

Fatigue life on a full scale test rig: Forged versus cast wind turbine rotor shafts

J Herrmann¹, T Rauert¹, P Dalhoff¹ and M Sander²

¹ Institute of Renewable Energy and Energy-efficient Systems, Hamburg University of Applied Sciences, Hamburg, GER

² Institute of Structural Mechanics, University of Rostock, Rostock, GER

E-mail: jenni.herrmann@haw-hamburg.de

Abstract. To reduce uncertainties associated with the fatigue life of the highly safety relevant rotor shaft and also to review today's design practice, the fatigue behaviour will be tested on a full scale test rig. Until now tests on full scale wind turbine parts are not common. Therefore, a general lack of experience on how to perform accelerated life time tests for those components exists. To clarify how to transfer real conditions to the test environment, the arrangements and deviations for the upcoming experimental test are discussed in detail. In order to complete investigations of weight saving potentials, next to getting a better comprehension of the fatigue behaviour by executing a full scale test, a further outcome are suggestions for the usage of cast and forged materials regarding the fatigue and the remaining life of the rotor shaft. It is shown, that it is worthwhile to think about a material exchange for the forged rotor shaft.

1. Introduction

The design of wind turbine main structures shall be examined concerning the fatigue strength to figure out weight-optimization possibilities. Therefore, by measurements on real components, a validation of the fatigue behaviour will be done. A fatigue test rig for the rotor shaft is designed and built in the joint research project BeBen XXL (project partners Fraunhofer IWES, Suzlon Energy and Hamburg University of Applied Sciences). The research project shall fill the gap in systematic testing in between common material investigations and entire turbine system tests. Recently full scale test facilities were developed to investigate the overall performance of wind turbines, less to analyze the fatigue behaviour of single components [1]. But, especially for the constructive design the fatigue strength of the main components is decisive. In today's design process, it is most common to use theoretical models to calculate the fatigue life of a component [2]. This contains high uncertainties regarding size effects and technological effects, that high safety margins need to be considered, which in turn leads to massive components. Separate main component test rigs for fatigue strength and behaviour only exist for a few wind turbine components, like rotor blades and gear boxes so far [1], [3].

The results of the six full scale single tests for the rotor shaft shall influence an adjustment of the guidelines for structural components of wind turbines. This may lead to more reliable rotor shafts with regard to the fatigue strength and simultaneously to a decreased use of material.

To reduce the component life of 20 years to a testing time of less than three months, the life cycle tests have to be highly accelerated in comparison to regular operation. Following, the transfer of real conditions to the test environment is shown and differences in the stress situation resulting from the



increased loading and loading frequency are discussed. These preparatory examinations are necessary to define limitations for the acceleration and an appropriate load level for the upcoming experiments.

At the same time the suitability of other materials for the rotor shaft is investigated. A movement towards cast materials for main components could be seen in the past. The relevance of this topic gets addressed in publications [4] and [5]. Also, numerous new research projects deal with the challenges of the fatigue assessments of large wind turbine components made of cast iron [6], [7] and with the development of new cast materials especially for wind turbine application [8]. There is a lack in detailed studies on advantages and disadvantages of cast turbine main components in comparison to forged components with respect to the fatigue life. Therefore, the following investigations are conducted for the example of the main shaft. Initially, a short insight into the market is presented and the manufacturing process as well as the material properties are considered. Finally, a fatigue life estimation for the rotor shaft regarding different material options is conducted and the fracture mechanical behaviour is analyzed.

To describe the transferability from real turbine conditions to the experimental setup, as well as for the comparison of different materials for the rotor shaft, finite element analyses are done. An FE-Modell of the whole test rig is built up for the consideration in section 2. With strain measurements of the test rig, a validation of this model is carried out. The simulation is performed in ANSYS Workbench. In section 3 the damage of cast and forged shafts after a turbine life of 20 years is also calculated by means of FE-simulations. For the risk assessment based on a fracture mechanical investigation, additional numerical crack propagation simulations are conducted with the software FRANC3D.

