Simplified design criteria for drivetrains in direct-drive wind turbines

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Abstract. Nominal power of wind turbines constantly increases. This goes along with an increase of the generator size, usually accompanied with an increase in mass. To limit at the same time the tower top mass, the mass increase of the generator needs to be restricted. Changing the design concept of the generator to reduce mass, influences its modal behavior and thereby possibly the dynamics of the whole drivetrain. This work identifies simplified criteria for the parameter space restrictions of drivetrain designs, while taking into account the interaction of the generator electromagnetic and structural characteristics. The drivetrain is analysed as an integrated system using a simplified two mass model to evaluate the dependency of system parameters and system stability. For two degrees of freedom an analytical solution is determined. Increasing the number of degrees of freedom to four, the parameter study is carried out using a full factorial analysis and a numerical solution approach. The results point out the necessity to take into account magnetic stiffness when designing the drivetrain to ensure system stability.

1. Introduction

Figure 1. Air gap characteristics with a) the functional dependency of the magnetic attraction force in the air gap ($F_{\text{mag}}$) and the air gap length ($\delta$) and b) a schematic illustration of the change of the radial magnetic force with a non aligned rotor in electrical machines
Current tendencies in industry and research show that the nominal power of wind turbines is still increasing [1]. This goes along with increasing generator size, especially for direct-drive wind turbines [2]. The increase in size also leads to an increase in mass, which is only possible to a limited extent (maximal tower top mass) until design adaptations become necessary, i.e. lightweight construction methods [3]. This influences the modal behavior of the generator and the whole drivetrain due to the changes in mass and stiffness. In the past, interactions of the electromagnetic and structural dynamic forces were neglected when optimising the design as conservative assumptions were used, for example, eccentricity of the generator had to be completely avoided [3]. The tendency to further increase the nominal power towards 20MW or even higher, might require the designer to move to less conservative designs in order to keep the increase of the tower top mass within a limit. Research results indicate the relevance of electromechanical interactions for large generator designs [4, 5], allowing lighter and more flexible designs. These results point out the necessity to examine more thoroughly the interaction of structural drivetrain dynamics and electromagnetic properties of the generator. The detailed analysis of electromechanical interactions in electrical machines have been studied by [6, 7]. The studies show that the structural dynamics of the machine are significantly influenced by the characteristics of the magnetic field. Nevertheless, this field of research is still at its beginning and general restrictions to the system design are unknown.

To evaluate the importance of electromechanical interactions to the system design, it is of interest to know how the drivetrain influences the loading of other wind turbine parts, e.g. blades, shaft and tower. The question arises from the fact, that deflections of the shaft, caused by wind shear or other wind related excitations are increased by the magnetic field as its force \( F_{\text{mag}} \) follows the physical law of \( F_{\text{mag}} \sim \frac{1}{\delta^2} \) with \( \delta \) being the air gap length (see Figure 1). If the overall displacement is increased by the interaction, this might lead to changes in fatigue loads and therefore influence the overall structural lifetime.

Before analysing detailed and computationally expensive generator designs, a sensitivity analysis of system parameters is desirable. Therefore, a simplified drivetrain model with generator and rotor modelled as circular disks is used to perform a theoretical pre-study. The results are presented here. The aim of this work is to identify sensitivities of drivetrain system stability to global design parameters (e.g. shaft stiffness and bearing distance). As a measure for system stability the eigenvalues of the drivetrain are evaluated. The analysis is performed taking into account the interaction of the generator electromagnetic and structural characteristics and ensuring a modal stable system. Thereby, the drivetrain is analysed as an integrated system. This is done in two steps: The first step, is to study the drivetrain system using a two mass model with two degrees of freedom (DoF). As a next step the number of DoFs is increased to four. This increases the complexity of the model in a controlled way. The first step allows to extract an analytic solution to evaluate the influence of the electromagnetic field. By increasing the model complexity the restrictions to the design space are adapted to more realistic scenarios. This leads to the need of numerical solution methods, as analytical solutions are not feasible anymore. Analysing the changes of the available design space, by increasing the number of DoFs, allows to evaluate the significance of specific parameters and DoFs. This can be used in the future to adapt the restrictions of design optimisations of the drivetrain and further reduce the tower top mass.

