A control cost criterion for controller tuning of two- and three-bladed 20 MW offshore wind turbines

Fabian Anstock$^1$ and Vera Schorbach$^1$

$^1$Hamburg University of Applied Sciences, Berliner Tor 21, 20099 Hamburg, GE
E-mail: fabian.anstock@haw-hamburg.de

Abstract. When wind turbine concepts are modified, the difference in characteristic behavior and the impact on the cost of energy are of great interest. However, before comparing it is mandatory to re-adapt controller gains or weights to altered turbine dynamics in order to guarantee a still suitable overall system design. As an expenditure-based solution, the proposed control cost criterion could help to assess changes of turbine concepts better due to a clear and objective controller tuning. Conflicting design objectives are therefore assessed on their approximated influence on the cost of energy in order to result in the most cost-beneficial trade-off. These objectives are variations of component loads, actuator usage, and energy production. The criterion is applied to three- and two-bladed 20 MW offshore turbines with turbulent wind for full load operation, exemplarily. This provides insight into the benefits and limitations of low controller integral pitch gains and results in potential cost reductions for all turbines.

1. Introduction

Caused by the urgency of reducing the levelized costs of energy the general concept of large two-bladed wind turbines has gained a renewed interest in the offshore wind energy market. Compared to three-bladed turbines, omitting a blade offers various theoretical cost benefits in the turbine’s life cycle of manufacturing, erection, maintenance and decommissioning [1, 2]. However, it turns out to be difficult to compare two types of turbines possessing completely different aeroelastics and consequential attributes, such as dynamical loads, suitable blade structures, eigenfrequency interactions, and controllers [3]. The latter is the focus of the work at hand.

When comparing different concepts one initial difficulty is to choose a well-thought-out basis. If two wind plant controller concepts should be opposed, this basis is typically the same turbine. If the aim is a fair comparison of two- and three-bladed turbines, one could suggest to use the reciprocal approach and to investigate both turbines with equal controllers. But, if the turbine concept changes, the corresponding controller – even if it inherits the same controller structure – has to be re-adapted, to achieve an overall favorable system. Otherwise, this would likely lead to a meaningless comparison of a well-designed machine with a poorly tuned one, due to a change in the basic dynamics of the system. Therefore, it is vital to apply a tuning strategy which does enhance none of the turbine concepts more than the other to ensure a good comparability. Being one of the most important target values during turbine design processes, the levelized costs of energy (LCoE) will thus serve as a weighting to assess the quality of control performance.
2. The necessity of a control cost criterion for comparable wind turbine systems

A major issue which arises during the comparison of two- and three-bladed turbines is that a standard two-bladed, fixed hub, upwind, variable speed, horizontal axis wind turbine (HAWT) generally possesses higher loads – in quasi-static operating conditions – if it is compared on the basis of an equal power curve and a similar blade design [4]. Dynamic loads are expected to be increased even further due to less evenly distributed aerodynamic forces over the rotor plane, the periodic change of the rotor inertia about the yaw axis and the increased blade shadow caused by a mostly larger chord [3]. If no passive load mitigation strategy, such as teetering [5], is implemented, it is therefore mandatory to adapt the controller to the best possible extent to these increased dynamics within appropriate boundary conditions. Concluding, it could be stated that higher dynamic loads lead to a demand for increased controller interference. Yet, a fair strategy to re-adapt the controller gains is not straightforward.

