An assessment of the effectiveness of individual pitch control on upscaled wind turbines

Z J Chen and K A Stol
Department of Mechanical Engineering, The University of Auckland, Private Bag 92019, Auckland Mail Centre, Auckland 1142, New Zealand
Email: zhenrong.chen@auckland.ac.nz

Abstract. The use of individual pitch control (IPC) based on loads transformed into non-rotating coordinates is explored on a range of wind turbines with ratings between 5MW and 15MW. Turbine models are generated using classical upscaling based on properties of the NREL 5MW reference wind turbine. The Ziegler-Nichols method is used with a low order linear model of each turbine to objectively tune a gain-scheduled, proportional-integral individual pitch controller. The performance of IPC is assessed by measuring reductions in blade and tower root damage equivalent loads from simulations at several wind speeds spanning Region 3. It is observed that the load reductions obtained with individual pitch control are maintained on upscaled turbines, with minimal impact on tower root loads, while actuator usage scales at a rate lower than expected with classical scaling.

1. Introduction
Over the past few years, one of the most visible trends in wind turbine design is the increasing size of the turbines. While the largest turbines in the present day have ratings of around 7-8MW [1], it is anticipated that even larger turbines may be designed in the future in order to provide increased generation capacity, maximise land usage, take advantage of potentially greater wind speeds and lower turbulence at higher altitudes.

1.1. Individual pitch control
Within research into active control methods for load reduction on multi-megawatt scale turbines, individual pitch control (IPC) has been shown to be capable of providing significant reductions in the blade (out-of-plane) and shaft fatigue loads [2]. There have been several approaches to the control aspect of IPC, ranging from simple proportional-integral (PI) controllers using tilt and yaw-axis loads to generate cyclic variations in blade pitch angles [2, 3], or local inflow measurements (angle of attack and relative velocity) to control differential pitch angles on each blade independently [3, 4]. More advanced control methods such as LQG [2, 5] have also been explored to deal with dynamic coupling, though their increased complexity and computational cost is a disadvantage, especially when simple classical control methods have been shown to provide good performance.

Current implementations of individual pitch control have mainly focused on the reduction of varying loads near the rotational (1P) frequency of the turbine, though other methods such as higher harmonic control are capable of reducing loads at higher frequencies [6]. However, this comes with increased controller complexity and may cause increased wear in the pitch actuator system due to higher actuation demands.
1.2. Previous studies on turbine upscaling and individual pitch control
At this point in time, no previous work on the use of IPC on a range of turbines with different ratings or on turbines with ratings over 5MW has been found. Earlier studies on turbine upscaling have focused on aspects such as the change in the cost of energy, with a 4% increase predicted over a range of turbines with ratings between 0.75 to 3MW as part of the WindPACT project [7, 8], while other studies have provided a brief look into the expected behaviour of extremely large (20MW) turbines [9] with standard collective pitch control, and found that the increased rotor inertia meant that that the rotor speed of the turbine was less sensitive to a wind gust compared to a smaller (5MW) turbine.

The scaling of the collective pitch control system and loads on turbines with ratings ranging from 5 to 20MW has also been investigated [10], and gain scaling rules proposed to ensure damping of the tower fore-aft mode was maintained above a specified level. Time domain simulations demonstrated that edgewise moments increased with scale due to higher blade mass, while reductions in rotor speed variation were observed with increasing scale in Region 2, potentially due to spatial averaging of the turbulent wind field, or the increase in rotor inertia. However, an increase in the rotor speed variation was observed in Region 3, which was attributed to the slower response of the pitch system.

1.3. Individual pitch control on upscaled turbines
In this paper, the use of IPC on large-scale turbines with ratings ranging from 5 to 15MW is investigated, to see if the benefits it offers to existing turbines can be realised on larger-scale turbines. A simulation based approach with turbine models generated from a baseline design using classical scaling provides the opportunity to see if there are any effects such as the excitation of higher modes, or from the impact of the spatially varying wind field, which may have an adverse impact on IPC performance on larger turbines.