2. Transfer of real conditions to a rotor shaft fatigue test bench

The fatigue test rig for the rotor shaft of a wind turbine is presented in figure 1. With a hydraulic pressure system and a load lever made of ductile cast iron a bending moment is introduced into the main shaft. The electric motor in the rear part of the test bench assembly provides the rotation of the shaft, to secure realistic rotating bending loading. The test specimen belongs to Suzlon's 2.1 MW wind turbine, has a weight of around 8 tonnes and a diameter of more than 0.7 m at the bearing seat. The drive train of this turbine type has a 3-point-suspension. The non-locating bearing is integrated in the 3-staged gearbox. In the test bench setup the real main bearing (locating bearing) of the wind turbine is used. As the gearbox is not a part of the test rig, the tilting degree of freedom of the torque support is provided by a replacement structure. The whole testing assembly is located in Bremerhaven at Fraunhofer IWES [9].

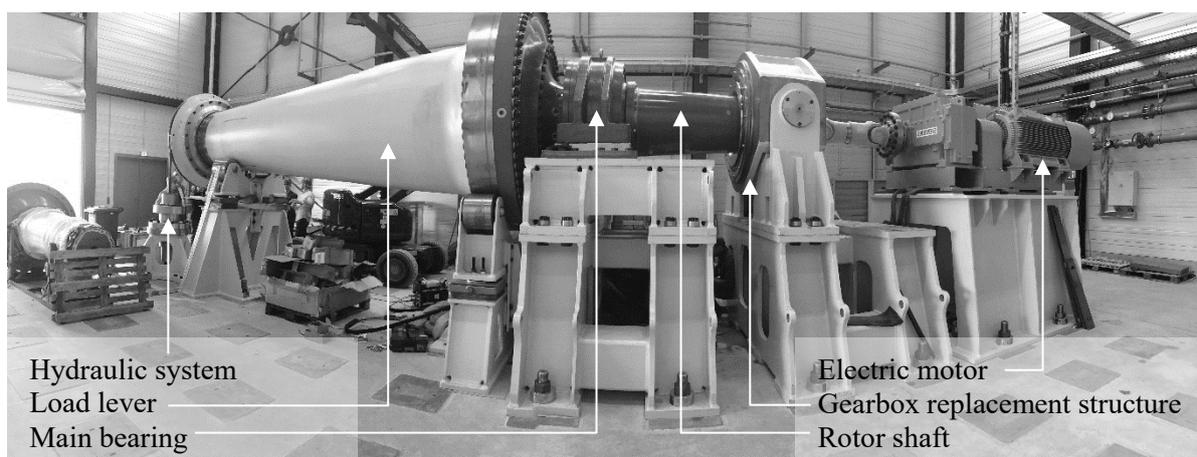


Figure 1. Rotor shaft fatigue test rig [9]

All six tests will be performed with a constant stress amplitude, whereby the load level is not identical for each test. The aim of the S-N tests is the measurement of the finite life regime of the component S-N curve of the rotor shaft.

The operational bending loading of the main shaft is caused by the stochastically distributed wind loads and the gravitational load of the rotor. Following, the simulated bending loads of the 2.1 MW wind turbine rotor shaft are compared to its loading situation at the test rig. These stress spectra are not damage equivalent. The loads at the test rig lead to a damage sum of one, respectively a technical crack. The cumulative frequency distribution of the simulated wind loads for a turbine life of 20 years, leads to a much smaller damage.

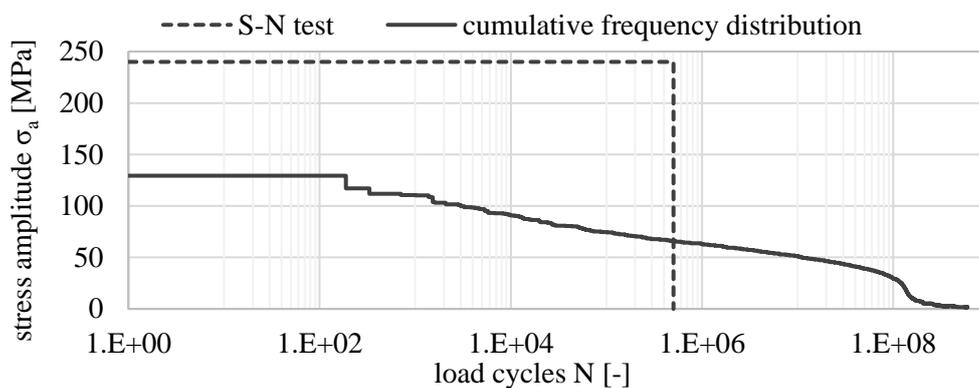


Figure 2. Comparison of realistic bending stress of the rotor shaft in a wind turbine and at the loading situation of the test environment