Methods of classical controller tuning can be a powerful tool to recognize the system’s dynamical behavior and to tailor a suitable controller with a reasonable damping and safety margin for stability [6]. Another more intuitive hands-on approach is to increase the ability of a quick step response of a wind turbine to wind disturbances with the help of wind steps. However, a wind turbine is a highly non-linear system and the assessment of key design figures like damage equivalent loads (DELs), in combination with time varying three-dimensional wind speeds, increases the non-linearity even further, bringing these methods to their limits. While classical tunings still result in solid controller structures and gain factors, especially to guarantee a certain system stability, they are neither completely objective nor do they necessarily lead to the most cost beneficial overall system [7, 8]. A control cost criterion (CCC) that bases its weighting directly on actual wind plant expenditures to ensure a reasonable trade-off between conflicting effects and provides final gains as a result of an objective value, e.g., a minimum, would thus be appreciated. In the work of [9] controller gains were cost optimized after an initial classical controller tuning, leading to overall turbine cost reductions. Based on this approach a minimization function, which is a prominent method for optimization tasks, is used here.

\[ J_{\text{CCC}} = \Delta \text{costs due to Loads} + \Delta \text{costs due to Actions} \]  

Similar to [8], but with a focus on expenditures as in [9], the CCC rates the control performance. It results in equally reasonable controller gains for differing systems after a fixed controller structure has been implemented. This finally enables to compare two wind turbine concepts of different designs with equally suitable controller tunings, if the cost elements are assessed and rated correctly.

3. Description of the control cost criterion

The main idea of the CCC is to relate a variation in maximum or fatigue loads \( (L) \) of components to the respective mandatory changes in masses that are necessary to withstand these varying loads. This relationship could be obtained by utilizing, potentially nonlinear, coefficients \( (C) \) for an influence on material \( (IoM) \) of e.g. tower and blades [10]. The coefficients might be derived either analytically, e.g., by scaling the second moment of area [4, 11], or numerically. The resulting variations in masses, in turn, can be multiplied by each component’s cost-share \( (CS) \) of the reference turbine’s capital expenditures (CAPEX). The total approximated variation in CAPEX of these components could thus be expressed as the total sum of each fraction.

\[ \Delta \text{costs due to Loads} = \sum \left( \frac{L_i}{L_{i,\text{ref}}} - 1 \right) C_{IoM,i} C_{CS,i} \]  

with

\( i = M_y, \text{blades} \) (mean of DEL and maximum flapwise blade root moment),
\( M_y', \text{tower} \) (DEL of tower base bending moment in worst direction),
\( M_x, \text{hub} \) (DEL of hub torsion moment to assess drivetrain damage)

2
The components to be evaluated \((i)\) can be extended straightforwardly. For now, the list consists of the most expensive components, assessed by loads that are highly dependent on the control actions. The blade’s edgewise loads and the drivetrain bending loads are assumed to be mainly gravity driven by the fixed weight of blade and hub. For the control cost evaluation the drivetrain’s torsional load is highly relevant in partial load operation, when the torque controller is active. The damage equivalent loads (DELs) are calculated with a Wöhler exponent of 10 for the blades and 4 for steel structures like tower and drivetrain. Since the tower exhibits an axisymmetric structure, the tower base bending DELs are evaluated for every 7.5 deg and the direction of the highest value \((M_y')\) is selected. For this first approach, all \(C_{IoM}\) are rated with one, because higher stresses – or forces per area – in the components can roughly be compensated by increasing the area and thus the material proportionally. The \(C_{CS,i}\) are illustrated in the diagram of fig. 1 together with the cost split of CAPEX, operational expenditures (OPEX) and balance of plant, resulting in the costs of energy (CoE).

![Diagram showing the cost distribution](image)

**Figure 1.** Approximated relative cost distribution of CoE (left) and CAPEX (right) of the 20 MW INNWIND turbine based on 10 MW values from [12] and upscaled according to [13]

The hereby approximated cost differences due to changes in loads can then serve as a weighting of importance to assess (active or passive) load mitigation actions, which again have to be related to costs. In case of pitch movement, these costs could potentially be approximated either directly by the CAPEX, as proposed above, or by the influence on the operational expenditures \(\) caused by an increased maintenance effort [14], or by the impact on the power consumption of the actuator. The latter can directly be subtracted from the power production. The pitch activity itself is measured by the sum of all absolute pitch movements \(\int |\dot{\beta}|\), hereafter named actuator duty cycle (ADC).