2. Turbine upscaling methods
An important part of studies into turbine upscaling is the generation of the turbine models with higher ratings, usually from a baseline model. One method of obtaining upscaled turbine models is through the use of classical scaling, as used by Riziotis et al. [11] and Sieros et al.[10], whereby turbine dimensions are scaled up in proportion to a scaling factor, \( s \), while the tip speed of the turbine and aerodynamic configuration of the blades (e.g. airfoil profiles and coefficients) are assumed to be scale invariant. This leads to an increase in the rated power of the turbine which is proportional to \( s^2 \), while the natural frequencies and rotational speed of the turbine are scaled in proportion to \( s^{-1} \). Whilst this method of scaling is by far the simplest, a drawback is that weight induced stresses increase in direct proportion to \( s \), meaning that an upper limit to turbine size exists when material properties are held constant.

Analysis of real world trends in blade and nacelle mass by Sieros et al. [10] initially indicated that both values scaled with exponents below three, contrary to what was expected with classical scaling. However, the scaling exponents were closer to those expected with classical scaling if only large turbines (rotor diameters over 80m) were considered. Blade mass scaling trends corresponding to classical scaling were observed within each technology level (i.e. use of a particular blade material), indicating that the use of classical scaling is somewhat realistic if it is assumed that no improvements to blade technology are implemented within the scaling range of interest.

A slight variation to classical scaling was used by Peeringa et al. [9] as part of the Upwind project, in which a 20MW turbine blade was initially generated by classical scaling, before being optimised to maximise annual yield and power coefficient. While simple classical scaling assumes that the aerodynamic configuration of the blades remains scale invariant, this study accounted for the change in Reynolds number in the operating region of the upscaled blade, which led to a slightly different design with a reduced chord and twist distribution.
In order to account for the increase in weight induced stresses when classical scaling laws are used, Capponi et al. [12] proposed a scaling law in which stresses in the baseline and upscaled blade are kept constant, essentially by increasing the material thickness while keeping the external shape of the blade constant. However, a large increase in blade mass exceeding a cubic law was observed, which would lead to detrimental effects in Region 2 where optimal power coefficient tracking is required.

Apart from the use of scaling laws, another method of generating an upscaled turbine model is to define a completely new model, as has been done with a 10MW reference turbine model developed by DTU Wind Energy and Vestas Wind Systems [13]. While such a model would be more realistic compared to those generated via scaling laws, it requires time consuming consideration of many factors such as the aerodynamics, structural dynamics and aeroelasticity of the turbine, and it may make direct comparisons between turbines of different sizes difficult, due to potentially significant differences in turbine design and operational parameters.

Upon consideration of the turbine upscaling methods outlined above, classical scaling as used by Riziotis et al. [11] and Sieros et al. [10] was chosen as the method for generating upscaled turbine models due to its simplicity, ease of use, and the availability of existing turbine data. A summary of the several scaling exponents for quantities related to the turbine properties is provided in Table 1.

| Quantity          | Units | Scaling exponent, \( n \) |
|-------------------|-------|--------------------------|
| Length            | m     | 1                        |
| Mass              | kg    | 3                        |
| Area moment of inertia | m\(^4\) | 4                       |
| Mass moment of inertia | kg.m\(^2\) | 5                   |
| Power             | W     | 2                        |
| Frequency         | Hz, rad/s | −1                  |
| Damping ratio     | -     | 0                        |
| Maximum pitch rate| °/s, rad/s | −1                 |

### Table 1. Classical scaling exponents (quantities scaled in proportion to \( \varepsilon^n \)).

### 3. Turbine model

The baseline turbine model for this study, on which all other turbine models are based, was chosen to be the NREL 5MW reference turbine [14]. This is a theoretical, pitch-regulated, variable-speed turbine model, commonly used in research, with properties chosen to be representative of a typical utility-scale, multi-megawatt turbine. Some properties of the baseline 5MW turbine are provided in Table 2, along with upscaled values for the other turbine models used in this study.

