A procedure to redesign a comparable blade structure of a two-bladed turbine based on a three-bladed reference

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Abstract. Two-bladed turbines might be an alternative for future offshore wind turbine installations due to the promising opportunity to be more economical compared to three-bladed turbines. Looking at the rotor blades, two-bladed turbines with a comparable design either rotate 22.5% faster or have up to 50% increased chord lengths. The latter result in a significantly higher second moment of area, which enables rotor mass savings and thus fewer costs while withstanding ~50% higher flapwise loads per blade. Unfortunately, such a design could also cause buckling issues. Increasing the design tip speed reduces stability issues but lowers the rotor mass savings. Consequently, the challenging task is to find an optimal compromise between preventing buckling and saving rotor mass, and thus to ensure a fair comparison of two- and three-bladed turbines’ blades. To overcome this challenge, a procedure for redesigning a comparable blade structure is introduced. The procedure is exemplarily demonstrated for large 20 MW offshore wind turbines. Compared to the three-bladed reference, the overall rotor mass reduction of the two-bladed turbine is 13.5% for the most beneficial compromise detected. Simultaneously, the blades showing good accordance with the reference concerning static stresses and buckling characteristics under their respective loads.

1. Introduction
The modern wind energy market is characterized by three-bladed turbines, which are constantly growing in rated power and rotor diameter [1]. At the same time, the competitiveness and commercial success of turbine manufacturers are dependent on the Levelized Cost of Energy (LCoE) of the developed and built machines. This offers the possibility to take account of two-bladed wind turbines as a promising alternative for the offshore wind energy market since they have fewer components to install, to maintain, and to decommission, and thus the advantage to decrease costs [2, 3, 4]. Furthermore, the growth trend enables to benefit from the two-bladed turbines’ improvement in blade stiffness. When increasing chord lengths and thicknesses, the blade material could be used more efficient [5], which would allow the usage of cheaper materials, reduces issues with the tip-tower-clearance, or enables even larger blades while maintaining the same design limits and materials.

In order to identify whether two-bladed turbines could be more economical than three-bladed turbines, a fair comparison of both turbine types is mandatory. During this comparison, the rotor blades are of particular importance, as they are the component that is influenced the most when changing the turbine concept, while they belong to the most expensive ones. For that
reason, this study investigates the rotor blades more in detail. But beforehand, the term of a *fair comparison* will be more clearly defined. Regarding this, it can be stated in simplified terms that both, the blades’ aerodynamics and the blades’ structures should show similar characteristics while applying as few changes as possible. One straightforward approach, utilized in the work at hand, is to achieve this by redesigning a two-bladed turbine’s blade based on a three-bladed reference. It has the advantage that every design step is very comprehensible and reproducible. In general, a distinction is made between the analytical aerodynamic redesign and the structural redesign.

For the analytical aerodynamic redesign of a turbine that has the same concept (e.g. upwind, variable speed, and fixed hub), there are two popular philosophies, which are mostly described in literature. On the one hand, the chord lengths and airfoil thicknesses of the two-bladed turbine’s blade can be increased by 50% while the rotor tip speed stays unaltered [6, 7, 8]. On the other hand, the same chord lengths and airfoil thicknesses can be maintained which leads to an increase of the two-bladed turbine’s tip speed by 22.5% [8, 9]. In addition to these two popular philosophies, there is also the possibility of a combination of both. From an analytical point of view, in all philosophies chord lengths and tip speed influence each other. In other words, defining the two-bladed turbine’s blade tip speed results in the chord scaling factor. In any case, the two-bladed turbine’s power coefficient $C_p$ will be under that of the three-bladed reference [10, 11]. Consequently, all loads and dynamics have always to be associated with a loss of energy. One way to get around this issue is to make an aerodynamic comparison based on the same absolute power curve instead of maintaining the same rotor diameter [11]. To achieve this, the blades have to be designed for an equal aerodynamic $C_p$ utilizing equal airfoils, equal relative chord distributions, and the same local angle of attacks. The blade length increases slightly by $\sim 2\%$. The approach is explained more in detail in [11].

