



Development of test methodologies for experimental lifetime investigations of tidal turbines

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Abstract

As a regenerative energy source, tidal energy can significantly contribute to greenhouse gas reduction, even though the potentially achievable energy output is lower than that of wind or solar energy. The decisive advantage of tidal turbines lies in the simply and reliably predictable energy output. However, their commercial use has so far been impeded by the fact that on the one hand complex mechanical systems are required to convert energy of tidal currents and on the other hand multi-axial loading conditions caused by turbulent ocean currents act on the turbine. For this reason, field tests on prototypes are an essential part of the development strategy to ensure operational reliability. However, in-field tests do not allow for accelerated lifetime testing, so that test bench experiments are becoming an increasingly important alternative. Today, established procedures for testing the turbines main bearings and gearing system are already available, both for setting up the required test configuration and for determining the corresponding test loads. However, the use of advanced calculation methods, such as the finite element method for stress calculation, requires a deep understanding of the examined components and hinders the transfer of the approaches to other components.

To simplify the process of test loads determination, a general methodology is presented, which relies exclusively on standardized empirical calculation rules. Doing this, fatigue equivalent loads can be determined for any component in a simple process. It was shown that the achieved reduction in complexity opens further potential for test acceleration, since several components can be tested simultaneously.

Availability of data and material Not applicable

Code availability The developed code is available upon request by contacting the correspondence author.

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Entwicklung einer Prüfmethodik für experimentelle Lebensdaueruntersuchungen von Gezeitenturbinen

Zusammenfassung

Tidenturbinen können als regenerative Energiequelle einen wichtigen Beitrag zur Energiewende leisten, auch wenn die potenziell erzielbare Leistung geringer ist als die der Windenergie. Ihr entscheidender Vorteil liegt in der einfach und zuverlässig prognostizierbaren Energieausbeute. Ihrem kommerziellen Einsatz steht bisher jedoch entgegen, dass einerseits komplexe mechanische Systeme zur Energiewandlung der Gezeitenströmung erforderlich sind und andererseits mehraxiale Lastzustände infolge turbulenter Meeresströmung auf die Turbine wirken. Zur Absicherung der Betriebssicherheit sind aus diesem Grund Untersuchungen an Prototypen im Feldeinsatz ein fester Bestandteil der Entwicklungsstrategie. Im Feld fehlen jedoch Möglichkeiten zur Prüfbeschleunigung, sodass Prüfstandversuche als Alternative in den Fokus rücken. Zur Prüfung der Komponenten Hauptlager und Getriebeverzahnungen stehen bereits heute etablierte Vorgehensweisen sowohl zum Aufbau der erforderlichen Prüfkongfiguration als auch zur Ermittlung entsprechender Prüflasten zur Verfügung. Der Einsatz höherer Berechnungsverfahren, wie der finiten-Elemente Methode zur Spannungsberechnung, erfordert jedoch ein tiefes Verständnis der untersuchten Komponenten und erschwert die Übertragung der Ansätze auf andere Komponenten. Um den Prozess zur Ermittlung von Prüflasten zu vereinfachen wird aus diesem Grund eine allgemeine Methodik vorgestellt, die ausschließlich auf genormte empirische Berechnungsvorschriften zurückgreift. In einem einfachen Prozess können für beliebige Komponenten Lasten ermittelt werden, die rechnerisch ermüdungsäquivalent zu einem Vergleichslastkollektiv sind. Es konnte gezeigt werden, dass die resultierende Komplexitätsreduktion weiteres Potential zur Prüfbeschleunigung eröffnet, da mehrere Komponenten gleichzeitig getestet werden können.

1 Introduction

Tidal turbines theoretically achieve a fully predictable energy output by accessing energy of the tidal flow [1]. They are therefore predestined to contribute to a greener footprint of power production [2]. The fact that they have not yet prevailed goes back to three major challenges [3]: High Maintenance costs are caused by poor accessibility at sea and by the necessity to get the turbine out of the water to repair the drivetrain or the rotor [4]. Three-dimensional ocean currents cause complex loading situations on the rotor which leads to many different load-superposition states in the drivetrain [5]. Finally, drive train interactions combined with complex loading situations require simplified load assumption in the technical design process. This, all together, results in uncertainties regarding operational safety, which ultimately would cause short service intervals. However, to improve the operational safety, tests on system level can be conducted [6]. Therefore, the typical development process of tidal turbines includes prototypical testing in the field for at least a full maintenance interval. Thus the development time of the turbine is largely dependent on test times, so that there is a considerable need for test acceleration [7].

Ways to substitute field tests were subject of research for many years. Especially test bench experiments were found a possible solution, as they offer ways to apply loads at higher frequencies and amplitudes. A common approach to prepare those experiments is to choose critical components, like main bearings, and develop a test procedure which is equivalent in fatigue to a reference loads expected in field tests, as fatigue in the drive train is a common reason for

failure of turbines next to damage on the rotor and the generator [8]. By means of dynamic rotor load simulations the necessary reference loads can be calculated precisely [9, 10], as this field has been extensively researched over the last decade. After the test procedure is defined, the specified loads are applied to a prototypical turbine in the test rig. Tests configurations including only the critical component instead of a prototypical turbine are not sufficient since the interaction of different components in the drive train significantly influences component loads.