| Property                  | 5      | 7.5     | 10     | 12.5   | 15     |
|---------------------------|--------|---------|--------|--------|--------|
| Rotor diameter [m]        | 126    | 154     | 178    | 199    | 218    |
| Hub height [m]            | 90     | 110     | 127    | 142    | 156    |
| Rated speed [rpm]         | 12.1   | 9.9     | 8.6    | 7.7    | 7.0    |

In this study, the HAWC2 nonlinear aeroelastic code [15-17], developed by DTU Wind Energy, is used for modelling and simulating wind turbine models in the time domain. The code uses a multibody formulation, where the structure of each body is modelled using Timoshenko beam elements, which includes bending and torsional degrees-of-freedom. Each wind turbine component (e.g. blades, tower and shaft) can either be modelled using a single or multiple bodies, which allow for large, geometrically nonlinear deflections to be accounted for. The aerodynamic model within HAWC2 is based on blade-element momentum (BEM) theory, with additional models and corrections included to account for the effects of dynamic inflow, tip loss and dynamic stall.
Alongside the HAWC2 code, the HAWCStab2 linear aeroelastic code [18] is also used for modelling and simulating wind turbines, but in the frequency domain. The aeroelastic model of the turbine in HAWCStab2 is similar to that implemented in HAWC2, with the code used in this study to generate linearised turbine models for controller design.

3.1. Baseline control system
The baseline control system for the 5MW reference turbine consists of a generator torque controller designed to maximise power capture in Region 2, and a gain-scheduled, proportional-integral (GSPI) collective pitch controller designed to regulate generator speed in Region 3. The generator torque curve was scaled proportional to \( s^3 \), with generator speeds marking transition regions between Regions 1 and 2, and Regions 2 and 3 scaled proportional to \( s^{-1} \). The PI gains for the collective-pitch control system were specified by Jonkman [19] (based on work by Hansen et al. [20]) to be

\[
K_p(\theta) = \frac{2I_{\text{Driverain}}\Omega_0\zeta_\phi\omega_n\sin \theta}{N_{\text{Gear}}} GK(\theta),
K_i(\theta) = \frac{I_{\text{Driverain}}\Omega_0\zeta_\phi^2 N_{\text{Gear}}} {\omega_n} GK(\theta)
\]

where \( I_{\text{Driverain}} \) is the drivetrain inertia about the low-speed shaft; \( \Omega_0 \), the rated rotational speed of the low-speed shaft; \( \omega_n \) and \( \zeta_\phi \), the closed-loop natural frequency and damping ratio of an idealised rotor-speed error model; \( N_{\text{Gear}} \), the gearbox ratio; \( \partial P/\partial \theta \), the sensitivity of aerodynamic power with respect to the collective pitch angle; and \( GK(\theta) \), a gain scheduling parameter designed to account for the change in the sensitivity of aerodynamic power with collective pitch angle (see [19] for values).

By scaling the quantities in Equation (1) with the scaling exponents provided in Table 1, it can be seen that \( K_p(\theta) \) is scaled in proportion to \( s^3 \), while \( K_i(\theta) \) is scale invariant.

3.2. Baseline blade pitch actuator
To account for the effects of actuator dynamics, each blade pitch actuator on the turbine has been modelled as an under-damped second-order system. For the 5MW turbine model, the natural frequency and damping ratio of the system are chosen to be 6.28 rad/s and 0.9 respectively, which are values close to those found in several studies with similarly rated turbines [8, 20, 21]. These values differ from those found in the report specifying the properties of the baseline turbine [14], as those values were chosen to minimise the effects of actuator dynamics to enable comparisons between different simulation codes.

Actuation limits were also imposed; with minimum and maximum blade pitch angles of 0° and 90° respectively, and a maximum pitch rate of 8°/s. For the upscaled turbine models, the actuator properties were scaled using the exponents provided in Table 1.