Considering the structure of a redesigned blade, the analytical aerodynamic redesign philosophies, mentioned before, defines the outer geometry of the blade, reducing the structural design parameters to a manageable extent. The focus of the structural redesign is thus to adapt the material thicknesses. For a fair comparison, an adaption is obligatory since also the blade loads are different [12]. According to the authors’ view, the usage of equal strength and stability limits would be reasonable so that the redesigned two-bladed turbine’s blade is neither over- nor under-designed. A positive side effect is that the actual design of the three-blade reference is also of less importance because the redesigned blade will be equally over- or under-engineered.

Summarizing the criteria for a *fair aerodynamic and fair structural comparison*, both turbine types should possess:

(I) the same absolute power curve,

(II) an equal aerodynamic $C_p$ design, which utilizes equal airfoils, equal relative chord distributions, and the same local angle of attacks, and

(III) equal structural strength and stability limits.

Since the analytical aerodynamic redesign procedure including (I) and (II) is clear [11], this study focus on the structural part (III). During a structural redesign, various aspects could be considered, e.g. the stress distribution along the blade span and the buckling characteristics. Overall, it leads to a multi-factorial and difficult task. This challenge of comparing the blade structure of two- and three-bladed turbines is explained more in detail in Section 2. Subsequently, this paper will present an opportunity to overcome this challenge by explaining a procedure to redesign a comparable blade structure and demonstrates the procedure for one specific wind turbine.
2. The challenge of comparing the blade structure of two- and three-bladed turbines

The basis of both, the aerodynamic and structural redesign is to presume that the static flapwise loading of the two-bladed turbine’s blade is $\sim 50\%$ higher compared to its three-bladed reference because the two-bladed turbine’s rotor has to withstand the same wind loads as the three-bladed reference. This fact is independent of the philosophy according to which the blade is to be redesigned (see Section 1). The increase of the flapwise loading is supposed to be constant over the whole blade length as shown in [11]. It has also been observed in dynamic simulations of DLC 1.2 [12]. In contrast, the edgewise loading of both blades is assumed to be almost completely driven by gravitational forces [12, 13]. Thus, the edgewise loads of the two-bladed turbine’s blade will be approximated directly from the change in blade mass.

The boundary conditions previously mentioned, are the basis of the so-called “method of progressive structural scaling” [5]. This approach describes a simplified way to perform a structural blade redesign based on the same strength by adapting the composite material thicknesses. As stated in (III), the usage of equal strength limits is one aspect for a fair comparison of two- and three-bladed turbines’ blades. After applying this method, both the three-bladed reference blade and the redesigned two-bladed turbine’s blade will feature approximately equal material stresses at the different airfoil positions (e.g. at the spar caps, the shell structure, the reinforcements, and the shear webs) along the whole blade span. In principle, the progressive structural scaling can be performed for any possible combination of increased tip speed and chord lengths, calculated out of the analytical aerodynamic redesign (see Section 1).

In this context, it is generally more beneficial for the blade structure to increase the chord lengths and airfoil thicknesses instead of the tip speed during the redesign due to an advantage in blade stiffness since the second moment of area is rapidly growing the longer the chord lengths. In contrast, the influence of adapting the composite material thicknesses, which represents the wall thicknesses of the airfoil, on the second moment of area is relatively small. However, the choice of the design tip speed and the corresponding chord lengths has a directly related impact on the structural stability of the redesigned two-bladed turbine’s blade.

Following this, on the one hand, maintaining the tip speed but increasing the chord lengths by 50\% [6, 7, 8], leads to a lighter rotor and thus fewer costs [6]. Due to the previously mentioned advantage in blade stiffness, the blade structure could utilize thinner composite material thicknesses [5] while withstanding $\sim 50\%$ higher flapwise loads. Nevertheless, increasing the chord lengths and decreasing the material thicknesses at the same time could cause stability issues, e.g. buckling. On the other hand, increasing the tip speed by 22.5\% but maintaining the same chord lengths when redesigning a three-bladed turbine into a two-bladed one [8, 9], leads to larger wall thicknesses to withstand the $\sim 50\%$ higher flapwise loads. The structural benefits given by the second moment of area are significantly lower, compared to the previous option. Furthermore, the rotor mass would be equal to that of the three-bladed reference [6], while the buckling stability might be higher. In both cases, the redesigned blades will present approximately equal material stresses compared to the three-bladed reference, based on the scaling approach of the progressive structural scaling [5]. But as explained, depending on the design tip speed, the buckling behavior as well as the predicted blade mass, are different. So, there is a conflict of interest between preventing buckling and saving rotor mass. Thus, a challenging task would be to find a good compromise between both.