Regarding the development of test procedures, there are many different approaches, which were developed for a specific component. Nevertheless, there are three common steps that are taken in any case. First, the number of load cycles in the dynamic rotor load simulation history is counted, then, the component fatigue is calculated and accumulated, and finally, test loads are derived, regarding the criterion of fatigue-equivalence [8]. The significance of the entire process is based on the correct selection of the critical component [11]. If the critical component is not known, the described process must be repeated for each possibly critical component, including the execution of experimental investigations. However, regardless of which component is specifically investigated, all other drive train components are also subjected to fatigue loads. This means the test speed can be increased if more than one component is considered in the development of the test procedure.

Consequently, the goal of this paper is first to introduce a general methodology for the development of test loads which can be applied to any component of the drive train and second to present a way of testing an additional

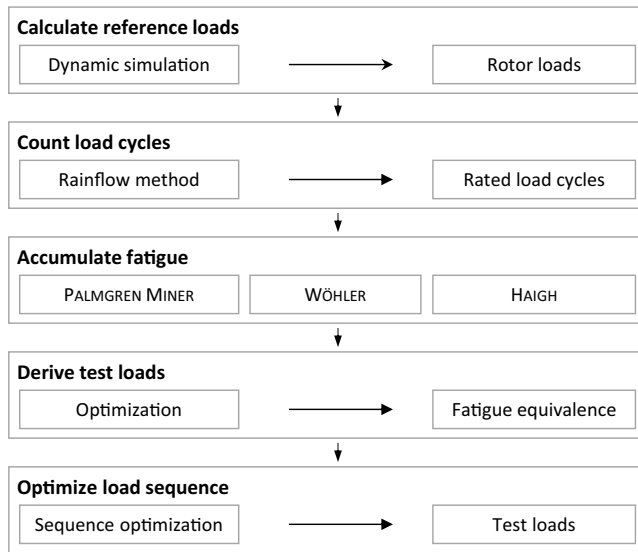


Fig. 1 steps required to determine test loads

component simultaneously without changing the resulting fatigue on the first component nor increasing the test duration. Several state-of-the-art investigation approaches for specific components are simplified and generalized for application to different components, while unavoidable load interactions are addressed by changing the load sequence. The final methodology is structured in a way that more detailed calculation approaches can be implemented.

2 Method

The developed methodology consists of five modules that complement each other and are summarized in Fig. 1. In the first step, rotor loads are determined by means of dynamic rotor load simulations. These loads represent the reference for all further steps.

In the second step, rotor loads are analyzed using the Rainflow method. Results are the number, amplitude, and the mean value of loads of all fatigue relevant load cycles. The evaluation of loads regarding fatigue provides the basis for determining test loads and is the third step. Finally, the sequence of the loads is rearranged to take load superposition into account. The following chapters describe the individual steps in more detail.

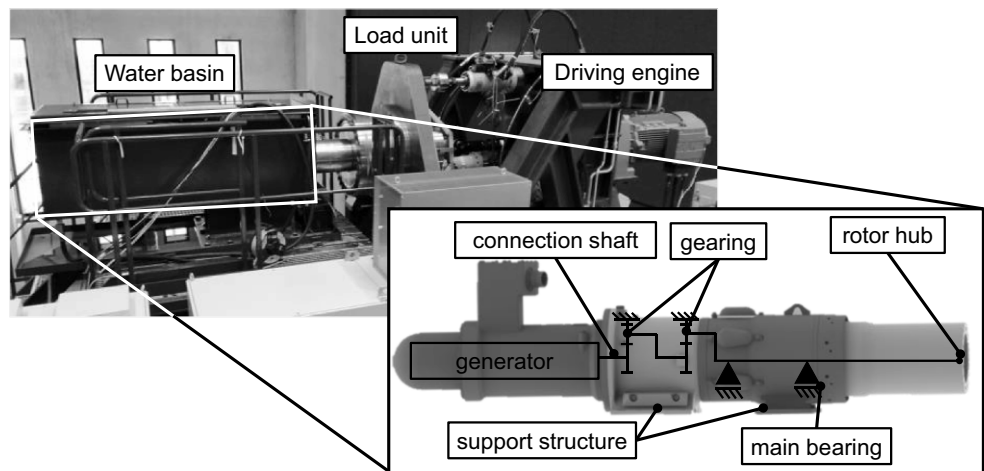
3 Research object and experimental configuration

The test loads for experimental lifetime investigations developed in this paper are applied to the Schottel in-stream turbine (SIT-250). The SIT-250 is a horizontal axis free flow turbine, which is mounted under a floating platform. These platforms are placed in areas of high three-dimensional floating currents and turbulences which causes high multi axial Loads on the 4 m diameter Rotor. The absorbed rotor power is converted to up to 70kW of electrical power by the asynchronous generator. The resulting heat waste of the generator is passively dissipated into the ambient water.