To accomplish the reduction of the testing time, the load level and the loading frequency are elevated. However, in this case, an adjustment of these two parameters is not sufficient, because before achieving the required increase, unintended side-effects can occur. These side-effects can lead to a damage of the component that is not caused by material fatigue, but rather by thermal fatigue due to a temperature increase or fretting fatigue in the press-fit between the bearing and the shaft. A further measure to accelerate the fatigue of a structure, can be attained by increasing the stress, through an intensified notch effect at the shaft shoulder. The applied methods to accelerate the fatigue life test of the rotor shaft are discussed below.

2.1. Increase of the loading frequency

The increase of the loading frequency provides a speed-up of the testing time. The averaged rotational frequency of a wind turbine rotor is about 0.23 Hz, at the test rig the frequency is set to 1 Hz. However, the change in frequency is not always possible without restrictions in the transferability from experimental setup to reality. According to [10] and [11], a loading frequency between 1 Hz and 1 kHz does not influence the fatigue behaviour of steel. This prediction only applies if the stress level has a sufficient distance to the yield stress of the material and if neither elevated temperatures nor corrosion effects occur. If the frequency decreases, the corrosion process is favoured. The reaction with corrosive media, like water and air humidity is supported by means of a longer dwell time. If the surface of a component corrodes, micro notches evolve and reduce the fatigue strength. Figure 3 shows this effect, at frequencies below 33 Hz the fatigue strength decreases. The stress amplitude also has a decisive impact on the intensity of fatigue corrosion and stress corrosion cracking. If the loading frequency in the accelerated fatigue test is significantly higher in regard to reality, effects like stress corrosion cracking cannot be taken into account. Thus, despite of experimental results, a distortion of the life time prediction can happen.

At high temperatures another fatigue strength-reducing effect arises, the creep deformation. This is fostered more at low loading frequencies and thus at a longer testing time and leads to a damage after a lower number of load cycles (see figure 4). Consequently, the crack propagation rate and the number of load cycles can vary considerably at high temperatures. When highly stressed areas heat up while testing, the loading frequency can gain in influence on the fatigue strength and hence lead to a divergent life time. In accordance with [10] creep processes do not influence the number of load cycles by a change

in the loading frequency above 10 Hz. At high frequencies the damage is due to material fatigue. This applies till a frequency of 20 kHz and at temperatures below approximately 600 °C.

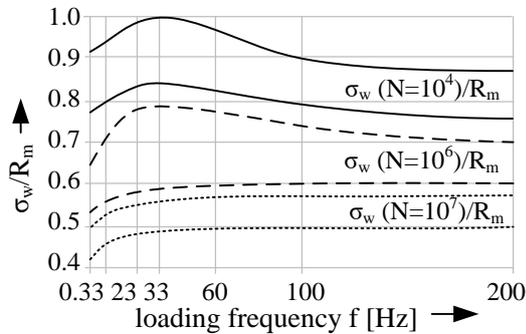


Figure 3. Influence of frequency to the fatigue strength of steel according to [12]

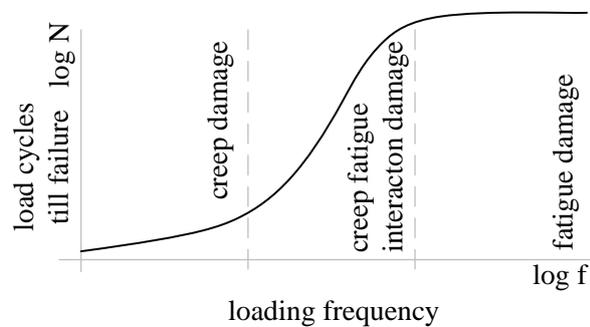


Figure 4. Schematic representation of the influence of frequency to the number of load cycles at temperatures above 0.4 of smelting temperature [10]