The presented work identifies restrictions to the design space of direct-drive wind turbine drivetrains, represented by a simplified two mass model with a maximum of four DoF. The system of four DoFs is analysed in stand still and rotating about its symmetry axis. This offers the opportunity to obtain a first understanding of possible influences of the electromechanical interaction to system behavior and where to focus in a more detailed analysis at a later stage. The paper is structured as follows: Section 2 introduces the methodology applied to the presented study, the results are given in Section 3, and Section 4 concludes the study.
2. Methodology

2.1. System specifications

This work focuses on direct-drive wind turbines. To start the analysis, the drivetrain of a direct-drive wind turbine is modeled as a two mass model, shown in Figure 2. Body 1 ($m_1$) represents the rotor of the wind turbine and body 2 ($m_2$) represents the generator rotor. The shaft in between is flexible (stiffness $E \cdot I$) and for simplicity is assumed to be massless. The bearings are assumed to be rigid. The spring with the stiffness $s_{mag}$ represents the magnetic field around the generator rotor. The system is rotationally symmetric and a cylindrical coordinate system with the axes $r$, $\varphi$, and $z$ is used to describe the system. The two masses can have two DoFs each. The first is lateral in the direction of the $r$-axis ($r_1, r_2$) and the second is rotating about the $r$-axis ($\gamma_1, \gamma_2$). The system can be described using the equation of motion according to equation 1 with the matrices $M$ (mass matrix) and $S$ (stiffness matrix), and $\vec{u}$, the vector of DoFs. To populate the system’s matrices, the parameters shown in Figure 2 and the stiffness method are used [8]. Using the steps described in the Sections 2.2 to 2.4 the parameters taken into account to populate the matrices differ.

It is assumed that the absolute masses of rotor and generator as well as the distances between bearings are not directly restricted, but their relation to each other. Therefore, the parameters of the generator are substituted with relative factors to the rotor parameters, e.g. $m_2 = m \cdot m_1$. Furthermore, the bearing distance $b$ is used as reference length for $a$ and $c$ ($a = o \cdot b$, $c = p \cdot b$). Assuming additionally a constant area moment of inertia $I = I_a = I_b = I_c$ and constant material properties (Young’s modulus $E=\text{const.}$) a mean shaft stiffness $s = E \cdot I$ is defined. This stiffness is used as reference for the relative magnetic stiffness $x = \frac{s_{mag}}{s}$. The remaining parameters of the rotor ($m_1, J_1, J_{P,1}$), the bearing stiffness $b$ and the mean shaft stiffness $s$ are taken from the DTU 10MW IEA Wind task 37 wind turbine described in [9]. This ensures a realistic baseline design as reference. All substitutions are listed and explained in column 4 of Table 1. The first column names the parameter, followed by the parameter description in column two and the unit in column three. The before mentioned reference values taken from the reference turbine design

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**Figure 2.** Simplified two mass model of the drivetrain in direct-drive wind turbines for analytic analysis (parameter description is given in Table 1)
Table 1. Parameter definitions, reference values (DTU 10MW [9]) and analysis ranges
(n.v.: not varied, n.e.: not extracted from reference model, s.b.: see below, Par.: Parameter, Ref.: Reference)