To reproduce field conditions in a controlled experimental environment, a test rig is developed. In this test setup the loads, according to the developed test procedure, are directly applied to the rotor hub. Here, driving torque is induced by an electric motor, while non-torque loads are applied by a hydraulic multi-axial load unit. As the turbine is normally operating in floating water which effects a strong passive cooling effect on the components, the turbine is placed in an actively cooled water basin to transport the heat off the system.

Fig. 2 shows the test configuration with some components of the turbine being highlighted. These components were examined as examples in this study. The turbine hub is connected to the main shaft to transfer the torque to two

Fig. 2 Experimental test configuration and overview of the turbine



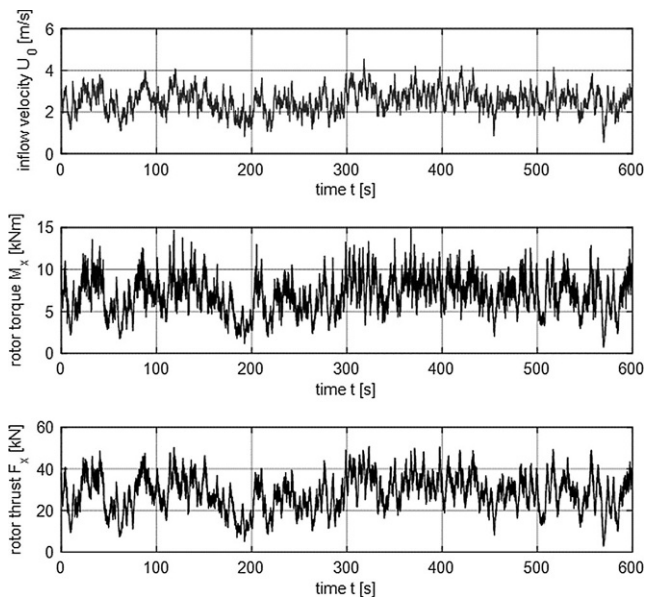


Fig. 3 Time series of inflow velocity at the rotor hub, rotor torque and rotor thrust for an exemplary FLS case

planetary gear stages. The connection shaft connects the sun gear of the seconds gear stage to the generator which is used for the conversion of kinetic energy into electrical energy.

4 Simulation of rotor loads

Dynamic load simulations of the SIT250 turbine (4m rotor) were performed using the Tidal Bladed® software by DNVGL-Garrad Hassan. Tidal Bladed® is the industry standard software for tidal stream turbine design calculations and has been validated against full scale measured data [12]. The hydrodynamic model of the software is based on the blade element momentum theory (BEM). Through the coupling to a structural model, the load related deformation of the blades can be resolved. The full model includes, besides hydrodynamic and structural properties, the turbine controller, and dynamic effects due to 3D turbulence, waves, and the wake of the structure upstream of the turbine.

The load cases are defined following the DNVGL standard for tidal turbines [13] which ensures capturing all contributions to fatigue damage within the targeted design life (fatigue limit states—FLS). The turbine operational states analysed for this paper are power production, stop and idling (freewheeling). Basis are data gathered from a measurement campaign that deployed an Acoustic Doppler Current Profiler at the installation site in the Minas Passage of the Bay of Fundy, Canada [14]. For each load case 10 min of samples are simulated. In cases of a stop (normal or forced), the simulation time is adapted accordingly.

The result of the load simulations used within this work are time series of the load components at the rotor hub. As an example, showing the flow speed at the rotor hub, resulting rotor driving torques and thrust are presented in Fig. 3. For each FLS case, the annual occurrence frequency is derived from the field data and a subsequently performed harmonic analysis. Thus, the number of occurrences of each FLS state within the investigated operating time of five years can be determined. This enables expanding the amount the counted load cycles from one FLS simulation results to the expected number within five years.

5 Generalized approach for determining test procedures

Determining test procedures which are equivalent in fatigue to given reference loads can be approached in different ways. In any case, however, a way is needed to compare accumulated damage of procedure and reference loads. This comparison is almost always based on linear damage accumulation according to Palmgren und Miner [15, 16], who describe the accumulated damage D as a function of the amount of load classes c and load cycles n as well as and the amount of bearable load cycles N according to Eq. 1.

$$D = \sum_{i=1}^c \frac{n_i}{N_i} \quad (1)$$

To fulfill the criterion of fatigue equivalence, test loads must be chosen in a way, that accumulated damage of test procedure and reference loads are equal. This requires determining the parameters amount of load cycles (Chap. 5.1) and amount of bearable load cycles (Chap. 5.2).

As a first step the decision must be made whether the investigation is performed at component or at material level. In the latter, stresses from occurring component loads are determined based on component geometry and material properties. In case of an investigation at component level, this determination is not carried out, so that forces and torques are used directly in calculations. If the calculation is carried out on material level, an advantage is that it is possible to infer tolerable loads directly from known material properties. If, however, only components are examined which are frequently deployed in mechanical engineering, the stress calculation can be omitted in most cases. For such components, standardized calculation rules are available, such as for shafts [17], bearings [18], or gearings [19]. In this study, all calculations were performed at component level. All loads are therefore forces or torques.