\[
\Delta \text{costs due to pitch actions} = \left(\frac{ADC}{ADC_{ref}} - 1\right)(C_{IoM,Pit}C_{CS,Pit} + \frac{OPEX_{Pit}}{CAPEX})
\]

However, there is no method present today to actually calculate the wear of the pitch system, including bearings, based on any pitch motion characteristic. A higher pitch activity, as present in individual pitch control, is often linked to an increased bearing wear, while a larger pitch movement in-between seems to help to increase lubrication and thus reduces its wear [15]. Beside the bearings, it has been determined that the pitch actuators and hydraulic systems possess the highest offshore failure rates from all wind turbine components [14]. The replacement per occurrence might be comparably cheap, but it is nonetheless a critical component regarding its contribution to overall turbine downtime [16]. Even though a certain correlation between excessive actuator usage and wear is present, there is currently no available data of the influence from specific pitch activities on wear and thus on costs. For the first CCC evaluation, the ADC is directly related to the cost share with a factor of one (cf. [9]). Note, that the influence of the
pitch movement on the cost criterion is vital, because it suppresses an excessive use during the optimization.

Another option to reduce loads is the down regulation of the power output [17], while the energy produced is actually most precious for the wind farm. The missing income of the power difference is thus subtracted from the control costs and amplified by the fraction of CoE to CAPEX, since the remaining elements are calculated in CAPEX. The described complete control cost criterion can now be written as

\[ J_{CCC} = 1 + \sum \left( \frac{L_i}{L_{i,ref}} - 1 \right) C_{IoM,i}C_{CS,i} \left( \frac{\int P_{ref}}{P_{ref}} - 1 \right) \frac{\text{CoE}}{\text{CAPEX}} \]

\[ + \left( \frac{\text{ADC}}{\text{ADC}_{ref}} - 1 \right) \left( C_{IoM,Pit}C_{CS,Pit} + \text{OPEX}_{Pit} \right) \frac{\text{CAPEX}}{\text{CAPEX}} \].

For the first evaluation of the CCC, simulations in DNVGL’s Bladed 4.9 are carried out with an aero-servo-elastic model of the offshore jacket-based three-bladed INNWIND 20 MW reference wind turbine [18] and two preliminary structurally scaled two-bladed 20 MW versions with an equal tip speed (2B90) and with a higher tip speed (2B100). Both possess an equal absolute power curve and thus \( \sim 2\% \) longer blades, compared to the three-bladed turbine. The scaling of the aerodynamic surface of the blades is shown in [4]. The scaling of the structural properties uses the presumption of 50 \% higher flapwise loads per blade and gravity driven edgewise loads as described by [19], resulting in major blade mass reductions. The wind disturbances applied are at around 15 m/s during full-load operation. However, the same procedure could be utilized for all other wind speeds analogously. Two wind options are investigated and opposed:

(i) The normal turbulence model of the fatigue design load case DLC 1.2 stated in the IEC 61400-1 ed. 4 is applied to analyze results based on a realistic wind setting. While the standard of six seeds mean is used, the authors have to acknowledge that loads can be highly dependent on the turbulence seeds (cf. fig. 6 and fig. 7 in the appendix).

(ii) A single point history wind step disturbance is used to show the step response between initial controller gains and cost optimized gains for the three-bladed reference.