3.3. Individual pitch controller
An individual pitch controller based on work by Bossanyi et al. [22] was chosen as the baseline controller for this study. This controller uses the Coleman transformation, as shown in Equation (2), whereby loads in a rotating reference frame (the out-of-plane blade root bending moments, \( M_{1,2,3} \)) are transformed into loads in a non-rotating frame (\( M_{\text{cos}}, M_{\text{sin}} \)), which are equivalent to the variations about the mean loads in the rotating reference frame.

Blade pitch angles in the non-rotating reference frame (\( \theta_{\text{cos}}, \theta_{\text{sin}} \)) are generated through the use of a GSPI controller, with the goal of reducing the loads in the non-rotating frame to zero, before being transformed back into the rotating reference frame as shown in Equation (3). These pitch angles, \( \theta_{1,2,3} \), are differential pitch angles which are added to the collective pitch angle generated from the speed regulation loop to give the final blade pitch command for each blade. A second-order low pass filter with a cutoff at the 2.5P frequency and a notch filter with a cutoff at the 3P frequency were also included in each control loop to avoid high frequency pitch actuation and an increase in 3P loads.
\[
\begin{bmatrix}
M_{\cos} \\
M_{\sin}
\end{bmatrix} = \frac{2}{3} \begin{bmatrix}
\cos(\psi) & \cos(\psi + 2\pi/3) & \cos(\psi + 4\pi/3) \\
\sin(\psi) & \sin(\psi + 2\pi/3) & \sin(\psi + 4\pi/3)
\end{bmatrix}
\begin{bmatrix}
M_1 \\
M_2 \\
M_3
\end{bmatrix}
\] (2)

\[
\begin{bmatrix}
\theta_1 \\
\theta_2 \\
\theta_3
\end{bmatrix} = \begin{bmatrix}
\cos(\psi) & \sin(\psi) \\
\cos(\psi + 2\pi/3) & \sin(\psi + 2\pi/3) \\
\cos(\psi + 4\pi/3) & \sin(\psi + 4\pi/3)
\end{bmatrix}
\begin{bmatrix}
\theta_{\cos} \\
\theta_{\sin}
\end{bmatrix}
\] (3)

In an attempt to maintain objective controller tuning for all turbine models considered, the Ziegler-Nichols closed-loop method [23] was used to calculate controller gains. As part of this procedure, linear models for each turbine were first generated using HAWCStab2 at several wind speeds in Region 3. In order to simplify the controller tuning process, lower order linear models approximating the response of \( M_{\cos} \) and \( M_{\sin} \) to \( \theta_{\cos} \) and \( \theta_{\sin} \) as second-order systems were fitted to the original linear models.

![Bode plot of frequency response between \( \theta_{\cos,ref} \) and \( M_{\cos} \) for the NREL 5MW reference turbine at \( \bar{U} = 15 \text{m/s} \).](image1)

Figure 1. Bode plot of frequency response between \( \theta_{\cos,ref} \) and \( M_{\cos} \) for the NREL 5MW reference turbine at \( \bar{U} = 15 \text{m/s} \).

A comparison of the frequency response of \( M_{\cos} \) to \( \theta_{\cos,ref} \) between the original and low order linear models is provided in Figure 1, where it can be seen that the low-order model has a similar frequency response to the original linear model in the frequency region of interest (below the 3P frequency), though the response in the frequency region above 0.7-0.8Hz is not as well approximated. However, for the purposes of individual pitch control, only reductions in low frequency loads are of interest, and it was observed that the gains obtained using the Ziegler-Nichols method with the low-order linear models were capable of providing good IPC performance.

From the gains obtained through the Ziegler-Nichols method for the various turbines, it was seen that \( K_P \) and \( K_I \) scaled approximately in proportion to \( s^{-3} \) and \( s^{-4} \) respectively, as expected. Some variation in the gain

![Variation in \( K_P \) and \( K_I \) with collective pitch angle for a 5MW turbine (third order polynomial curves fitted to each series).](image2)

Figure 2. Variation in \( K_P \) and \( K_I \) with collective pitch angle for a 5MW turbine (third order polynomial curves fitted to each series).
values across Region 3 was also observed, as shown in Figure 2. As such, it was decided that a third-order polynomial gain-scheduling law based on the collective pitch angle (analogous to mean wind speed) would be implemented to account for this. As the collective pitch angle is a relatively slow varying parameter, this method of gain-scheduling is able to safely account for the variation in proportional and integral gains without a significant increase in the complexity of the controller.