Table 1 Components of different fatigue categories

Category	Component
Drive torque, rotation	Gearing
Drive torque, load variation	Connection shaft
Non-torque force, rotation	Inner ring
Non-torque force, load variation	Support structure

5.1 Determining the amount of load cycles

Counting the load cycles largely depends on the investigated component, that specifies which amount of load cycles must be accounted for. Looking at the potentially critical component of the generator shaft, it is obvious that this shaft is not subjected to fatigue by non-torque loads. Bearings and support structure absorb these non-torque loads, so that the generator shaft is only subjected to fatigue-relevant loads by load cycles of the drive torque. The inner rings of the main bearings, on the other hand, are subjected to fatigue even under constant drive torque since each point on an inner ring is subjected to a load cycle when in contact with a roller. As a result, the fatigue of the inner rings is proportional to the number of rotations of the drive shaft and the non-torque force during these rotations, since this force defines the pressure of the roller on the inner ring. In general, when considering the loads drive torque and non-torque load, each of the loads can be looked at in two domains. In the rotation domain, the amount of load cycles is proportional to the amount of rotations of the main shaft. In the load variation domain, the amount of load cycles is proportional to the number of turnarounds in the respective load curve. In total there are four fatigue categories, which each consists of a combination of a fatigue relevant load and a fatigue relevant domain. The fatigue of each component in the drive train can be mostly related to a single fatigue category, as summarized in Table 1.

The process of calculating the amount of load cycles depends only on the domain and is always based on rotor loads described in Chap. 5. That means category 1 and 3 as well as category 2 and 4 can be treated with the same approach.

Calculating the amount of load cycles in rotation domain can be approached by counting fractions of rotations of different loads, which was introduced by Elasha et al [20]. to investigate gearings. As described in Chap. 4, each discrete load point is assigned a rotational speed \dot{n}_{ii} of the drive shaft. Considering the step size Δt , i.e. the time interval between data points, the amount of rotations n_{ii} for each load point can be calculated according to Eq. 2.

$$n_{ii} = \dot{n}_{ii} \cdot \Delta t \tag{2}$$

When subdividing loads into a finite number of classes, the amount of rotations at each class n_i can be calculated by counting fractions of rotations in this class. The result of this calculation for the load drive torque and 50 classes is shown in Fig. 4. The individual load classes have a comparable number of load cycles, whereby the number decreases only at very high torques. This appears surprising at first glance, since the turbine is operated in the range of very low torques most of the time. However, since the torque determines the rotational speed, comparatively few rotations are obtained at low torques.

The Rainflow Counting Method (RCM) according to ASTM E 1049-85 [21] can be used to calculate the amount of load cycles in the load variation domain. In this method the load data is analyzed regarding complete oscillation cycles, determining the amplitude and the mean value of these oscillations. The result is a rainflow matrix as shown in Fig. 5. This matrix was calculated from load data regarding the driving torque at the rotor and clearly indicates that most of the cycles are happening in the very low amplitude and low mean value region. Nevertheless, there are also load cycles in the range of very high amplitudes.

Overall, the amount of load changes can be reliably determined using the RCM method or the counting method according to Elasha et al. [20]. It must be considered, however, that in both methods, information of load sequence and velocity or frequency is neglected.

5.2 Determining the amount of bearable load cycles

The lifetime calculation goes back to WÖHLER [22] and led to a mathematical description of failure probabilities of materials, subjected to specified stress cycles [23]. This so called S-N curve can also be applied to components [24] and has been developed over many years for increasingly accurate predictions. Calculation rules such as ISO 281 [18] for bearings or DIN 3990 [19] for gearings enable the calculation of how many load cycles N the respective component reaches with a certain probability of failure. These rules work with a value pair of load amplitude L_0 and number of bearable load cycles N_0 at this amplitude as well as the slope of the S-N curve p . If the factors that represent further external and internal influences are combined to factor k , Eq. 3 is obtained.

$$N_i = k \cdot N_0 \left(\frac{L_0}{L_i} \right)^p \tag{3}$$

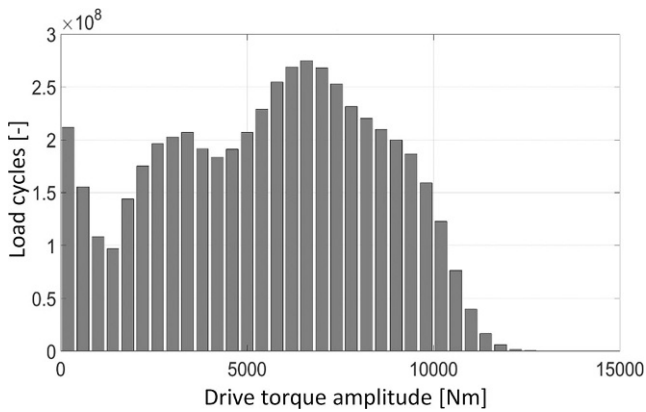


Fig. 4 Accumulated load cycles of the drive torque in five years

If this equation is inserted into Eq. 1, assuming the fatigue of the test loads D_{exp} must be equal to the fatigue of the reference loads D_{ref} , Eq. 4 is obtained.