The controller utilized for the three-bladed turbine is the DTU reference controller [20] and the corresponding parameter values [18], which represent a classical tuning approach (\( \text{ref}, 3B \)). The changes of the two-bladed turbines’ initial (\( \text{ref}, 2B \)) proportional (\( K_P \)) and integral (\( K_I \)) pitch gains are derived with the method of [21] regarding the change in rotor speed (\( \Omega \)), tip speed (\( v_{tip} \)), and rotor inertia ratio. The resulting factors are listed in tab. 1.

| \( v_{tip} \) | \( l_{blade} \) | \( m_{blade} \) | \( \Omega_{2B,\Omega_{3B}} \) | \( K_P,2B \) | \( K_P,3B \) | \( K_I,2B \) | \( K_I,3B \) |
|---|---|---|---|---|---|---|---|
| 3B | 90 m/s | 122.14 m | 117.89 t | - | - | - | - |
| 2B90 | 90 m/s | 124.75 m | 96.89 t | 0.583 | 0.979 | 0.571 | - |
| 2B100 | 100 m/s | 124.66 m | 134.92 t | 0.803 | 1.089 | 0.708 | - |
4. Evaluation
Evaluating the control cost criterion $J_{CCC}$ of eq. (4) with varying factors on

$$f_{K_P} = \frac{K_{P,new}}{K_{P,ref}} \quad \text{and} \quad f_{K_I} = \frac{K_{I,new}}{K_{I,ref}}$$  \hspace{1cm} (5)

delivers some insight into issues of typical controller tuning methods regarding the achievement of minimum loads. A well-designed PI-pitch-controller of a wind turbine typically possesses a fast transient response of its control variable, the rotor speed. This behavior is mostly attributed to a relatively high integral gain fraction [21]. However, the three-dimensional plot of fig. 2 reveals that the highest cost benefit of the three-bladed 20 MW reference turbine at 15 m/s is achieved using small I-gains.

Figure 2. $J_{CCC}$ control costs for the three-bladed turbine and turbulent wind at 15 m/s.

To get a better understanding, a slice of fig. 2 (black dash-dotted line) for a P-Gain factor of 1.0 is plotted left in fig. 3 with one line for each CCC element. The two-dimensional plot shows that a softer regulation of the rotation speed has in fact a load mitigating effect (cf. [21]) and is apparently cost beneficial for all evaluated components. Nevertheless, a direct response of the pitch to changes in rotation speed from wind gusts is still given by the almost unchanged P-gain. Yet, the slower regulation after a gust helps to reduce tower and blade excitation [7], even without sacrificing mean energy production, fitting well to the results of [9, 21]. Nevertheless, this behavior leads to an increased variation of the rotor speed. Depicted by red dots in fig. 2 and by red stripes in fig. 3 are the values and regions, where the maximum rotation speed exceeds 8% of its rated value, which here is chosen as a safety margin to prevent rotor and generator overspeed during normal operation. Thus, the integral gain should be as low as possible without increasing the rotation speed bandwidth to a harming extent. Nonetheless, it might be necessary to adapt the overspeed limit with respect to the extreme load case DLC 1.3, since therein normal production without shutdown is required (see IEC 61400-1 ed. 4). For DLC 1.2, the feasible minimum of $J_{CCC} = 0.975$ is achieved with $f_{K_P} = 1.0$ and $f_{K_I} = 0.2$ (see fig. 2). It represents an approximated turbine cost reduction of about 2.5% if the wind velocity in the turbine’s operating life cycle is always 15 m/s and if all non-assessed components stay unaltered. For an overall controller tuning, this procedure needs to be applied analogously for all relevant wind
Figure 3. 2D slice from fig. 2 (left) and fig. 5 (right) for a P-gain factor of 1.0