With regard to the usage of equal stability limits as defined in (III), a solution would be to identify the one combination of increased tip speed and chord lengths, which result in the same buckling stability as that of the three-bladed reference. This specific combination can be called “break-even-point” since, from a structural perspective, it might be a good compromise between preventing buckling and saving rotor mass while enabling a fair comparison. In the following Section 3, a procedure to detect such a fair comparable two-bladed turbine’s blade structure, the “break-even-point”, respectively, is described.
3. Methodology to detect a comparable two-bladed turbine’s blade structure

The procedure to detect a comparable blade structure of a two-bladed turbine’s blade is basically a combination of a fair aerodynamic and a fair structural redesign taking account of the defined aspects (I) to (III). For the aerodynamic redesign, one of the two popular philosophies or a combination of both is used, as explained in Section 1. The structural redesign will be based on the “method of progressive structural scaling” [5]. After the redesign has been performed, the strength and stability characteristics, as well as the masses, are evaluated.

In detail, the procedure to detect the “break-even-point” can be described in five steps. Note, that the procedure is not a structural optimization per se. It is only reasonable in the context of a comparison of two- and three-bladed turbine’s blades.

(1) **Initial tip speed:** Choose an initial tip speed of the two-bladed turbine’s blade for which the procedure should be applied.

(2) **Aerodynamic redesign:** Convert a three-bladed reference turbine into a two-bladed one by adapting the chord lengths and blade twist in correlation to the chosen tip speed. In all cases, the same relative chord layout, airfoil shapes and positions are utilized [11].

(3) **Structural redesign:** Adapt the material thicknesses of the different blade parts (spar caps, shell structure, reinforcements, and shear webs) according to the method of progressive structural scaling [5] utilizing two internal iteration loops for simplicity.

(4) **Evaluation of the structural characteristics:** Analyze the stress distributions along the blade span, the buckling load factor, and the blade weight by use of FEAs. Compare the results to corresponding analyses of the three-bladed reference.

(5) **Repeat procedure:** The procedure can be performed as an iterative process by stepwise approximation to reach nearly the same buckling load factor as the three-bladed reference.

It can therefore be summarized that the design tip speed has to be increased when the buckling resistance is smaller than the reference value, or vice versa, until the buckling stability is close enough to the reference value within a suitable tolerance. For this purpose, the redesigned two-bladed turbines’ blades are analyzed with finite element analyses (FEA). The program ANSYS® Mechanical APDL (Ansys Parametric Design Language) is used for the analyses. To represent a reasonable load situation initially for the three-bladed reference, aeroelastic load simulations of the DLC 1.2 according to IEC 61400-1 Ed. 4 [14] are performed in the software “Bladed” from DNV GL®. Using wind bins of 2 m/s from cut-in to cut-out wind speed, six seeds, and three different yaw angles (-8°, 0°, +8°), results overall in 198 simulations. Ultimately, the maximum flapwise loads of DLC 1.2 are implemented for the three-bladed turbine’s reference blade. Remember, the flapwise loads of the two-bladed turbines’ blades are 50% higher, according to the assumption described in Section 2. Note, that the maximum flapwise loads of DLC 1.2 are not automatically the turbine’s overall extreme loads. Nevertheless, they have been selected here because they are very representative.