$$D_{\text{ref}} = \sum_{i=1}^c \frac{n_{i,\text{ref}}}{k_{\text{ref}} \cdot N_{0,\text{ref}}} \cdot \left(\frac{L_{0,\text{ref}}}{L_{i,\text{ref}}} \right)^{-p} = D_{\text{exp}} \quad (4)$$

$$= \sum_{i=1}^c \frac{n_{i,\text{exp}}}{k_{\text{exp}} \cdot N_{0,\text{exp}}} \cdot \left(\frac{L_{0,\text{exp}}}{L_{i,\text{exp}}} \right)^{-p}$$

As described in Chap. 3, the experimental configuration ensures that the conditions in test and real operation are similar. Since a prototype is tested, it is also ensured that

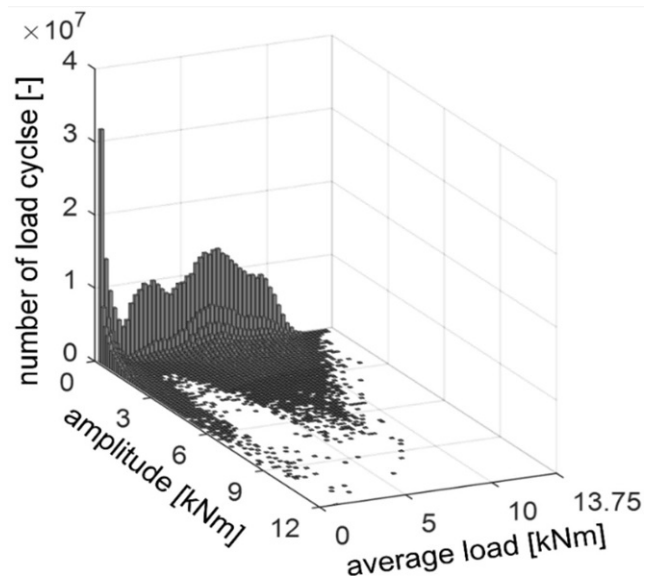


Fig. 5 Rainflow Matrix of reference driving torques

components tested are identical in test and in real operation. By that, Eq. 4 is simplified to Eq. 5.

$$\sum_{i=1}^c n_{i,\text{ref}} \cdot L_{i,\text{ref}}^p = \sum_{i=1}^c n_{i,\text{exp}} \cdot L_{i,\text{exp}}^p \quad (5)$$

With Eq. 5 it is possible to compare two load collectives if the exponent p is known. Table 2 provides a summary of values for p for different components.

If a test procedure is to be derived, KARIKARI proposes to apply load cycles at a constant amplitude until the reference fatigue is reached [25]. In this way the test duration can be adjusted. According to Qui et al. [26] and Choi et al. [27], however, it is known that lifetime calculation is always accompanied by errors. Therefore, test loads represent the fatigue of reference loads the better, the more loads are applied unchanged. For this reason, the approach used in this work is to apply as many load cycles as possible without any changes and to introduce as few substitution loads of a predefined amplitude as possible.

To derive test loads, a calculation routine was set up using the software Matlab which determines test loads by optimizing the ratio of substituted loads and unchanged reference loads. Starting point of the optimization is a set of load cycles which is composed of those reference load cycles that cause the highest fatigue per load cycle. The amount of cycles is chosen in a way that the experimental investigation can be conducted within the available testing time of six months using the test rig presented in Chap. 4. During the optimization process, reference fatigue to be achieved by the test loads and current fatigue of the test loads are calculated and compared. For this, every combination of amplitude and average load in Fig. 5, is summed up based on Eq. 5 and on mean stress correction according to Haigh [28]. By gradually exchanging the load cycles which cause the lowest fatigue per cycle by predefined substitute loads, fatigue of test loads can be adjusted to the reference fatigue. Fig. 6 shows test load cycles for the connecting shaft (left) and the support structure (right), calculated with this optimization.

Substitution loads are predefined before the optimization to simplify the process. To illustrate the effect high or low substitution loads, comparatively low substitution loads were selected for the connecting shaft, which results in a high number of substitution load cycles. If very high substitution loads are selected, as in the case of the support structure, this reduces the number of load cycles considerably. In any case, however, the total reference fatigue can be achieved within the available testing time. That means that applying the calculated test loads within six month corresponds to the same fatigue to the respective component than five-year in-field use.

Table 2 Overview of Wöhler exponents for various components

Component	Value for p	Reference
Gearing	13.6 for pitting 6.22 for root break	<i>DIN 3990-6:1994-12</i> [19]
Connecting shaft	5 for torsion 8 for bending	FKM [17]
Roller bearings	3	<i>DIN ISO 281:2010-10</i> [18]
Support structure	5	FKM [17]

Fig. 7 shows the normalized fatigue of the reference loads, again calculated based on Eq. 5. and on mean stress correction according to Haigh [28]. Load cycles which are taken over into the test loads without a change are highlighted. It becomes clear that a major part of the damage is caused by a very small proportion of the load cycles, when comparing the resulting fatigue to the original rain-flow Matrix in Fig. 5. For the connection shaft and the support structure over 75% and 77% of the normalized fatigue can be taken over unchanged, respectively.

In summary, it is possible to derive fatigue-equivalent test loads from reference loads with the presented methodology, if a value for the exponent p (see Table 2) is available. The following describes how the results can be used to investigate several components simultaneously.