2.2. Increase of the load level and resulting change of the stress situation

To determine the implications of the elevated load to the stresses and thus to predict precisely the fatigue behaviour of the component, clarification of the loading-stress-relation has to be provided by means of a finite element simulation. Figure 5 shows the simplified version of the ANSYS-model. At the end of the load lever, a lateral force is representing the hydraulic load introduction unit. The bending moment on the rotor shaft results out of this lateral force and the gravitational force of the load lever. The bearing rollers are simplified with non-linear spring elements. The frame structure is neglected and the front bearing housing is fixed, the rear bearing housing is also fixed, only the tilting degree of freedom is possible. To investigate the loading-stress-relation, all boundary conditions and consequences of deviations in a certain tolerance field to the stress situation of the rotor shaft are considered.

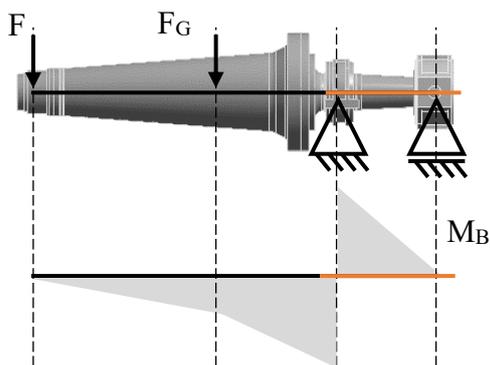


Figure 5. Schematic representation of the rotor shaft fatigue test rig, with F as the hydraulic force, F_G as the gravitational force of the load lever and the bending moment M_B

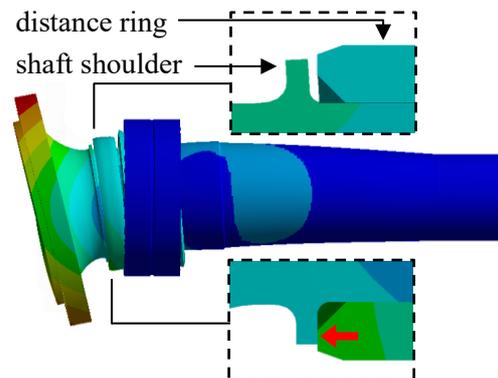


Figure 6. Interaction between shaft and distance ring (scale factor of deformation: 400)

Even the adjacent components, like the distance ring between the shaft shoulder and the main bearing, influence the notch stresses of the shaft. When the rotor shaft deforms because of the bending moment, at the compression side the distance ring presses against the shaft shoulder in axial direction. This behaviour is shown in figure 6. In this picture only the shaft, the distance ring and the main bearing are presented. Due to the deformation of the shaft shoulder by the distance ring on the compression side, tension stresses are induced in the notch radius, which result in a reduction of the compression stresses.

This leads to different effects. First, the mean stress is shifted from zero to a tension stress, whereby the fatigue life decreases and second, the stress range diminishes, which results in a life extension. The second effect has a stronger impact. Because of the non-linearity of the deformation, the resulting fatigue life assumption should be considered very critically. The higher the stresses are elevated, the larger this effect gets.

2.3. Fatigue test acceleration by increased notch intensity

To reduce the fatigue life of 20 years to a few weeks, additionally to the raised frequency, the stress in the notch root of the shaft is increased. Initially, this is realized by the higher load level. Another option, to affect the notch stress is to reduce the notch radius (see figure 7). In this case the design of the component is modified, so the results of the fatigue test apply exclusively for this re-machined new design and have to be converted to the original design afterwards. The reduction of the notch radius raises the stress concentration factor and thereby the mean stress changes. Due to the press-fit of the main bearing, internal tension stresses are induced into the notch environment, which leads to a different stress ratio. With an increased stress concentration factor, the magnitude of the press-fit influence on the notch stresses changes and results in a shift of the mean stress. Also, the geometric and statistical size effect changes, when increasing the stress gradient and thus minimizing the highly stressed volume. However, the environmental conditions, the material and the material conditions, the surface roughness and the type of stress are preserved. Because of the mentioned critical side-effects when elevating the loads and the frequency, for an additional reduction of the testing time, without a continuous increase of the load and frequency, a higher notch impact is needed.