Analysis of the closed-loop properties of the low-order linear model indicates that the controller tuning keeps the natural frequency and damping ratio of the dominant poles relatively constant throughout Region 3, as shown on the left in Figure 3 with the 5MW turbine used as an example. A comparison of the natural frequency and damping ratio of the dominant closed-loop poles between the various turbines as shown on the right in Figure 3 indicates that the natural frequency scales in proportion with $s^{-1.11}$ ($R^2 = 1.00$), while the damping ratio is virtually scale invariant, which compares favourably with the scaling rules set out in Table 1.

![Figure 3](image)

**Figure 3.** (L) Variation in natural frequency and damping ratio of closed-loop response in Region 3 with the low-order 5MW turbine model; (R) Variation in natural frequency and damping ratio of closed-loop response between various turbine sizes at $\bar{U} = 15\text{m/s}$.

4. Results and discussion

4.1. Deterministic wind simulations

Initially, the performance of the individual pitch controller was tested in a steady wind field with a power law wind shear profile with an exponent of 0.2, neglecting the effects of tower shadow. These tests were used to gauge the control demands imposed on the various turbines, and to ensure that the individual pitch controllers were working as expected.

On the left in Figure 4, it can be seen that the pitch angles resulting from the use of IPC are roughly equal on all turbines in a deterministic wind field. The slight difference may be due to the effects of wind shear, with the power law profile used giving the same change in wind speed between the bottom and top of the rotor for all turbines, though the distance over which this change occurs increases with turbine size. In general, it further demonstrates that the controller tuning is approximately scale invariant.

The plot on the right in Figure 4 shows the effect of IPC on the out-of-plane blade root bending moments of the 5MW turbine. It can be seen that the use of IPC is able to greatly reduce the 1P variations in the load signal, though the mean load is unaffected. Reductions in the RMS values of the out-of-plane blade root moment with IPC between 87-88% are observed for all turbines at a wind speed of 15m/s (reductions of 78-81% observed at wind speeds of 20 and 25m/s), showing that the steady state performance of the controller is scale invariant.
4.2. Stochastic wind simulations

Following the simulations in the deterministic wind field, a variety of simulations with a turbulent wind field were performed. Five runs were conducted for each turbine at mean hub-height wind speeds of 15, 20 and 25m/s, with turbulence intensities corresponding to Class A NTM values as specified in IEC 61400-1 (3rd Ed.) [24]. Each of the five simulation runs at a particular wind speed used a different, randomly generated turbulent wind field, measuring 230 × 230 × 180 m, with 64 points along the width and height of the box, and 16384 points along its length, which allowed for the same turbulence box to be used for all five turbines considered. The length of the wind field allowed for 10 minute simulation runs, excluding start-up time, which provided enough data to observe trends.

A comparison of the Kaimal spectrum and the spectra of several turbulent wind files with various length-wise resolutions (as shown in Figure 5) indicated that 16384 points was enough to capture the frequency content in the wind up to around 3-4Hz. This is sufficient for the 5MW turbine, and should be enough for the upscaled turbine models, as the natural frequencies decrease with scale.

As the same turbulence box is used for all turbine models, the 5MW turbine samples a smaller part of the box (approximately 35 points along the width and height) compared to the 15MW turbine. By examining the convergence of blade root out-of-plane damage equivalent loads with turbulent boxes of increasing resolution (width and height-wise) with the 5MW turbine, it was concluded that the distance between each grid point was small enough to capture the spatial variation in the wind field. It was assumed that this resolution would be sufficient for all upscaled turbine models as the grid spacing relative to turbine dimensions would decrease with increasing turbine size.