According to Table 1, the components support structure and connecting shaft are in different categories and therefore independently fatigued by different loads. It follows that the test procedure for the connection shaft and the test procedure for the support structure could be carried out on the test rig (Chap. 3) simultaneously, if there were no interactions between the loads. Unfortunately, this requirement cannot be verified in experiments, as component loads cannot be measured directly.

6 Load superposition

To verify whether the superposition of loads has a significant effect on component fatigue, a multi-body simulation (MBS) model of the entire drive train was designed (Fig. 8), using the MBS-software Simpack. In the model, unlike in experiments, it is possible to derive cutting loads at any position.

Simulations were carried out at several stages of constant drive torque and non-torque load. It was found that the load on the support structure is almost independent of the applied drive torque. Loads caused by the drive torque are always several orders of magnitude smaller than loads caused by non-torque loads and bending Moments coming from the turbine rotor as parasitic loads. For the load on the connecting shaft, however, a significant influence of non-

torque loads was found. As a result of non-torque loads, an increase in the eccentricity of the first planetary stage and in the rolling resistance coefficient of the main bearings were observed. Both effects reduce the efficiency of the drive train and thus reduce the torque acting on the connecting shaft. Fig. 9 shows that the torque applied to the connecting shaft is reduced by 40 Nm at maximum non-torque load. This corresponds to approx. 1.8% of the maximum torque but can also account for a significantly larger proportion at lower drive torques.

As non-torque loads influence the load on the connecting shaft, they consequently also influence the shafts fatigue. This goes for both, the test loads and the reference loads. The strength of this influence significantly depends on which loads occur at the same time and by that depends on the load sequence. Load sequences in which high drive torques are mainly present simultaneously with high non-torque loads cause less fatigue than load sequences with a random sequence. This effect goes back to the fact that the load amplitude is included in the calculation of the fatigue to the power of five, according to Eq. 5. Since the use of RCM causes the time reference of individual load cycles to be lost, there is no way of applying test load cycles in the sequence of the reference loads. Consequently, the challenge when trying to test more than one component simultaneously is to first quantify sequencing effects and second to modify the sequence of the test loads to factor in these effects.

As a first step, the MB-simulation results are used to determine the actual torque proportional to loads on the connecting shaft. For this purpose, the simulatively determined correlation in Fig. 9 is transferred into an analytical equation via linearization (Eq. 6). The effective torque $T_{\text{effective}}$ can be calculated depending on the driving torque T_{driving} and the non-torque load F_{NTL} via the constant k approximately according to Eq. 6.

$$T_{\text{effective}} = T_{\text{driving}} - F_{\text{NTL}} \cdot k_1 \quad (6)$$

By applying Eq. 6 to driving torque loads, effective driving torque loads can be determined which, when applied alone, cause the same fatigue as the original driving torque loads when applied simultaneously with the respective non-torque loads. The effective driving torques are then evaluated regarding fatigue based on Eq. 5 and on mean stress correction according to Haigh [28]. As a result the fatigue caused by reference loads is reduced by 8.7%, while the reduction in randomly sequenced test loads is only 4.9% (see Fig. 10). This results in a discrepancy of about 3.8% between fatigue of test loads and reference loads. Presumably, this correlation is caused by the fact that high drive torques especially occur simultaneously with high non-torque loads, which are both caused by high flow velocities in the ocean.

Fig. 6 Rainflow Matrices of test loads for the connection shaft and for the support structure