A further reason for the reduction of the notch radius is to ensure that the location of a first initial crack can be predicted with a high reliability. Therefore, an obvious critical area has to exist. This is also caused by the reduction of the notch radius. The shaft design with an additional notch at the flange side (see figure 7) shows a significant reduction of the notch stresses in comparison to the former design with a classic shaft shoulder. This step of optimization entails that the stress range on the flange side (the right side of the shaft shoulder, in figure 7) is higher than in the notch radius. Indeed, the radius on the flange side is larger, but the distance ring does not minimize the stresses on the compression side like in the notch radius. However, the stress ranges only differ about a few MPa, so the prediction of the location of an initial fatigue crack is not obvious, without reducing the initial notch radius on the right of the shaft shoulder.

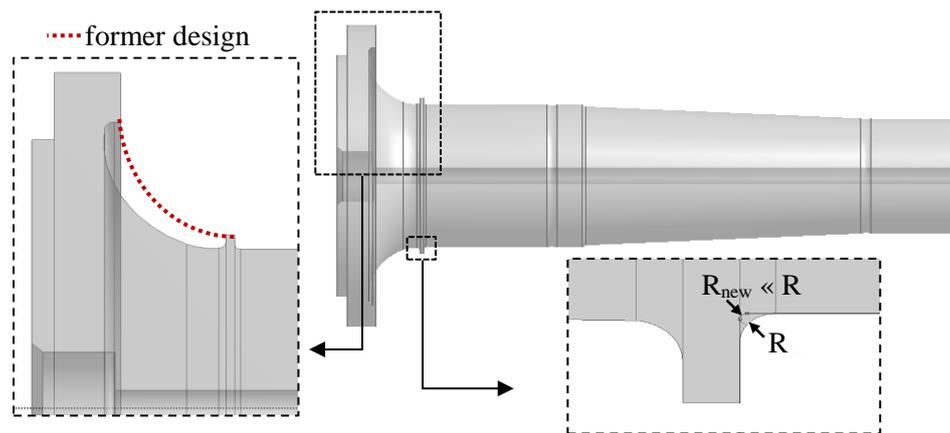


Figure 7. Re-machining of the notch radius to increase the stress concentration factor

2.4. Influence of a temperature rise

With the elevation of the bending moment and the rotational frequency, the temperature in the hotspot of the rotor shaft also increases. This can lead to a variation in the material properties and thus to a reduced component life. This difference in the material properties at high temperatures results out of the

higher diffusion when the temperature increases. To avoid consequential damages, the operating temperature of a component should not exceed 0.4 of the smelting temperature of the material [10]. Due to the alloying of different elements like nickel, cobalt or titanium, the properties of the alloyed substances influence the properties of the base material, which can lead to an improvement, but also to a deterioration of the material properties. If the temperature rises while the test is running, high-temperature corrosion can commence because of the higher reactivity of the material with air and gases.

Furthermore, at temperatures above 0.4 of the smelting temperature a reduction in strength can arise and thus creep processes can occur. Cyclic deformation can cause thermal fatigue and out of heating and cooling processes thermo-mechanical fatigue can take place. Internal stresses in the material can be relieved and thus the conditions change. The size accuracy can be influenced as well [13]. On this account, in addition to the strain gauges, temperature sensors are attached to the test specimen. The temperature in the shaft hotspot is monitored permanently. Moreover, the main bearing is equipped with a cooling system to rule out temperature influence.

2.5. Selection of the test execution

Based on the conducted examination, limitations for an appropriate testing field are defined. In addition to the temperature, the upper limit for the rotating bending moment is given by the influences represented in section 2.2 and 2.3, which increase with rising loads and thus depart from the real wind turbine situation. The lower threshold value for the load is defined by the endurance limit, plus an adequate safety margin because of the large scatter of the S-N curve knee point.