In this paper, the blades of large 20 MW two-bladed turbines are exemplarily investigated. The three-bladed reference turbine will be the INNWIND 20 MW turbine [15]. Its model (aeroelastic and FEA) is based on publicly available data [15, 16, 17]. The turbine has a tip speed of 90 m/s and a rotor diameter of 252.2 m. For the sake of simplicity, the aerodynamic redesign of step (2) will be performed using the analytically calculated chord lengths without additional chord iterations [11]. Furthermore, blades of the same lengths are investigated, compared to the three-bladed reference. After the “break-even-point” has been identified, a blade extension (∼2%) according to classical cubic scaling laws [10, 18] is realized as defined in (I). This finally allows a fair comparison of aerodynamics, loads, and costs.

A demonstration of the introduced procedure is presented in Section 4. A set of five design tip speeds for the two-bladed turbine will be evaluated to show the increase of the buckling
Table 1. Tip speeds and chord factors of the redesigned two-bladed turbines’ blades, compared to the three-bladed reference.

| blade name            | tip speed | chord factor | redesign philosophy (see Section 1) |
|-----------------------|-----------|--------------|-------------------------------------|
| (i) 3B reference blade| 90 m/s    | 1.0          | -                                   |
| (ii) 2B90 blade       | 90 m/s    | 1.5          | only 50 % higher chord lengths     |
| (iii) 2B95 blade      | 95 m/s    | ≈ 1.35       | combination of both                |
| (iv) 2B100 blade      | 100 m/s   | ≈ 1.22       | combination of both                |
| (v) 2B105 blade       | 105 m/s   | ≈ 1.1        | combination of both                |
| (vi) 2B110 blade      | 110.25 m/s| 1.0          | only 22.5 % increased tip speed    |

resistance with the increase in tip speed, compared to the three-bladed reference. In this way, an appropriate overview and understanding of the performance of the introduced procedure is given. Table 1 presents an overview of all considered designs including the three-bladed reference.

The following results include first the static stress distributions of the different blades along their blade spans. They are shown only for one airfoil position, namely the spar caps at the suction side, exemplary. All other airfoil positions were also evaluated but will be not presented here since the static stress distributions are not the main focus of the paper. Second, the results include the buckling load factors and the masses, all compared to the three-bladed reference blade. The way to detect a fair comparable blade structure, the “break-even-point” blade, respectively, is shown graphically. In addition, the fatigue damages of the redesigned “break-even-point” blade and the reference are compared, although the progressive structural scaling does take account of them.

4. Results

After the newly presented procedure has been performed for the two-bladed turbines’ blades with different design tip speeds (see Table 1) initially the maximum nodal stresses were examined at predefined stations along the blade spans by taking account of potential modeling uncertainties. The stations are neither located at the load application positions nor at the locations of material changes. Since the chord lengths of all blades under consideration are different, the same relative chord positions have been examined. Figure 1 shows exemplarily the stress distributions along the blade span of all considered blades in the middle of the spar caps at the suction side.

Each graph represents one two-bladed turbine’s blade (ii) to (vi) from Table 1 in different colors, whose material stresses can be compared to that of the three-bladed reference (i) marked in red (see Figure 1). When looking e.g. at the 2B90 blade (ii), represented by the green line in Figure 1, the average deviation of the material stresses compared to the three-bladed reference (i) is only ~2 %. The average deviation of the other blades (iii) to (vi) is even smaller, the higher the designing tip speed. Note, that the progressive structural scaling (see procedure step 3) was performed here with only two internal iteration loops. More iteration loops would decrease the average deviation of all blades again even more. Overall, it can be said that all two-bladed turbines’ blades (ii) to (vi) have almost the same stress distribution (same material stresses) for 50 % higher flapwise loads compared to the three-bladed reference (i), which corresponds to the statements in Section 2. Furthermore, this fact applies not only to the stress distributions at the suction side spar caps, shown here in Figure 1, but also to all other airfoil parts [5].