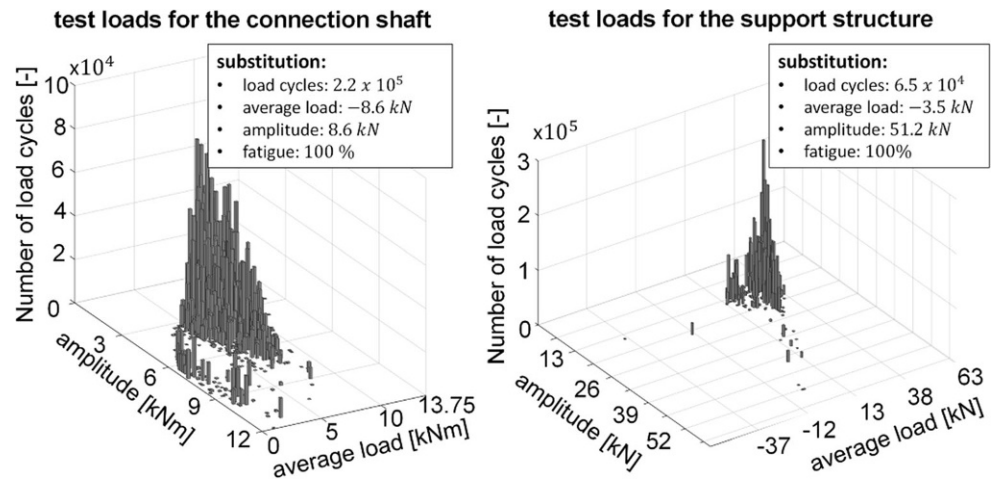


Fig. 7 Normalized accumulated fatigue of driving torque and non-torque loads

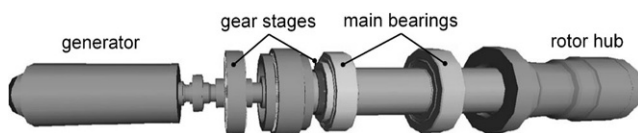
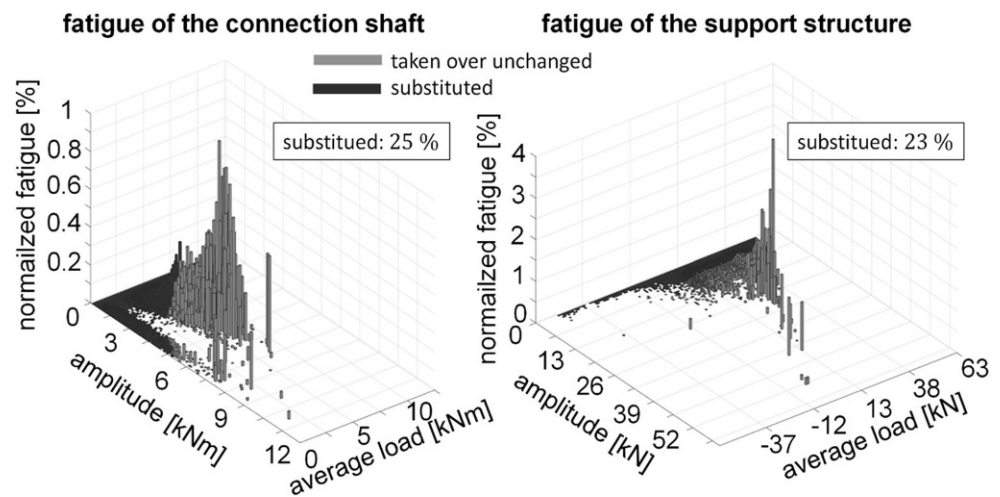


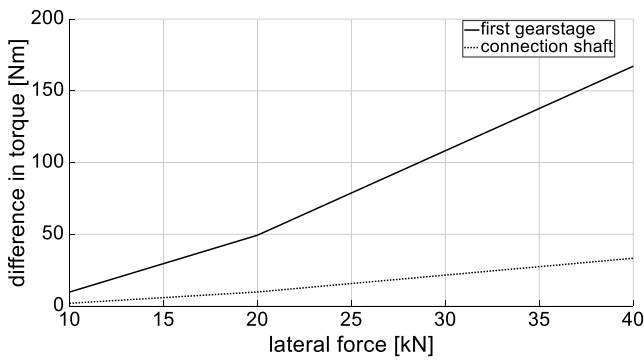
Fig. 8 Basic structure of the MBS-model of the turbine

Due to the very large number ($> 10^6$) of load cycles in the test loads and the resulting number of possible sequence combinations ($> 10^{12}$), no effect could be detected by repeated random arrangement of load sequence. For this reason, an optimization algorithm was developed in Matlab, which varies the load sequence of the test loads until the fatigue of the reference loads is reached. The fatigue is calculated based on Eq. 5 and on mean stress correction according to Haigh [28] in every iteration step. Since sequence problems are computationally expensive due to the large number of possible solutions, the optimization strategy is based on an evolutionary optimization approach, according to Biaoletti et al. [29]. With this approach convergence was achieved, so that the discrepancy between the

fatigue of reference and test loads was reduced by 3.4% to less than 0.4% (see Fig. 10).

Fig. 11 summarizes the described process of sequence optimization. The simplified example shows that, in the first step, an adjustment of reference loads is carried out according to Eq. 6. From the adjusted effective loads, the resulting fatigue is determined to 91.3% of the reference fatigue, using the methodology described in Chap. 6. In the next step, the effective test loads are determined based on those test loads that were calculated according to the methodology described in Chap. 5. The resulting effective fatigue of 95.1% is higher than the effective reference fatigue, which is the reason for the sequence optimization. However, by varying the sequence of the non-torque loads, the effective fatigue can almost be brought into agreement, with a remaining difference of 0.4%.

Application of the methodology to real loads shows the principle applicability for determining test loads for the simultaneous investigation of the components connection shaft and support structure. However, the deviation between



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Fig. 9 Difference in component-torque at 10kNm drive torque