For the acceleration of the test, an elevated loading frequency of 1 Hz is chosen. This proposal only applies because the stress level has a sufficient distance to the yield stress of the material, the temperature elevation of the highly stressed part of the rotor shaft is observed and corrosion effects are not expected in this area.

3. Comparison of forged and cast wind turbine rotor shafts

Nowadays the rotor shaft is mostly manufactured out of cost-intensive forged steel, next to some few shafts made of normal strength cast iron. However, it is investigated, whether high strength cast iron and austempered ductile iron are also suited for the rotor shaft in particular with regard to fatigue. Thus, a calculative comparison considering the operational loads is done to show if a replacement of forged steel by cast iron can be expedient. To constitute the motivation of this examination, the following aspects are regarded additionally.

3.1. Application of cast shafts and axles in wind turbine drive trains

The investigation in [14] shows, that distributed or partly integrated drive train concepts have significant advantages over compact drive trains in multi-megawatt turbine classes. Thus, it can be expected that in the future, the rotor shaft will further remain one of the important main components of the wind turbine drive train. In the consideration of the market the used materials for the main components of wind turbines are regarded more closely. For the rotor hub, the main frames, the torque support, the planet carrier of the gearbox, the stator elements, the bearing housing, the blade root and the tower top adapters, the brake disk of the yaw system as well as the rotor axles are made of cast materials. For the bearings chrome steel or high-alloyed steels are used. Only for the shaft there is a disagreement. The main share of rotor shafts is made of forged steel. Normally, the rotor shaft is subjected to the rotating bending moment as well as the torsional moment. With an increase in the rated power the share of forged shafts is reduced from about 75 % to around 40 %. A shift to direct-driven concepts for larger turbines leads to a higher usage of axles instead of rotor shafts. Axles normally are made of ductile cast iron. They solely have to absorb a non-rotating bending moment. Axles made of cast iron are also sometimes applied in geared turbines. These concepts have an axle to absorb the bending moment and a thin shaft, which is exclusively responsible for torque transmitting. But, cast materials are not only used for axles, in some drive trains of multi-megawatt turbines the rotor shaft is also made of normal-strength ductile cast iron. One of the results out of [14] is presented in figure 8.

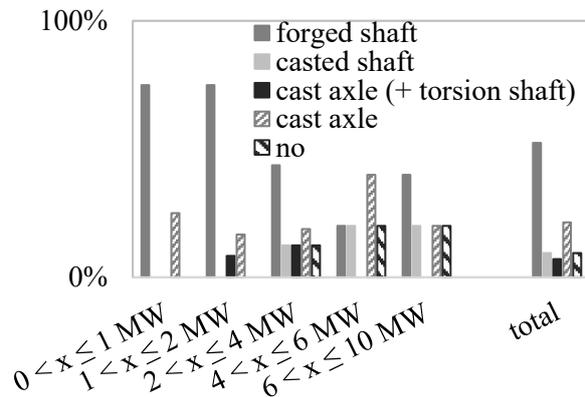


Figure 8. Percentage share of forged steel and cast iron in rotor shaft and axle application for each megawatt class [14]

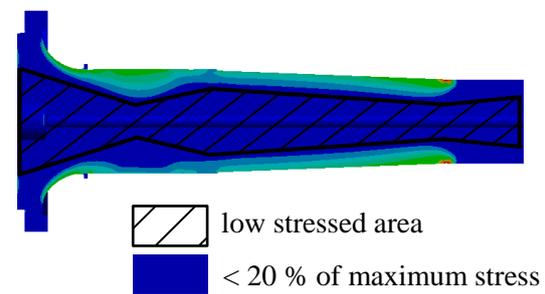


Figure 9. Qualitative representation of the bending and torsional loaded rotor shaft (von-Mises stress)

3.2. Advantages and Disadvantages of forged and cast shafts

A comparison shows the particular advantage of cast iron in the manufacturing process. Due to the mould casting there are hardly restrictions in shaping for components made of cast materials. The design can adjust to the individual loading situation. Thus, by volume reduction in low stressed areas, a substantial saving of material can be achieved (see figure 9). The power consumption for the smelting process of cast iron is lower than the use of energy for smelting steel, because of the lower smelting temperature. The manufacturing costs for the mould of the cast part are not crucial in view of high quantities.