| Par. | Description                  | Unit | Definition   | Ref. value | Variation |
|------|------------------------------|------|--------------|------------|-----------|
| $m_1$ | rotor mass                   | t    | -            | 225        | n. v.     |
| $m_2$ | generator mass               | t    | -            | 168.75     | s. b.     |
| $J_1$ | rotor inertia about r-axis   | kgm$^2$ | - | $2.1 \cdot 10^8$ | n. v. |
| $J_2$ | generator inertia about r-axis | kgm$^2$ | - | $1.1 \cdot 10^6$ | s. b. |
| $J_{P,1}$ | gyroscopic rotor inertia | kgm$^2$ | - | $8.6 \cdot 10^7$ | n. v. |
| $J_{P,2}$ | gyroscopic generator inertia | kgm$^2$ | - | $2.15 \cdot 10^6$ | n. v. |
| $c$   | bearing distance             | m    | -            | 5.67       | s. b.     |
| $a$   | rotor overhang               | m    | -            | 3.48       | s. b.     |
| $b$   | bearing distance             | m    | -            | 0.88       | n. v.     |
| $s_{mag}$ | magnetic stiffness       | Nm$^2$ | - | n. e. | s. b. |
| $E$   | Young’s modulus              | N/m$^2$ | - | n. e. | n. v. |
| $I_{a/b/c}$ | shaft area moment of inertia | m$^4$ | I | n. e. | n. v. |
| $s$   | mean shaft stiffness         | Nm$^2$ | - | $1.6 \cdot 10^{11}$ | n. v. |
| $\Omega$ | rotational speed             | rad/s | - | 1 (rated) | 0:1:1 |
| $d$   | proportional damping coefficient | s | - | n. e. | 0.0:0.01:0.1 |
| $m$   | mass relation factor         | -    | $m_2/m_1$   | 0.75       | 0.15:0.3.3.75 |
| $j$   | inertia relation factor      | -    | $J_{P,2}/J_{P,1}$ | 0.0052 | 0.001:0.002:0.025 |
| $j_p$ | gyroscopic relation factor   | -    | $J_{P,2}/J_{P,1}$ | 0.025 | n. v. |
| $o$   | overhang relation factor     | -    | $a/b$       | 3.95       | 0.79:1.58:19.75 |
| $p$   | bearing distance relation factor | -    | $c/b$   | 6.44       | 1.288:2.576:32.2 |
| $x$   | stiffness relation factor    | 1/m$^3$ | $s_{mag}/s$ | n. e. | 0:0.1:10 |

are given in column five, if extracted from the design description. The last column lists the analysis range of varied parameters including lower boundary, increment and upper boundary, separated by "". More details about the analysis range are given in Section 2.3.

Using the standard solution approach for ordinary differential equations $\ddot{u}(t) = \ddot{u} \cdot e^{\lambda t}$ with $\lambda = \alpha + i \cdot \omega$ as eigenvalues of the system [10], the characteristic polynomial as in equation 2 can be determined. As system stability criterion the real part of the eigenvalues ($\alpha$) are used. The imaginary parts ($\omega$) representing the eigenfrequencies are not relevant for this study. For a general system all real parts of the system’s eigenvalues need to be non-positive ($\alpha \leq 0$) to avoid self-excitation of eigenmodes and ensure system stability. This is a simplifying assumption, which is only valid for the isolated system without external excitation, underlining the fact, that this study is the first step in a broader research. In case of an undamped system the real parts equal zero ($\alpha = 0$) and the system is only of marginal stability. In a first step the system is assumed to be undamped, which has two advantages. First, it keeps the equation of motion simple and easier to solve. Second, it leads to a conservative solution for the parameter restrictions as the damping has positive effects to the system stability and will probably allow a larger design space in reality. In Section 2.3, damping is included into the study.

$$M \cdot \ddot{u} + S \cdot \dot{u} = 0 \quad (1)$$

$$q_0 + q_1 \cdot \lambda + q_2 \cdot \lambda^2 + \ldots + q_n \cdot \lambda^n = 0 \quad (2)$$
2.2. Two degrees of freedom

\[
m_1 \begin{bmatrix} 1 & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} r_1 \\ r_2 \end{bmatrix} + \frac{s}{b^3} \begin{bmatrix} 3 \alpha^2(1+\alpha) \\ 6(1+\alpha+2p+\alpha p) \end{bmatrix} \begin{bmatrix} r_1 \\ r_2 \end{bmatrix} + \begin{bmatrix} \frac{3}{\alpha(1+\alpha)} \\ \frac{6}{p(1+2\alpha)} \end{bmatrix} \begin{bmatrix} r_1 \\ r_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}
\]