Figure 4. (L) Comparison of blade 1 pitch angles vs. rotor azimuth in a deterministic wind field with $\bar{U} = 15$ m/s; (R) Comparison of blade 1 out-of-plane root moment for a 5MW turbine with and without IPC in a deterministic wind field with $\bar{U} = 15$ m/s.

Figure 5. Comparison of normalised spectra for turbulent wind files of various length-wise resolutions and the Kaimal spectrum [25] ($\bar{U} = 15$ m/s).
4.2.1. Blade root loads. Figure 6 shows the change in the out-of-plane blade root bending moment when IPC is used. Reductions in the deviations about the mean bending moment can be observed with the use of IPC, which is also reflected in the power spectral density of the load signal, shown in Figure 7, where it can be seen that IPC is able to achieve significant reductions in 1P loads on all turbines with minimal excitation of higher frequency components.

An estimate of fatigue damage is obtained by computation of 1Hz damage equivalent loads (DELs) for the blade root out-of-plane bending moment with a rainflow algorithm and a Wohler-curve exponent, \(m\), of 10. Damage equivalent loads were blade-averaged to obtain a single value for each turbine from each simulation, with a performance index defined as the ratio between DELs with and without the use of IPC used to gauge the performance of IPC on the various turbines.

From Figure 8, it can be seen that the reductions in blade root out-of-plane DEL with IPC are fairly consistent with scale, with reductions of around 15-17% observed at a wind speed of 15m/s, and reductions of around 20-22% observed at 20 and 25m/s. The increased performance at higher mean wind speeds is most likely due to the pitch actuators occasionally saturating at their lower limit of 0° in simulations at a mean wind speed of 15m/s, while the higher collective blade pitch angle at mean wind speeds of 20 and 25m/s mean that actuator saturation is not a factor at these operating points.

The lack of variation in IPC performance between turbine ratings is due to the fact that the IPC control scheme is designed to reduce 1P loads. As the rated speed of the turbine decreases with scale, along with actuator rate limits and the natural frequency of the pitch actuator, there is no difference in the scaling of these quantities which may negatively impact IPC performance.

![Figure 6. Comparison of blade 1 out-of-plane root moment for a 5MW turbine with and without IPC in turbulent wind with \(\bar{U} = 15\text{m/s}\).](image)

![Figure 7. PSD of blade 1 out-of-plane root moment for a 5, 10 and 15MW turbines with and without IPC in turbulent wind with \(\bar{U} = 15\text{m/s}\).](image)
4.2.2. Tower root loads. A similar process to that used in the calculation of blade root DELs and performance indices was used for the tower root fore-aft and yaw moment DELs, but with $m = 3$.

From Figure 9, it can be seen that the effect of IPC on tower root fore-aft and yaw moment DEL is minimal for all turbines considered, with absolute changes in the mean performance index of less than 5% observed. As with blade root out-of-plane moment DELs, no difference in performance with turbine scale can be observed.

4.2.3. Actuator usage. In order to assess actuator usage, the RMS values for blade pitch rates were averaged over the five simulations (in turbulent wind) run at each wind speed for each turbine. These averaged RMS values can be seen on the left in Figure 10. It is observed that higher RMS pitch rates are required at higher wind speeds, potentially due to higher controller gains.
The normalised RMS pitch rates shown on the right in Figure 10 indicate that RMS actuator rates decrease at a greater rate than the $s^{-1}$ relationship predicted by classical scaling. This is most likely due to the same reason as that noted for results with a deterministic wind field, in that the wind shear profile is slightly different for each turbine, as the rotor size changes even though the difference in wind speed between the bottom and top of the rotor is identical between turbines.

4.2.4. Turbulent inflow effects. As the natural frequencies and rotational speed of the turbine decrease with increasing scale, they move closer to the part of the turbulent spectrum where the largest amount of energy is contained. However, as the rotational speed and natural frequencies of the turbine scale identically in this study, any adverse effects on structural loads are not observed in the results presented. In reality, these two quantities are not likely to scale identically, and care has to be taken to ensure that excitation of structural modes is avoided.