Moreover, owing to the diverse forms of microstructure of the different cast sorts, special combinations of the properties can be produced. This range of possibilities for variations cannot be achieved with other metallic materials [15]. Furthermore, cast iron has advantages in regard to weight saving potentials. The density of cast iron is around 7.2 g/cm^3 and the density of steel is 7.8 g/cm^3 . This means a difference of about 8 % from cast iron to forged steel. The reuse or rather the recycling of cast iron and also of steel is quite feasible. It is also possible to produce a cast iron sort with higher strength characteristics, after melting down an old cast part [15]. Additional reasons for the usage of cast iron are the high damping capacity against mechanical and acoustic vibrations, the surface hardenability as well as the suitability for wear-stressed parts.

Due to the forging process for parts made of steel, pores, cavities and other microstructural degenerations in the material can be excluded or at least minimized. But still existing pores or cavities in steel after forging, result in defects with a curvature radius of nearly zero due to the rolling process, which leads to a far earlier crack propagation. This can be very critical, because in quality controls these flattened defects can easily be missed. For cast components the existence of small pores cannot be excluded. Thus, strict quality controls have to ensure accuracy and prevent that flawed parts go into operation. A further disadvantage of cast iron is the higher brittleness at comparable strength to forged steel. Therefore, in wind turbines primarily ductile normal-strength cast iron variants, with a far lower strength than forged steel, are used. In addition, the cyclic strength behaviour, especially for large components made of high strength cast iron is not completely investigated. Thus, the research is lacking possible applications for high strength cast iron for the rotor shaft. Especially with regard to the fatigue behaviour, the brittle high strength cast iron, has to be critically assessed concerning fracture mechanical aspects.

3.3. Fatigue life and fracture mechanical behaviour

The failure behaviour of a component under variable loads is not comparable with the one under static load. That means, the strength limit for static loading is totally different and far higher than the limit for continuously changing load. During practical application the rotor shaft experiences the torsional stress from the rotor rotation, in addition to the bending stress. The bending moment is composed of a moment

out of the stochastically distributed wind loads and a moment out of the gravitational load of the rotor. To examine the efficiency of materials for main components of wind turbines, the fatigue strength plays a crucial role. The component S-N curves of the rotor shaft made of different materials with a stress ratio of -0.9 are determined according to [2] (see figure 11). The finite life regime is calculated by equation (1). The part below the endurance limit is calculated according to the modified version of Haibach by equation (2). The number of load cycles N for the stress amplitude σ_a depends on the endurance limit with the stress amplitude σ_D , the load cycle number N_D and the slope parameter k .

$$N = N_D \left(\sigma_a / \sigma_D \right)^{-k} \tag{1}$$

$$N = N_D \left(\sigma_a / \sigma_D \right)^{-(2k-1)} \tag{2}$$

The comparison is made for the shaft design of the forged shaft regarding the fatigue behaviour of the forged steel 42CrMo4, the normal-strength ductile cast iron GJS-400-18-LT, the high strength cast iron GJS-600-3 and the austempered ductile iron GJS-800-10. The material properties are taken out of the standards, DIN EN 10083, DIN EN 1563 and DIN EN 1564 (see table 1).

Table 1. Considered materials for rotor shaft application [16], [17], [18]

Material	Tensile strength R_m (t < 30 mm)
42CrMo4s	1100 MPa
EN GJS-400-18-LT	400 MPa
EN GJS-600-3	600 MPa
EN GJS-800-10	800 MPa

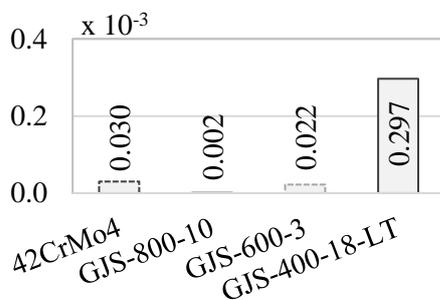


Figure 10. Damage sum D from realistic variable wind turbine bending moment (with N_i as the number of tolerable stress cycles in one bin of stress range)

