, C_D is the drag coefficient, α is a possible offset pitch of the blades, R is the radius, N is the number of blades, and C_T is the tangential component of the lift and drag coefficient,

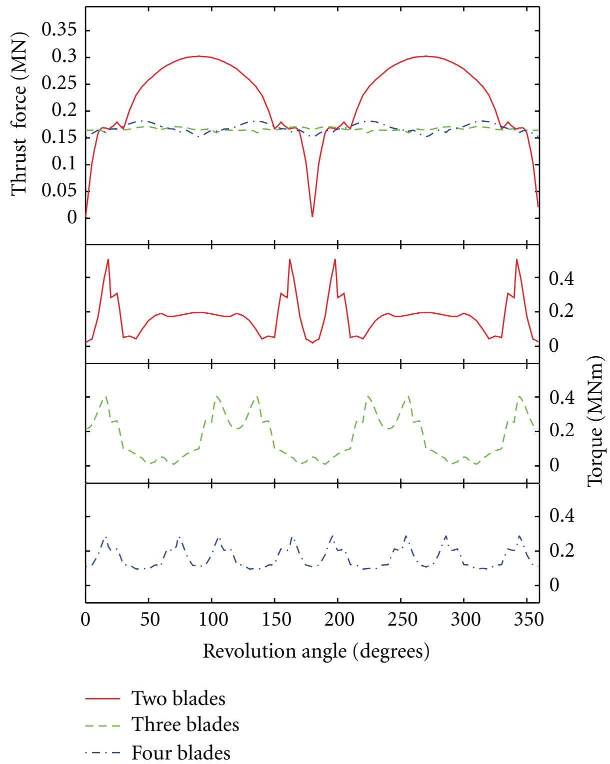

Figure 2: Thrust force and torque for the parked turbine at storm (wind speed 70 m/s). The solidity is constant in this comparison and the blade pitch is zero. The Reynolds number is $3.6 \cdot 10^6$ for three blades and correspondingly scaled according to the chord length for the other cases. The plots are based straightforwardly on measured data for the NACA0018 profile [11].

$$C_T(\theta) = C_L(\theta + \alpha)\sin(\theta) - C_D(\theta + \alpha)\cos(\theta). \tag{2}$$

Given the data in Table 1, which corresponds to the turbine described in [8, 9] (Figure 1), the thrust force and torque are shown in Figure 2. In addition to the reference turbine having three blades, the cases of two and four blades are also shown. The wind speed used here is $v = 70$ m/s, which is a realistic rated survival wind speed of this turbine. This implies a Reynolds number of about $4 \cdot 10^6$ for the three bladed turbine where the chord length is close to 1 m.

3. Parking Strategies

We will now discuss different scenarios for parking the turbine during storm load and also indicate implications of different blade counts in this context. An H-rotor with a generator at ground level is considered, the generator being connected to the turbine through a shaft.

3.1. Orientation and Tower Dimensions. It is evident from Figure 2 that the thrust force $F(\theta)$ is close to constant for three and four blades. As a consequence, no special or preferred orientation emerges with respect to the wind direction when parking the turbine. For two blades, on the other hand, the thrust force is highly angle dependent. It might, therefore, be tempting to orient the turbine at the thrust force minimum, to minimize the load on the tower. The wind direction may, however, change quickly at storm. We see from Figure 2 that a change of only 15° from the minimum gives rise to a thrust force comparable to the ones for three and four blades. By comparison, the European standard IEC 61400-1 states that the turbine, during extreme wind conditions, shall withstand wind direction changes of 30° in 5 seconds. It is therefore not possible to save on tower dimensions for two blades, as compared to three and more blades. As we will see, orienting the two-bladed turbine at the thrust force minimum might actually be a position that should be avoided.

3.2. Nondamped Braking. Consider initially a design where a nondamped brake is situated at ground level. With shaft torsional stiffness k_s and turbine moment of inertia I_t, the dynamic equation for the azimuth θ is

$$-k_s\theta + \tau(\theta) = I_t\ddot{\theta}. \tag{3}$$

For small oscillations around an equilibrium position θ_0, we may write the torque as $\tau(\theta) = \text{const} + k_\tau\theta$, where $k_\tau = \tau'(\theta_0)$. The solution to (3) is then $\theta = \theta_0 + A\sin(\omega t)$ where A is the amplitude and the frequency is

$$\omega = \sqrt{\frac{k_s - k_\tau}{I_t}}. \tag{4}$$

Since τ (and consequently k_τ) is proportional to the wind speed squared, the frequency of the oscillation is both wind speed and wind angle dependent. Consider now a turbine with two blades. We see from Figure 2 that the force on the tower is angle dependent, especially near the $\theta = 0$. When parking the turbine near $\theta = 0$ we note that an oscillating angle will imply oscillations in the tower force. Hence, when dimensioning the tower for the two-bladed turbine, one has to take into account the risk of resonance between the tower frequencies and the θ oscillations, where the frequency ω is both wind speed and wind angle dependent. For the three- and four-bladed turbine, on the other hand, the tower force is nearly constant for all angles, so tower frequency coupling is not an issue in this case.