Figure 4. Wind step response of pitch and rotation speed from the three-bladed turbine with reference and optimized PI-gains
tip speed of 90 m/s and an equal power curve (see [4] and [19]) results in the $J_{CCC}$-surface of fig. 5 and in the second 2D-plot of fig. 3. It is still the case that low integral gains lead to a larger bandwidth of rotation speed. However, the maximum rotation speed is in general higher compared to the three-bladed turbine. This issue presents a challenge for objectivity of the CCC, since there is potentially a low value for the constraint of the maximum rotation speed that can not be satisfied by the 2B90 turbine. For the chosen safety margin, the red striped area of the two-bladed turbine in fig. 3 is visibly larger. Furthermore, there is a tipping point visible at an I-gain factor of 0.2 in the 2D-plot where the pitch activity changes its monotonic decrease and increases rapidly. Interestingly, in contrast to the blade flapwise loads, lower I-gains do not lead to overall reduced tower fatigue damage as observed for the three-bladed turbine. This is a result out of a close proximity ($\sim 12.7\%$) between the 2P-frequency at rated rotation speed and both first tower bending eigenfrequencies, which had also been an issue in the work of [22] for the 10 MW INNWIND reference turbine. The higher bandwidth of the rotation speed for lower I-gains leads intermittently to a smaller gap of these frequencies during the simulation. Hence, the system is more dependent on the wind changes in order not to cause slow rotation speeds which would result in an increased resonance behavior of rotor 2P- and tower eigenfrequency, coincidentally. Concluding, it also makes sense that the seed convergence of the two-bladed turbine is much slower for runs with lower I-gains, which can be observed in fig. 6. Still, the divergence is reduced well after the mean values of three seeds and while results would be more robust for larger amounts of seeds (cf. fig. 7), the general characteristics seem to be represented fair enough.

![Figure 5. Control cost criterion $J_{CCC}$ with six seed mean values for turbulent wind series of 15 m/s for a 20 MW two-bladed turbine with 90 m/s tip speed.](image)

Especially in this case with conflicting design objectives, the used criterion offers the appreciated advantage that simultaneously increased and decreased loads of components can be weighted, respectively, in order to achieve a controller tuning, that results in an expedient and objective compromise. Again, the feasible minimum of $J_{CCC}$ is reached for a lower I-gain and an almost unchanged P-gain. However, the maximum approximated turbine cost reduction is just around 1%. To achieve an actually cost efficient two-bladed turbine the complete system has to be redesigned to improve its own dynamic interactions. On the one hand, this could be tried passively, by dampers and tower frequency modulation [22]. On the other hand, it
might help to increase the controller’s feasible close-loop bandwidth by another PI-structure for an improved rotor speed regulation [23], or to implement active tower damping, or to use an advanced, possibly model predictive, control approach. Nonetheless, the CCC might facilitate an objective tuning of other, potentially non PI based, control methods as well [8]. While a simultaneous improvement of turbine and controller might be most suitable, it is emphasized for further investigations to implement one change at a time to help tracing back load mitigating effects to their actual cause. The mere PI-gain variations of the original controller used in the paper at hand can not improve the tower loads vastly.

A preliminary comparison between the components’ loads of the studied turbines can be found in tab. 2. Note, that the 2B blades had originally been structurally scaled to withstand 50 % higher flapwise loads [19] and had been subsequently up-scaled by ∼2 % to match the same power curve due to a ∼4 % lower aerodynamic power coefficient [4]. Therefore, the 2B blade loads \( (L) \) are scaled down \( (L') \) to compare equivalent values for the same rotor diameter. With reference to [3] aerodynamically driven flapwise bending moments are scaled by the cube and gravity driven edgewise moments by the power of four of relative rotor length deviation. Interestingly, the flapwise fatigue and maximum loads are even lower than expected from previous, rated tip speed independent, quasi-static approximations of 50 % per blade, especially for the faster rotating machine 2B100. In contrast, the edgewise fatigue loads are slightly higher than assumed from the ratio of blade weights compared to the three-bladed turbine (see tab. 1). Yet, the ratio between both two-bladed turbines’ blade masses are close to the ratio of their edgewise

![Figure 6. Seed convergence of elements i from the two-bladed 2B90 turbine for the initial P-gain and an I-gain factor of 1 (left) and 0.01 (right).](image)

Table 2. Preliminary load comparison before and after applying the CCC; Loads are DELs except for the maximum blade flapwise load. \( L' \) marks loads of blades with an equal length.