As the swept area of the turbine increases with scale, the spatial coherence of the turbulent inflow across the rotor plane decreases. A slight reduction in power variation in Region 2 due to spatial averaging is mentioned by Riziotis et al. [11], but there is also the potential for localised events to affect only a certain part of the rotor on larger turbines, causing increased blade/shaft loads due to asymmetric loading across the rotor plane.

With regards to the wind model used in this study, the Mann model [26, 27] has been used to generate the turbulent wind fields used for simulations, using parameters based on the Kaimal spectrum. For upscaled turbines, the von Karman spectrum could potentially be considered as it has been claimed to represent measured data well for altitudes above 150m [28], though anisotropic atmospheric conditions will need to be accounted for. Due to the sharper, higher peak of the von Karman spectrum, it is anticipated that load variations could potentially increase, which may lead to reduced IPC performance.

5. Conclusions
This study has investigated the use of IPC on a series of turbines with ratings ranging from 5 to 15MW, upscaled from the NREL 5MW reference turbine based on classical scaling rules. A GSPI individual pitch controller based on the Coleman transformation is tuned using the Ziegler-Nichols rules in conjunction with low-order models of the turbines, with gain scheduling included to account for the variation in gains within Region 3. Assessment of the natural frequency and damping ratio of the dominant poles shows that they scale according to classical scaling rules with the chosen set of controller gains.

Time domain simulations in a deterministic wind field demonstrated the consistency of controller performance with scale, before simulations in stochastic wind fields were used to assess the performance of IPC. It was found that reductions in out-of-plane blade root bending moment damage
equivalent loads were able to be maintained over the range of turbines studied, most likely due to the fact that the IPC scheme targeted reductions in 1P loads, with no difference in the scaling of rated rotor speed and pitch actuator properties. The use of IPC also had minimal impact on tower root fore-aft and yaw moment damage equivalent loads.

It was observed that RMS pitch actuation rates scaled at a lower rate than expected with classical scaling, potentially due to the difference in wind shear observed with rotors of different sizes. This may imply that load reductions with IPC are able to be achieved with slightly less actuator duty on larger scale turbines, which is advantageous as it would lead to improved actuator life and reduced actuator power usage relative to turbine rating.

6. Future work

There is still some potential for the IPC controller used in this study to be improved, as the Ziegler-Nichols method used for tuning is unlikely to have led to the best performing controller. Development of a set of tuning rules (e.g. to achieve certain specified closed-loop properties) would be a simple way of maintaining consistent performance throughout Region 3 on turbines of varying scale.

The scaling laws used in this study are extremely basic, and are not representative of true scaling trends, especially with such a large difference in the turbine sizes studied. As such, the use of other scaling laws could be investigated, to see if lower scaling exponents for mass and stiffness lead to changes in IPC performance.

There should also be further investigations into the true requirements for pitch actuation on upscaled turbines. It is not clear at this stage whether the pitch rates and actuator response used in this study are achievable in a real world situation, and what the true power requirements for actuation are, accounting for effects such as friction in the pitch bearings.

In this study, the wind conditions (turbulence intensity, power law exponent, mean wind speeds) used for simulations have been chosen to be identical between turbine models, so as to not favour a particular turbine scale. However, in reality, it is likely that the wind conditions experienced by upscaled turbines will be different, and it is worth exploring what these wind conditions are and how they affect loads on upscaled turbines, as well as whether they can be exploited to improve turbine performance.