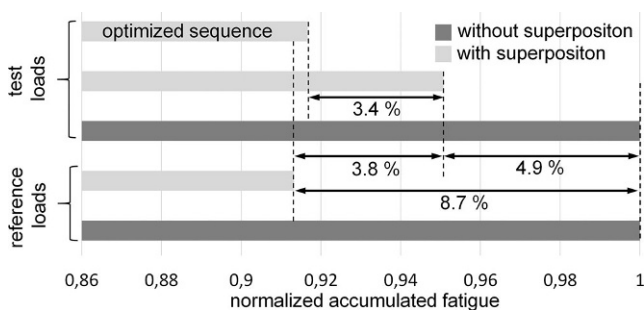


Fig. 10 Comparison of test and reference loads regarding fatigue

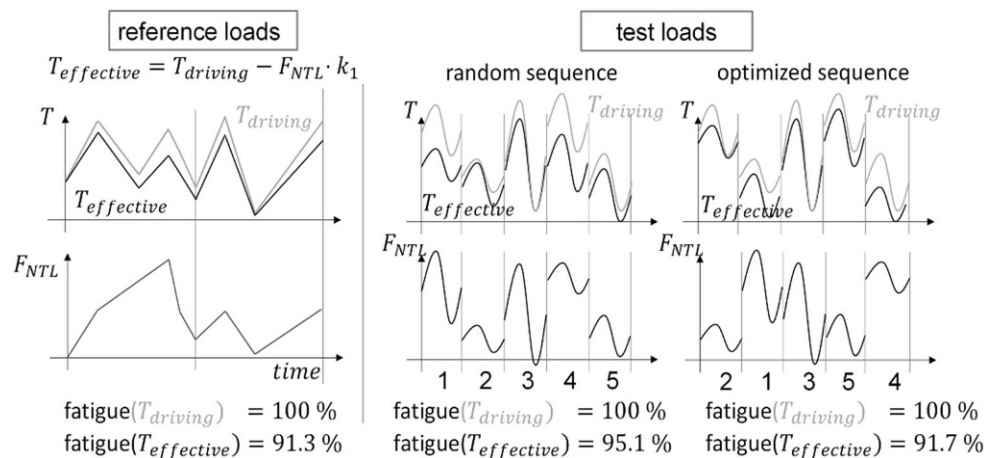
reference and achieved fatigue of 3.8% is comparatively small even without sequence optimization. It is to be expected that the error due to the use of empirical calculation methods is larger than the error due to superposition effects. Nevertheless, neglecting the influence of load superposition effects might cause greater inaccuracies, depending on the investigated component and the used rotor reference loads. The presented methodology provides an approach to calculate these inaccuracies and, if the high computing effort for sequence optimization is justified, to reduce them.

7 Discussion: similarities to wind power drives

When comparing the proposed methodology for the determination of test loads for tidal turbines to procedures usually used for the design and testing of wind turbines (WT), it becomes obvious that both technologies are at different development stages. Both technologies generate electrical energy from natural flow by using a rotor with a connected generator. Nevertheless, tidal technology is still in the testing phase, while wind turbines are used successfully on an international scale. Accordingly, the requirements for experimental testing procedures differ greatly. For the development of wind turbines, IEC 61400-1 and following are established regulations, compliance with which in most cases leads to the turbine achieving the targeted service life. However, the DNVGL standard for tidal turbines [13] is currently far from the level of detail of the IEC 61400. For this reason, the approach described in IEC 61400 of verifying operational safety by experimental tests on sub-systems, such as the gearbox, and mapping the influence of the surrounding system by means of influencing factors cannot be used for tidal turbines, since corresponding influencing factors are not available. Thus, for the safe design of tidal turbines, significantly more prototypical investigations on system level are required, which allow the determination of the critical components.

The methodology developed in this work also shows that many findings from years of wind turbine development can be applied to tidal turbines without restriction. The simulative determination of rotor loads (Chap. 4), for example, is very similar to the simulation methods used for the design of wind turbines—the biggest differences lie in the fluid represented and in the turbulence models used as a basis. The determination of loads in the drive train also shows many similarities. For example, the approach described in Chap. 5.1 for counting load cycles in the load-modification

Fig. 11 Simplified process of sequence optimization



domain via RCM is also used to estimate gearbox service life in IEC 641400-4. However, the next step of inferring fatigue-relevant component loads from the rotor loads already differs considerably. For example, IEC 641400-4 presents to calculate the influence of oil temperature on effective gearbox loads. For tidal turbines it is hardly possible at the present time to calculate oil temperatures considering water temperature and -flow. Recommendations for tests on the complete system are also given in IEC 641400 only to exclude very specific failure modes, such as the occurrence of critical resonances or excessively high oil temperatures. Due to the advanced development stage of wind turbines, concept changes tend to be incremental. Rarely, as in the case of the tidal turbine investigated in this work, completely new concepts are established. However, for newly designed WTs, the presented methodology can serve as a starting point for the development of test loads, which can be used to identify potentially critical components.

8 Conclusion

In this publication a methodology for experimental lifetime investigations of tidal turbines was introduced. The methodology is designed in a way which does not require a deep understanding of the investigated components to be applied. Starting points are a test rig for multi-axial load application and simulated reference load data on the rotor. These loads are decomposed into oscillation cycles and then evaluated regarding fatigue. An optimization algorithm finds test loads which are equivalent in fatigue to the reference loads while taking over as many reference load cycles as possible and meeting a specific test duration. The simplified methodology was applied exemplarily to a tidal turbine's components generator shaft and support structure. By accounting for superposition effects using multi-body simulation, it was shown that the test loads determined for each component can be investigated simultaneously on the presented test rig. Load superposition led to a deviation in generator shaft fatigue between reference and test loads of up to 3.4%. This deviation was reduced to below 0.4% with the help of load sequence optimization.

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Conflict of interest T. Rapp, G. Jacobs, D. Bosse, T. Schröder, R. Starzmann, N. Kaufmann, M. Grassow, S. Scholl and M. Zweiffel declare that they have no competing interests.

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