Starting with the simplest system possible, only the two lateral DoFs \((r_1 \text{ and } r_2)\) are taken into account. Populating the matrices \(M\) and \(S\) the equation of motion is given with equation 3. The standard solution approach named in Section 2.1 leads to \(\det(M \cdot \lambda^2 + S) = 0\), resulting in a characteristic polynomial with \(n = 4\). Therefore, it applies \(q_1 = q_3 = 0\), so that only even terms are included. Using the substitution method \(l = \lambda^2\) the degree of the characteristic polynomial can be further reduced to \(n = 2\). The magnetic stiffness \(s_{mag}\) is set to zero in a first step, in order to investigate the influence of the magnetic field to the design space. As the magnetic attraction force increases with decreasing air gap, the magnetic stiffness lowers the system stiffness \(s_{22}\) at the generator, when considered. With the assumption of an undamped system, the real parts of the four eigenvalues equal zero. Calculating the zeros of the characteristic polynomial under consideration of the condition for the real parts of the eigenvalues, limitations to the parameter space can be extracted.

2.3. Four degrees of freedom in standstill

Taking additionally the DoFs \(\gamma_1 \text{ and } \gamma_2\) into account, the number of DoFs of the system increases to four (see equation 4). Due to space limitations the populated system matrices are not given here. For standstill the undamped system has a characteristic polynomial with \(n = 8\) and in analogy to two DoFs only containing even terms \((q_1 = q_3 = q_5 = q_7 = 0)\). Therefore, the method of substitution can be used again to reduce the characteristic polynomial to \(n = 4\). If damping is included, this method is not applicable and the degree of the characteristic polynomial can not be reduced.

Mathematically, only polynomial up to \(n = 4\) have an analytical solution. For polynomials of higher degree a numerical evaluation becomes necessary. To have a better comparison between the two cases, the solution for the undamped system is also determined using the numerical approach. As stated in Section 2.1, the values are found, based on the DTU 10MW design. For the parameters varied in the study a parameter range of \(1/5 : 2/5 \text{ to } 5\) around the reference value was created. This offers the opportunity to limit the number of numerical evaluations and at the same time analyse a broad parameter space. The resulting values are listed in Table 1 in the last column. A full factorial study is conducted using the parameter variations. The scope of this work is to analyse the sensitivity of the system stability to the different parameters which also includes extreme values and value pairs, to identify dependencies more clearly.

Assuming an undamped system, the restriction \(\alpha = 0\) is applied comparable to the 2-DoF system. The found correlations will only give the conditions of the parameter space fulfilling the initial assumption of an undamped system being of marginal stability. In a second step, proportional damping (also known as Rayleigh damping) is included so that the equation of motion changes to the form given in equation 5. Solutions found then, can be assessed concerning system stability. Only parameter combinations leading to eigenvalues with only non-positive real parts are suitable \((\alpha \leq 0)\).

\[
\vec{u} = [r_1, \gamma_1, r_2, \gamma_2]^T
\]

\[
M \cdot \vec{u} + d \cdot \vec{S} \cdot \vec{u} + \vec{S} \cdot \vec{u} = \vec{0}
\]
2.4. Four degrees of freedom in rotating system
To analyse the influence of the rotation $\Omega$ about the $z$-axis to the system stability, a gyroscopic moment for the wind turbine rotor and for the generator rotor is added (see equation 6), changing the equation of motion to equation 7 with $G$ defined in equation 8. Due to the negative sign of the term, its influence on the system stability is comparable to that of a negative damping. It only occurs for the generator if the bearings are not symmetric around the generator ($p = 1$). With the adapted equation of motion the procedure is repeated.