Let alone these additional problems for the two-bladed turbine, we see that any turbine will suffer from angle oscillations with the nondamped brake design proposed above. It is probably wise to try to avoid any type of oscillations during storm load. One alternative might be to place the brake closer to the turbine, thereby shifting k_s significantly upwards. Then the frequency will be more stable, and the amplitudes will decrease. However, it is not obvious that these oscillations will be less dangerous since the angular acceleration will still be of the same magnitude, owing to the fact that the exciting forces are unchanged.

3.3. Damping by Short-Circuiting the Generator. The natural solution is to introduce damping. It is probably possible to construct a damped brake that maintains the direction, one that merely adds a $-k\dot{\theta}$ term to the left-hand side of (3). When a direct driven permanent magnet synchronous generator is used, a much more direct (and, consequently, safer) choice is at hand: simply short-circuit the generator. The generator is then heavily loaded, producing a torque that is proportional to the rotational speed. The effect is then roughly to switch the $-k_s\theta$ term for a $-h\dot{\theta}$ term. The direction is not maintained, but the damping is simple and efficient. More exactly, the system is described by the equations

$$\begin{aligned} -k_s(\theta - \varphi) + \tau(\theta) &= I_t\ddot{\theta}, \\ k_s(\theta - \varphi) - h_g\dot{\varphi} &= I_g\ddot{\varphi}, \end{aligned} \tag{5}$$

where θ is the turbine angle, φ is the generator angle, h_g is the torque per angular speed due to the short-circuited generator, and I_g is the moment of inertia of the generator. In what follows we use the data in Table 1 for the calculations. As for the generator data, these correspond to the generator described in [8]. To calculate h_g a simplified circuit model for the synchronous generator may be used, where the contribution from the inner reactance to the circuit impedance is neglected due to the slow revolution. The power absorbed in the generator is then $P = \tilde{\tau}\dot{\varphi} = 3E^2/R_i$, where $\tilde{\tau}$ is the torque directly on the generator, E the no-load voltage and R_i the inner resistance. The no-load voltage, is proportional to the angular speed, $E = \beta\dot{\varphi}$. As $\tilde{\tau} = h_g\dot{\varphi}$, the angular frequency dependence indeed cancels out and

$$h_g = \frac{3\beta^2}{R_i}. \tag{6}$$

Hence, for the low a.c. frequency range that we are considering here, h_g may be treated as a constant. For the current generator we get $h_g = 9 \cdot 10^5$ Nms.

The average value of the torque τ for one revolution is about 170 kNm (for two blades, slightly lower for three and four blades due the difference in Reynolds number), so if τ were constant the turbine would have made one revolution in 33 seconds. Depending on the amount of variation of τ along the revolution, this time will be stretched.

Solving the system of (5) numerically, we get a revolution time for the three-bladed turbine of about 90 seconds and an average power absorption of 12 kW, which is only slightly higher than the losses at nominal load, losses which the cooling system of the generator has to be designed to handle. As expected, for four blades the revolution time is lower, about 40 seconds, due to a more stable τ, and the power absorption is consequently doubled. More surprisingly, perhaps, the case of two blades shows a revolution time of about 50 seconds, owing to the fact that τ has less deep minima than for three-bladed case. For power absorption in the range 20–25 kW, which seems to be the case for two and four blades, cooling of the generator might have to be dimensioned more robustly.

Is it a problem that the turbine position is not maintained? For the two-bladed turbine, we note that the tower force will be almost twice that for the three-bladed one for a wide range of angular positions. The tower might as a consequence need to be dimensioned more robustly. Besides that, the slow revolution is probably only beneficial. There will be no torsional vibrations, and unfavourable positions where blade flutter could possibly occur are quickly left.

3.4. Pitch Angle Dependence. So far the pitch angle has been set to zero. From an aerodynamic point of view, a well-chosen nonzero pitch angle is typically advantageous for turbine performance [12]. For the parked turbine at storm, it is clear from (2) that the torque curves will change by the introduction of a pitch angle. In particular, the torque can become partly negative for some pitch angles of the three bladed turbine. This means that for a perfectly constant wind direction this would make the turbine come to rest (which is good at least from a power absorption perspective). However, as the wind direction is never perfectly constant, the turbine is in reality not expected to come to rest. Prescribing a model where the wind direction is constantly changing would consequently lower the revolution time (for all pitches), especially for the three bladed turbine.