|                              | 2B90/3B-1 | \((2B90/3B)_{opt}\)-1 | 2B100/3B-1 | \((2B100/3B)_{opt}\)-1 |
|------------------------------|-----------|-----------------------|------------|-----------------------|
| \( L'_{M_y,blade} \)         | +46.8 %   | +52.2 %               | +38.1 %    | +41.2 %               |
| \( L'_{M_y,blade,max} \)     | +50.0 %   | +51.2 %               | +49.4 %    | +48.4 %               |
| \( L'_{M_x,blade} \)         | -6.5 %    | -6.4 %                | +20.6 %    | +21.8 %               |
| \( L'_{M_y,tower} \)         | +178.1 %  | +212.8 %              | +74.0 %    | +89.4 %               |
| \( \hat{\Omega}_{\text{max}} \) | +2.9 %    | +0.2 %                | +0.8 %     | +0.7 %                |
| \( \int P \)                | +0.0 %    | +0.0 %                | -0.0 %     | +0.0 %                |
loads, fitting well to the suggestion of being mainly weight dependent. Vastly increased are the
tower base bending moments due to the unsuitable first tower eigenfrequency. The superiority
regarding tower loads of the faster rotating machine, 2B100, can mostly be explained by an
increased 2P-frequency in rated conditions and thus a larger safety gap to the resonance range
of the tower. The power generation in tab. 2 has automatically been close to equal for all runs
due to the general speed regulation concept of the used controller in above rated conditions. To
ensure equal power curves during all operational states – one main comparability factor [4] – the
criterion could be extended by an appropriately weighted quadratic function of the deviation
from the desired power curve.

5. Conclusions
In this work a criterion has been presented to tune gains of wind turbine controllers in an
objective manner in order to achieve a straightforward and explainable trade-off between
conflicting design objectives, such as component loads, actuator movement, maximum rotation
speed or power production. The main advantage is its direct relationship to actual financial
costs, enabling to assess control quality corresponding to the most important turbine design
parameter, the levelized costs of energy. Therefore, the control cost criterion can be of help to
tune controllers which are equally suitable for different wind turbine concepts inheriting diverging
system dynamics. Regarding objective comparability this method thus has an advantage
compared to classical controller tuning methods.

Nevertheless, it is important to bear in mind the limitations of this approach: the criterion
evaluates the approximated relative change in the costs of energy, which is a value that includes
the complete turbine’s life cycle until its decommissioning. Even though the criterion does
not attempt to predict the quantitative changes of these costs, there are various uncertainties
in the qualitative cost-based weighting between each evaluated element. A major issue is the
calculation of the pitch system wear, failure and operational costs.

On the premise that the chosen weights are correct, the control cost criterion’s minimum
evaluated at a 15 m/s wind bin results in turbine cost savings of around 2% and 1% of the
examined three- and both two-bladed 20 MW turbines, respectively. All turbines utilized
smaller integral pitch gains to reduce overall component loads while satisfying a constrained
maximum rotor speed. Eventually, the overspeed limit has to be re-adapted for extreme
turbulent conditions. For the slower rotating two-bladed turbine, load mitigation objectives
are more conflicting since a higher bandwidth of the rotor speed brings the 2P-frequency into
interference with the tower eigenfrequency. As a consequence the bending loads of the tower are
thus higher, instead of lower, for smaller integral gains. To obtain a suitable two-bladed turbine
design, this issue has to be addressed in the future design processes. Comparing the preliminary
blade flapwise loads to a three-bladed turbine it has been observed that dynamic fatigue and
maximum loads match well the quasi-static presumption of an increase of 50% per blade.

Finally it is stated, that the introduced method is a further and necessary step for comparing
two- and three-bladed turbines more objectively, in order to answer the still unsolved question
about which offshore turbine concept to prefer in the future.

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**Appendix**

![Figure 7](image.png)

**Figure 7.** Seed convergence of elements i from the three-bladed reference turbine for normal turbulence model at 15 m/s and 100 seeds.