References

[1] Enercon GmbH [Internet] 2014 E-126 / 7,580 kW [cited 10/03/2014], available from: http://www.enercon.de/en-en/66.htm.
[2] Bossanyi, E 2003 Individual blade pitch control for load reduction. Wind Energy 6 119-28.
[3] Kragh, K A, Hansen, M H and Henriksen, L C 2013 Sensor comparison study for load alleviating wind turbine pitch control. Wind Energy (Early View).
[4] Larsen, T J, Madsen, H A and Thomsen, K 2005 Active load reduction using individual pitch, based on local blade flow measurements. Wind Energy 8 67-80.
[5] Selvam, K 2007 Individual pitch control for large scale wind turbines - multivariable control approach. Energy Research Centre of the Netherlands, Report No. ECN-E-07-053.
[6] van Engelen, T G 2006 Design model and load reduction assessment for multi-rotational mode individual pitch control (higher harmonics control). Proc. EWEC, Athens, Greece.
[7] Griffin, D A 2001 WindPACT turbine design scaling studies technical area 1 - composite blades for 80- to 120-meter rotor. National Renewable Energy Laboratory, Report No. NREL/SR-500-29492.
[8] Malcolm, D J and Hansen, A C 2006 WindPACT turbine rotor design study. National Renewable Energy Laboratory, Report No. NREL/SR-500-32495.
[9] Peeringa, J, et al. 2011 Upwind 20MW wind turbine pre-design: blade design and control. Energy Research Centre of the Netherlands, Report No. ECN-E--11-017.
[10] Sieros, G, et al. 2012 Upscaling wind turbines: theoretical and practical aspects and their impact on the cost of energy. Wind Energy 15 3-17.
[11] Riziotis, V A, et al. 2012 Implications on loads by up-scaling towards 20 MW size. Proc. EWEA, Copenhagen, Denmark.
[12] Capponi, P C, et al. 2011 A non-linear upscaling approach for wind turbines blades based on stresses. Proc. EWEA, Brussels, Belgium.
[13] Bak, C, et al. 2012 Light Rotor: The 10-MW reference wind turbine. Proc. EWEA, Copenhagen, Denmark.
[14] Jonkman, J, et al. 2009 Definition of a 5-MW reference wind turbine for offshore system development. National Renewable Energy Laboratory, Report No. NREL/TP-500-38060.
[15] Larsen, T J and Hansen, A M 2012 How to HAWC2, the user's manual. DTU Wind Energy, Report No. Risø-R-1597(ver. 4-3)(EN).
[16] Larsen, T J, et al. 2013 Validation of the dynamic wake meander model for loads and power production in the Egmond aan Zee wind farm. Wind Energy 16 605-24.
[17] Kim, T, Hansen, A M and Branner, K 2013 Development of an anisotropic beam finite element for composite wind turbine blades in multibody system. Renew. Energ. 59 172-83.
[18] Hansen, M H 2011 Aeroelastic properties of backward swept blades. Proc. 49th AIAA Aerospace Sciences Meeting including the New Horizons Forum and Aerospace Exposition, Orlando, FL, USA.
[19] Jonkman, J M 2007 Dynamics modeling and loads analysis of an offshore floating wind turbine [Ph.D. Thesis], University of Colorado, Boulder.
[20] Hansen, M H, et al. 2005 Control design for a pitch-regulated, variable speed wind turbine. Risø National Laboratory, Report No. Risø-R-1500(EN).
[21] Larsen, T J and Hansen, A M 2006 Influence of blade pitch loads by large blade deflections and pitch actuator dynamics using the new aerelastic code HAWC2. Proc. EWEC, Athens, Greece.
[22] Bossanyi, E 2003 Wind turbine control for load reduction. Wind Energy 6 229-44.
[23] Ziegler, J G and Nichols, N B 1942 Optimum settings for automatic controllers. Trans. ASME 64 759-68.
[24] International Electrotechnical Commission 2005 Wind turbines – Part 1: Design requirements, IEC 61400-1, 3rd Ed.
[25] Burton, T, et al. 2011 Wind Energy Handbook. 2nd Ed. John Wiley & Sons, Ltd, Chichester, England.
[26] Mann, J 1994 The spatial structure of neutral atmospheric surface-layer turbulence. J. Fluid Mech. 273 141-68.
[27] Mann, J 1998 Wind field simulation. Prob. Engng. Mech. 13 269-82.
[28] Engineering Sciences Data Unit 1985 Characteristics of atmospheric turbulence near the ground. Part II: Single point data for strong winds (neutral atmosphere), ESDU 85020.