$$M_{\text{gyro},k} = -i J_{P,k} \Omega \gamma_k \quad k \in 1..2$$

(6)

$$\mathbf{M} \cdot \ddot{\mathbf{u}} + d \cdot \mathbf{S} \cdot \dot{\mathbf{u}} + G \cdot \dot{\mathbf{u}} + \mathbf{S} \cdot \ddot{\mathbf{u}} = \mathbf{0}$$

(7)

$$G = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & M_{\text{gyro},1} & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & M_{\text{gyro},2} \end{bmatrix}$$

(8)

2.5. Postprocessing
All parameter combinations found with the full factorial study, fulfilling the requirement of non-positive real parts are theoretically possible drivetrain designs. The last step to extract restrictions to drivetrain designs is then, to analyse the found parameter solutions for correlations. Comparing the found solutions for the different steps described above helps identifying changes of parameter restrictions depending on the DoFs taken into account. This method gives first indications which parameters are correlated advising for future research where to start more detailed investigations for electromechanical interactions.

3. Results
3.1. Two degrees of freedom
Using the methods and constraints described in Section 2.2 an analytical solution can be identified, which is given in equation 9. From this result two conclusions can be drawn. First, setting the magnetic stiffness to zero the system is in any case stable as overhang relation factor $o$ and bearing distance relation factor $p$ are defined to be positive so that the condition is fulfilled for any system design. Second, including the magnetic stiffness, system instability is possible for strong magnetic fields in combination with highly flexible shafts and a minimal structural stability $s_{\text{min}}$ compared to the magnetic stiffness is needed.

$$0 \leq \frac{6(1 + o + p + op)}{b^3 p (1 + 2o)} - x \iff x \leq \frac{6(1 + o + p + op)}{b^3 p (1 + 2o)} = s_{\text{min}}$$

(9)

$$\frac{s_{22}}{s} = \frac{6(1 + o + 2p + op)}{b^3 p (1 + 2o)} - x = s_{\text{struct}} - x$$

(10)

$$s_{\text{struct}} = s_{\text{min}} + \frac{6p}{b^3 p (1 + 2o)}$$

(11)

$$\frac{6p}{b^3 p (1 + 2o)} \leq s_{\text{struct}} - x = \frac{s_{22}}{s}$$

(12)

The stiffness matrix $\mathbf{S}$ derived using the stiffness method, contains for the DoF at the generator the stiffness coefficient $s_{22}$ given in equation 10. This term can be divided into a structural stiffness $s_{\text{struct}}$ and a magnetic stiffness $x$. The structural stiffness is always positive due to the allowable parameter space for $b$, $p$ and $o$. As $x$ is also defined positive, the magnetic stiffness lowers the overall stiffness of the generator against bending. Comparing equation 9 and 10
shows a direct dependency of the stability criteria and the stiffness coefficient, given in equation 11. Substituting this in equation 9, gives the reformulated stability criteria in equation 12 which demonstrates, that the overall stiffness of the generator against bending \( s_{22} \) is forced to be positive. Therefore, this simplified example already demonstrates the importance of the magnetic field for the overall system stability.

3.2. Four degrees of freedom in standstill

![Figure 3](image_url)

**Figure 3.** Results of full factorial study for four DoFs showing parameter dependencies of relation factors for bearing distance \( p \), overhang \( o \), structural damping \( d \) and magnetic stiffness \( x \).

As a next step the system is extended to four DoFs in standstill and the methods described in Section 2.3 are applied. Thereby, the solution approach changes to full factorial study. The found solutions for the parameters \( o, p, x \) and \( d \) are drawn in Figure 3. The blue points in the scatter plot represent the parameter combinations, that lead to a stable system (i.e. \( \alpha \leq 0 \)). The plot shows all parameter combinations as a scatter plot matrix with the histograms of the different parameters on the main diagonal. This way, two dimensional parameter dependencies can easily be identified. If the whole space is filled with dots, all combinations are stable. Blank space indicates instability of the system. The results show a general dependency between the parameters \( p \) and \( x \) and independence of \( o \) and \( d \). Due to the visible correlation of \( p \) and \( x \) the discretisation of \( p \) for small values (\( p \leq 2 \)) has been additionally increased. This allows a better
identification of the characteristics of the correlation. The higher the bearing distance relation factor \( p \), the lower the allowable magnetic stiffness relation factor \( x \) (see light blue rectangle). Damping has no significant influence to the system stability in this study. Additionally to the previous four parameters the mass factor \( m \) and the inertia factor \( j \) are varied. The results are given in Figure 4 and Figure 5. The parameters \( m \) and \( j \) have no significant influence to the system stability, as any parameter combination occurs in the scatter plot, indicating a stable system.