Although the material stresses are almost equal, the masses of the redesigned two-bladed turbines’ blades are different as presumed. All weights are calculated directly from the FEA models. Non-structural masses like e.g. additional resin, adhesive, and coating, are not considered. It is assumed that they can be scaled linearly to the blade chord afterwards.
Therefore, the mass of the three-bladed reference blade (i) out of the FEA model is 106.8 t. In comparison, on the one hand, the lowest weight of a redesigned two-bladed turbine’s blade is given by the 2B90 blade (ii) with 104.9 t. Note, this blade has 50 % higher chord lengths and thus the most advantage in blade stiffness by the highest second moment of area (of all redesigned blades). The weight is slightly under that of the three-bladed reference (i), which results in rotor mass savings of $\sim 33\%$ [5]. On the other hand, the highest weight of a redesigned two-bladed turbine is given by the 2B110 blade (vi) with 159.9 t. This turbine has the same chord lengths as the three-bladed reference (i). As a consequence, it does not benefit as much from structural advantages, with the results that the rotor weight is almost equal compared to the reference (i). Besides, the weights of the other redesigned two-bladed turbines’ blades (iii) to (v) are between the minimum and maximum weight of the blade (ii) and (vi).

| blade name        | blade weight | buckling load factor | buckling position                    |
|-------------------|--------------|----------------------|--------------------------------------|
| (i) 3B ref blade  | 106.8 t      | 1.16                 | 23.1 % (spar cap at suction side)    |
| (ii) 2B90 blade   | 104.9 t      | 0.23                 | 22.5 % (rear shell at suction side)  |
| (iii) 2B95 blade  | 115.5 t      | 0.65                 | 42.6 % (front shell at suction side) |
| (iv) 2B100 blade  | 128.2 t      | 1.07                 | 38.2 % (front shell at suction side) |
| (v) 2B105 blade   | 142.1 t      | 1.56                 | 23.1 % (spar cap at suction side)    |
| (vi) 2B110 blade  | 159.9 t      | 1.99                 | 23.1 % (spar cap at suction side)    |
Focusing on the buckling characteristics, they also differ as presumed. A redesigned two-bladed turbine’s blade should use an equal stability limit compared to the three-bladed reference (i), according to (III). Thus, it should possess an approximately equal buckling resistance, simplified here by the buckling load factor, that indicates to which extent the applied loads have to be increased or decreased that the respective buckling shape occur. Table 2 compares the calculated buckling load factors with the masses described before, and further addresses the buckling positions. The corresponding first buckling shapes are shown in Figure 2.

The buckling load factor of the three-bladed reference blade (i) is 1.16 with the maximum deformation at the spar caps at the suction side (see Figure 2 (i)). Looking at the redesigned two-bladed turbines’ blades, the lowest buckling load factor is given by the 2B90 blade (ii) as expected. The factor is 0.23 with the maximum deformation at the rear shell structure at the suction side (see Figure 2 (ii)). The buckling factors increase together with the design tip speeds (see Table 2). The positions of the maximum deformations differs depending on the thickness changes given by the progressive structural scaling (see procedure step 3) [5]. Overall, the different buckling positions are in good agreement with the change in wall thicknesses. In line with this, the buckling position and shape of the 2B105 blade (v) is relatively similar to that of the three-bladed reference (i) (see Figure 2 (i) and (vi)), due to an almost equal ratio of material thicknesses and chord lengths at the spar caps. However, the highest buckling load factor of 1.99 occurs for the 2B110 blade (vi) as expected.

After describing the results from the strength and stability analyses in detail, the buckling load factors and masses of the two-bladed turbines’ blades are summarized graphically. Figure 3 (a) presents the absolute values as described before. For better handling and easier interpretation, Figure 3 (b) shows the buckling load factors and masses normalized to the values of the three-bladed reference (i). Note, that all blades possess almost equal strength characteristics as shown in Figure 1. Hence, a factor representing the material stresses is added.
For a fair comparison, the redesigned blade should possess equal structural strength and stability limits according to (III). The “break-even-point” is the one combination of increased tip speed and chord lengths that enables such a fair comparison, as mentioned in Section 2. It can thus be detected in Figure 3 (b) at the crossing of the material stress factor (gray line) and the buckling load factor (orange line) at a normalized value of 1. At this point both, strength and stability are equal to the three-bladed reference. For the investigations of the related 20 MW turbines done here, the tip speed at the “break-even-point” results in approximately 101 m/s (see vertical red line). The appropriate blade mass is 130.8 t, given by a mass factor of 1.225 (see red dotted lines). The authors like to remember, that these results correspond to the simplification of using the analytical chord lengths during the aerodynamic redesign (see procedure step (2) and Section 3). However, in the case of the 20 MW turbines considered here, the iterative design process for an aerodynamic similar blade (without simplifications) [11] leads to slightly smaller chord lengths by 0.17 % and thus the same “break-even-point” at 101 m/s tip speed.