Due to the coarse model used, and the strong dependence on regions in the Figure 2 where the separate blade torques add up to a small total torque, the revolution time results may not be very accurate. On the other hand, 33 seconds is a reasonable lower limit, since we know that τ is likely to be somewhat overestimated (no correction for aspect ratio, for example). Also, changing the pitch angle has little effect on the average torque. As a bottom line, then, with only small improvements on the generator cooling, this concept for parking the turbine at storm should be viable for any turbine configuration.

3.5. Damping of Torsional Vibrations. In the solutions to (5), it is seen that all kinds of excited initial conditions are effectively damped. If we neglect the contribution from I_g and consider τ as an external driving source, the damping ratio becomes

$$\zeta = \frac{\sqrt{k_s I_t}}{2h_g}. \tag{7}$$

For the numbers given above, we note that the damping ratio is $\zeta \sim 0.5$, that is, not far from critical damping. This means that all torsional vibrations are very effectively damped.

An even better damping could be achieved by, for example, making the shaft stiffer, which means increasing k_s. However, the shaft is already quite robustly dimensioned with respect to expected ultimate torque on the turbine, and it is not desirable to further increase the cost and weight of this component. The moment of inertia of the turbine I_t is also a quantity that is not desirable to increase. The generator torque per angular speed, h_g, could in principle be lowered by increasing the resistance in the circuit. However, this would at the same time decrease the revolution time and consequently raise the need for cooling due to higher power absorption. It is therefore reasonable to keep the current trade-off between damping and power absorption.

4. Conclusion

A straight-bladed VAWT (H-rotor) with a direct-driven generator has been considered, with respect to parking strategies at storm load. It is noted that nondamped braking will, for the two-bladed turbine, imply oscillations in the thrust force with a frequency depending on both the wind speed and the turbine azimuth, especially near the thrust force minimum. Possible coupling to the tower frequencies has to be taken into account when dimensioning the tower. An elegant way to avoid oscillations for a direct-driven turbine with a permanent magnet synchronous generator is to short-circuit the generator. The turbine then becomes heavily loaded with a torque proportional to the rotational speed, and any torsional vibrations are efficiently damped. It is found that the maximum rotational frequency leads to a power absorption that is of the same order of magnitude as the losses under nominal operation. Hence, cooling of the generator need only be slightly improved to render this concept for parking at storm possible for all turbine configurations.

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References

[1] L. Fingersh, M. Hand, and A. Laxson, "Wind turbine design cost and scaling model," Tech. Rep. NREL/TP-500-40566, National Renewable Energy Laboratory, Golden, Colo, USA, 2006.

[2] S. Eriksson, H. Bernhoff, and M. Leijon, "Evaluation of different turbine concepts for wind power," *Renewable and Sustainable Energy Reviews*, vol. 12, no. 5, pp. 1419–1434, 2008.

[3] G. Darrieus, "Turbine having its rotating shaft transverse to the ow of the current," US patent 1.835.018, December 1931.

[4] H. Sutherland, D. Berg, and T. Ashwill, "A retrospective of VAWT technology," Tech. Rep. SAND2012-0304, Sandia National Laboratories, Albuquerque, NM, USA, 2012.

[5] I. D. Mays and C. A. Morgan, "The 500 kW VAWT 850 demonstration project," in *Proceedings of the European Wind Energy Conference*, 1989.

[6] H. Riegler, "HAWT versus VAWT: small VAWTs find a clear niche," *Refocus*, vol. 4, no. 4, pp. 44–46, 2003.

[7] A. Bowen, A. Huskey, H. Link et al., "Small wind turbine testing results from the national renewable energy laboratory," in *Proceedings of the International Small Wind Conference*, Glasgow, Scotland, 2010, NREL/CP-500-48089.

[8] S. Eriksson, H. Bernhoff, and M. Leijon, "A 225kW direct driven PM generator adapted to a vertical axis wind turbine," *Advances in Power Electronics*, vol. 2011, Article ID 239061, 7 pages, 2011.

[9] J. Kjellin, S. Eriksson, and H. Bernhoff, "Electric control substitutingpitch control for large wind turbines," Submitted to *Renewable Energy*.

[10] J. Kjellin, F. Bülow, S. Eriksson, P. Deglaire, M. Leijon, and H. Bernhoff, "Power coefficient measurement on a 12 kW straight bladed vertical axis wind turbine," *Renewable Energy*, vol. 36, no. 11, pp. 3050–3053, 2011.

[11] R. E. Sheldahl and P. C. Klimas, "Aerodynamic characteristics of seven symmetrical airfoil sections through 180-degree angle of attack for use in aerodynamic analysis of vertical axis wind turbines," Tech. Rep. SAND80-2114, Sandia National Laboratories, Albuquerque, NM, USA, 1981.

[12] P. Klimas and M. Worstell, "Effects of blade preset pitch/offset on curved-blade darreius vertical-axis wind turbine performance," Tech. Rep. SAND81-1762, Sandia National Laboratories, Albuquerque, NM, USA, 1981.