\[
\frac{s_{33}}{s} = \frac{3(1 + o + 4p^3 + op^3)}{b^3 \cdot p^3 \cdot (1 + o)} - x = s_{\text{struct}} - x
\]  

(13)

The system of two DoFs revealed the importance of the stiffness matrix \( S \) to the system stability. Therefore, the stiffness matrix of 4 DoFs is analysed thoroughly. The entry \( s_{33} \) represents now the stiffness of the generator against displacement in direction of \( r_2 \) and is given in equation 13. Equation 13 can be split into two parts. First, the positive, structural stiffness resulting out of the shaft properties \( s_{\text{struct}} \). Second, the negative stiffness induced by the magnetic field.
and indicated with $-x$. As is readily seen in equation 13, the structural stiffness maximises for $p \to 0$ and $o \to 0$. Physically, this means, that with decreasing overhang relation factor $o$ and bearing distance relation factor $p$ the structural stiffness against bending of the generator $s_{\text{struct}}$ increases. At the same time the term $-x$ lowers the overall stiffness $s_{33}$ with an increase of the magnetic attraction force. If the magnetic field becomes stiffer than the shaft ($x > s_{\text{struct}}$), the overall stiffness $s_{33}$ at this location becomes negative.

To extract the restriction to the parameter space, the correlation of $p$ and $x$ is analysed more thoroughly. $s_{\text{struct}}$ can be further split into a summand dependent on $p$ and one dependent on $o$ as given in equation 14. The split shows clearly, that the bearing distance relation factor $p$ influences the structural stiffness more than the overhang, as $p \to 0$ means $s_{\text{struct}} \to \infty$ and $p \to \infty$ means $s_{\text{struct}} \to 0$ whereas $o \to 0$ means $s_{\text{struct}} \to \frac{3}{b^3 p^3} + \frac{12}{b^3}$ and $o \to \infty$ means $s_{\text{struct}} \to \frac{3}{b^3 p^3} + 1$. This fits with the results shown in the scatter plot, that only $p$ influences significantly the system stability, depending on $x$. The summand, dependent of $p$ indicates a correlation of $p$ and $x$ with the power of three. The scatter plot indicates a reciprocal dependency. The upper limit of the scatter plot is fitted, as shown in Figure 6, with the fitting function of equation 15. The resulting parameter restriction is given in equation 16. Physically, this constraint to the parameter space means, that the stronger the magnetic field, the closer the bearing needs to be to the generator, which is a reasonable result. Compared to the system of two DoFs the parameter space constraints are tighter. The additional DoFs change the degree to which the stiffness is dependent to $p$ (2 DoFs: $s_{33} \sim \frac{1}{p}$, 4 DoFs: $s_{33} \sim \frac{1}{p^3}$, compare equation 9 and 16). Therefore, the allowed bearing distance in combination with the same magnetic stiffness is smaller. This is illustrated for constant $o$ and $b$ in Figure 7.

$$s_{\text{struct}} = \frac{3}{b^3} \cdot p^3 + \frac{3(4 + o)}{b^3(1 + o)}$$  \hspace{1cm} (14)

$$p = \frac{f_0}{x^3 + f_1 \cdot x^2 + f_2 \cdot x + f_3}$$  \hspace{1cm} (15)

$$p \leq \frac{2.7 \cdot 10^6}{x^3 - 2.6 \cdot 10^4 \cdot x^2 + 5.2 \cdot 10^5 \cdot x + 3.3 \cdot 10^4}$$  \hspace{1cm} (16)

At the same time, it must be underlined that this study analyses the drivetrain as an isolated system without external excitation. This means that the found constraints to the parameter space only ensure a system free of self-excitation. The system behavior under external loads needs further investigations. Therefore, this study is the first step towards a better understanding of the influence of the modelling detail of the drivetrain on the design constraints.