From an analytical point of view, designs below 101 m/s with larger chord lengths would lead to lower masses [6, 9] but they won’t be practically feasible because the stability limits are too low (see region in light red). Neglecting this, the highest mass saving could be achieved for a redesign maintaining the same tip speed (here 90 m/s) and increasing the chord lengths by 50 %, which fits to the results of [5, 6, 7]. This fact is contrasting the statements of [9], who mentioned that maintaining the same rotor solidity negates the benefits of two-bladed turbines to decrease rotor costs. We would still like to note, that [9] also mentions that an increase in blade weight is proportional to an increase of tip speed, which presents a conflict to the sentence before, but matches the finding of the paper at hand and of [5, 6, 7].

Finally, a blade designed for the tip speed (and the corresponding weight) at the “break-even-point” provides the best compromise between preventing buckling and saving rotor mass. The rotor mass reduction of such a two-bladed turbine (currently with the same rotor diameter) is 18.4 % compared to the three-bladed reference (106.8 t), regarding both blades without non-structural masses. However, to maintain the same absolute power curve (see (I)) the rotor diameter needs to be increased slightly, in this case by 1.92 %. As a consequence, a linear scaling of the rotor mass according to classical cubic scaling laws [10, 18] results in a blade mass of 138.5 t and an overall rotor mass saving of 13.5 %, compared to the reference. Including the non-structural masses, the reduction would be the same since a linear relation is assumed.
In addition to the previous results, the fatigue damages of the three-bladed reference blade, as well as the “break-even-point” blade, have been investigated and compared. The calculations have been performed with the software BECAS from DTU [19], using own pre- and post-processors. To minimize the computational effort, only a specific number of pre-defined cross-sections along the blade spans have been analyzed. Each of these sections is a meshed geometry and thus consists of several elements, including complete material and orientation allocations. To calculate the stresses at each element in a considered cross-section, BECAS initially performs unit load analyses. Afterwards, the unit load stresses are scaled by the actual loads (linear superposition) to get the complete stress time histories. For this purpose, load time series of DLC 1.2, as described in Section 3, have been used as input parameters. Based on the complete stress time histories, a rainflow count of each stress component is carried out to determine the actual cycle number $n_i$, as well as the amplitudes and mean stress values for each element of the considered cross-section. The index $i$ denotes one specific simulation (wind bin, yaw angle, and seed). The maximum permissible number of cycles $N_i$ is determined by means of a linear Goodman diagram in conformity with the GL guideline [20]. Safety factors are neglected. The Miner’s rule [21] is used for the linear damage accumulation, according to [14, 20, 21]:

$$d_{\sigma_r} = \sum_i \frac{n_i}{N_i}$$

BECAS calculates the uni-axial damages $d_{\sigma_r}$ for the longitudinal, transversal, and shear stress components for each element separately (index $r$). To take into account the influence of the different wind speeds, a Rayleigh distribution is implemented afterwards. Furthermore, the damages are extrapolated to 25 years lifetime. Once the uni-axial fatigue damages $d_{\sigma_r}$ have been calculated, a multi-axial damage index $d$ can be computed according to [17, 22, 23]:

$$d = d_{\sigma_1}^{2/m} + d_{\sigma_2}^{2/m} - (d_{\sigma_1} d_{\sigma_2})^{1/m} + d_{\sigma_6}^{2/m}$$

The exponent $m$ is the inverse slope of the S-N curve and the longitudinal, transversal, and shear stress components are represented by the indices 1, 2, and 6. Finally, Figure 4 presents the damage indices $d$ for predefined evaluation positions along the blade span. In order not to mix different influences, e.g. of the ~2% longer blade span, all blades have the same lengths.