3.3. Four degrees of freedom in rotation

When assuming the whole system to be rotating about the global $z$-axis, the parameter space stays constant for the analysed rotational speed as Figure 8 points out. The allowed parameter combinations of $p$ and $x$ are exactly overlapping. This means, that even if the absolute value of the eigenfrequencies $\omega$ is shifted, the shift is not affecting the system stability. The importance of the exact eigenfrequency concerning possible resonances under external dynamic loading needs separate investigation.

4. Conclusion

The presented results show a dependency between the system’s stability and the magnetic stiffness. Focusing on two DoFs, it can be concluded, that taking the magnetic stiffness into account introduces the possibility of instability for the system. Thereby the allowed parameter space is narrowed. The increase to four DoFs further narrows the parameter space, as the
The bearing distance relation factor has to be small enough to withstand the magnetic stiffness. The rotation of the system about the $z$-axis and the proportional damping showed no significant influence to the parameter space.

The used model is a simplified description of the drivetrain and the determined stiffness matrix is dependent on the chosen bearing concept. Therefore, the found parameter restrictions are case specific. For other bearing concepts the calculations have to be repeated to determine the exact parameter restrictions. Nevertheless, the general conclusion can be drawn that the magnetic field can have significant influence to the overall system stability making further investigations on electromechanical interactions necessary. Therefore, coupled simulations using multi-body simulations for wind turbines and finite element simulations for electromagnetic fields in generators are planned. This will allow to study the influence of external loading and evaluate the load distribution.
This work is the first step to better understand the multi-physical effects in wind turbines. Research building on this will allow, in the future, to adapt drivetrain designs more easily to the needs in wind turbines.

5. References
[1] Selot F, Fraile D and Brindley G 2019 Offshore Wind in Europe Tech. rep. WindEurope Business Intelligence
[2] Shrestha G, Polinder H and Ferreira J A 2009 Scaling laws for direct drive generators in wind turbines 2009 IEEE International Electric Machines and Drives Conference, IEMDC ’09 797–803
[3] Hayes A, Sethuraman L, Dykes K and Fingersh L J 2018 Structural Optimization of a Direct-Drive Wind Turbine Generator Inspired by Additive Manufacturing Procedia Manufacturing 26 740–752 ISSN 23519789
[4] Gallego-Calderon J 2015 Electromechanical drivetrain simulation phd Technical University of Denmark
[5] Matzke D, Rick S, Hollas S, Schelenz R, Jacobs G and Hameyer K 2016 Coupling of electromagnetic and structural dynamics for a wind turbine generator Journal of Physics: Conference Series 753 082034 ISSN 1742-6588
[6] Boy F and Hetzler H 2017 On the electromechanical coupling in rotordynamics of electrical machines Pamm 17 365–366 ISSN 1617-7061
[7] Boy F and Hetzler H 2016 On the rotordynamic stability of two-pole synchronous electric machinery considering different load cases Pamm 16 265–266 ISSN 1617-7061
[8] Sack R 1994 Matrix Structural Analysis (Waveland Press) ISBN 9781478610090
[9] Bortolotti P, Canet Tarrés H, Dykes K, Merz K, Sethuraman L, Verelst D and Zahle F 2019 Systems Engineering in Wind Energy - WP2.1 Reference Wind Turbines Tech. rep. IEA Wind TCP Task 37
[10] Matsushita O, Tanaka M, Kanki H, Kobayashi M and Keogh P 2017 Vibrations of Rotating Machinery (Springer, Tokyo) ISBN 978-4-431-55455-4