The red lines represent the multi-axial damages $d$ of the three-bladed reference blade. Looking at the spar caps on the suction side, the general trend is in line with the material stresses

![Figure 4](image_url)

**Figure 4.** Multi-axial lifetime fatigue damages of DLC 1.2 for blades of same length: (a) relative middle of spar caps at suction side, (b) relative middle of reinforcement at leading edge.
presented in Figure 1, which is reasonable. Since the two-bladed turbine’s blade structure is analytically designed to withstand 50% higher flapwise loads and gravitational driven edgewise loads [5], the black lines illustrate the “break-even-point” blade with upscaled loads from the three-bladed turbine, for plausibility. The results at the different positions, e.g. at the suction side spar caps and the leading edge reinforcement shown here, are as expected in good accordance with the three-bladed reference (red lines). The green lines depict the multi-axial damages $d$ of the “break-even-point” blade based on corresponding load time series out of the software “Bladed”. It can be seen that the damages are slightly below the reference with some exceptions at the outer blade part of the leading edge reinforcements. Nonetheless, detailed aeroelastic analyses have shown that the two-blade turbine has a problem with eigenfrequency excitations and more harmful dynamics, which leads to 163% higher tower fatigue loads [24]. To reduce these loads three controller extensions including a pitch driven fore-aft tower damper had been implemented, which all together decreases the tower loads by 70% [24]. The fore-aft damper itself increases the blade root fatigue DEL by 11% [24], while it did still not exceed the design value of 50% higher DELs. However, the two-bladed turbine’s flapwise fatigue loads tend to decrease less rapidly along the blade compared to the three-bladed turbine. This increases the fatigue damages in the outer blade region (green dotted lines) above the three-bladed turbine’s values (red lines), while the qualitative progression compared to the blade without dampers (green lines) is the same. Since the fore-aft damper represents an artificial and aggressive trade-off between tower and blade loads, the blade without dampers (green lines) is more representative for this study.

5. Conclusions and outlook

The introduced procedure describes a transparent and practicable option to redesign a comparable blade structure of a two-bladed turbine based on a three-bladed reference. It combines an aerodynamic and a structural redesign approach that interacts with each other. In this context, the choice of the blade tip speed and the related chord lengths affects the structural stability and the corresponding blade mass, while structural strength could be held constant. This results in a conflict of interest of preventing buckling and saving rotor mass [5]. The best compromise between both can be called “break-even-point”, which is the most beneficial combination for the blade’s structure of increasing the tip speed and chord lengths, while the blade is designed for equal stress and stability limits. When realizing an aerodynamic redesign based on the same absolute power curve [11], the redesigned “break-even-point” two-bladed turbine’s blade enables a fair comparison with the three-bladed reference, concerning the blades’ aerodynamics, the blades’ loads, and finally the blades’ structures.

For the redesigned two-bladed turbine’s blade of the INNWIND 20 MW reference turbine considered here, the “break-even-point” is located at a tip speed of 101 m/s. As a consequence, the resulting two-bladed design features a simultaneous increase of tip speed from 90 m/s to 101 m/s and chord lengths by 19.1% (for same blade lengths), which matches approximately the results of [7]. This compromise outperforms both extremes of an increased chord or tip speed in contrast to most descriptions in literature [6, 7, 8, 9]. The redesign results in a reduction of the rotor mass by 13.5% and thus fewer costs. The limiting factor here is the buckling stability.

Moreover, the results of the fatigue analyses of the “break-even-point” blade are in good agreement with the values of the three-bladed reference. This applies to both, load time series that are upscaled from the three-bladed turbine by 50%, for which the two-bladed turbine is designed, and loads from actual simulations of the two-bladed turbine.

In summary, this work gives a deeper insight into how the blade’s structural characteristics should be included in the redesign process of a two-bladed turbine. For a practically feasible design, not only aerodynamic aspects and structural strength are important since structural stability is as well an essential design driver. The described procedure to redesign a comparable
blade structure can be easily repeated for any turbine and blade size, for which a fair comparison of two- and three-bladed turbine’s blades should be investigated. In the end, the findings are an important step to finally answer the question of whether two-bladed or still three-bladed turbines could be the more suitable concept for future offshore wind installations.

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