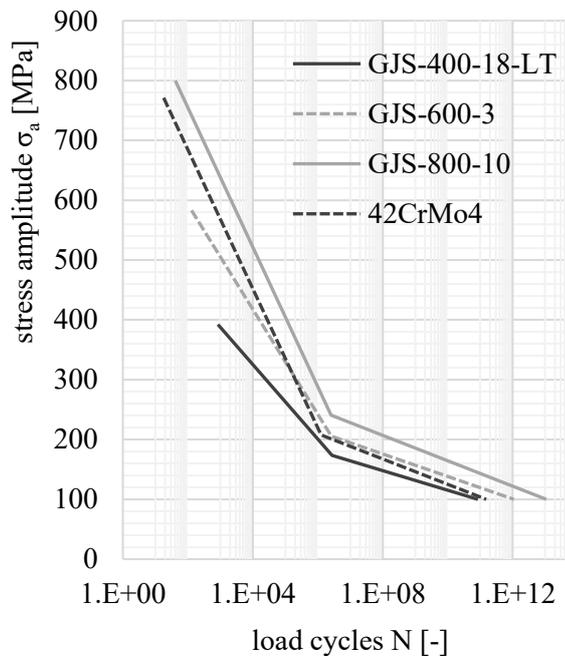


Figure 11. Component-S/N-curves of a wind turbine rotor shaft made of different materials with a probability of failure of 50 % (with material properties from [16], [17], [18])

The austempered ductile iron shaft shows the highest fatigue behaviour at room temperature. Whether the high strength cast iron materials are better suited for this component than forged steel depends on the cumulative frequency distribution of the loads. Only in the low-cycle fatigue regime forged steel is more resistant to fatigue loading, because of higher static properties. The normal-strength cast shaft shows the lowest fatigue strength.

The total damage sum D of a rotor shaft under realistic wind turbine loads of 20 years-service life (out of figure 2) is calculated by equation (3) considering the linear damage accumulation in accordance with the Palmgren-Miner rule modified by Haibach [19]. The damage is determined by the sum of the load cycles at each load level n_i divided by the maximum permissible number of load cycles N_i .

$$D = \sum \frac{n_i}{N_i} \quad (3)$$

The damage sum in figure 10 shows that two of the considered cast iron variants, the high strength cast iron GJS-600-3 and the austempered ductile iron GJS-800-10 are more resistant against the operational load than the forged shaft, using the same design. Because the maximum stresses are almost completely below the endurance limit, the resulting damages of the austempered ductile iron and high strength ductile cast iron are far lower than the damage of the forged shaft. This damage analysis is based on the S-N curves with a probability of failure of 50 % and no safety factors are considered.

The fracture mechanical consideration can be done to minimize the risk of an undetected imperfection. Especially for brittle materials the fracture mechanical determination is important. Nowadays, according to [2], [8] only the ductile cast iron EN GJS-400-18-LT and EN GJS-350-22-LT can be used for structural components of a wind turbine. Because of the low ductility the application of high strength cast materials is not allowed without additional fracture mechanical evidence to make a risk assessment and to specify inspection intervals. In figure 12 the crack propagation curves, calculated by equation (4) with the parameters of the different materials in accordance to [21], [20], are presented. Next to the Forman/Mettu parameters C , n , p and q , the crack opening function γ , the fracture toughness K_C , the threshold value ΔK_{th} and the stress ratio R are required to determine the crack growth rate.

$$\frac{da}{dN} = C \cdot \left[\frac{1-\gamma}{1-R} \cdot \Delta K \right]^n \cdot \left[1 - \frac{\Delta K_{th}}{\Delta K} \right]^p \cdot \left[1 - \frac{K_{max}}{K_C} \right]^q \quad (4)$$

The iterative procedure to determine the crack propagation is pictured in figure 13. A closed solution of the integral is not possible, as the stress intensity range depends on other quantities, which vary during the crack growth. In a cycle-by-cycle analysis the crack propagation of each load cycle can be added up. If the maximum stress intensity factor exceeds the fracture toughness K_C unstable crack growth occurs and the remaining life is reached.

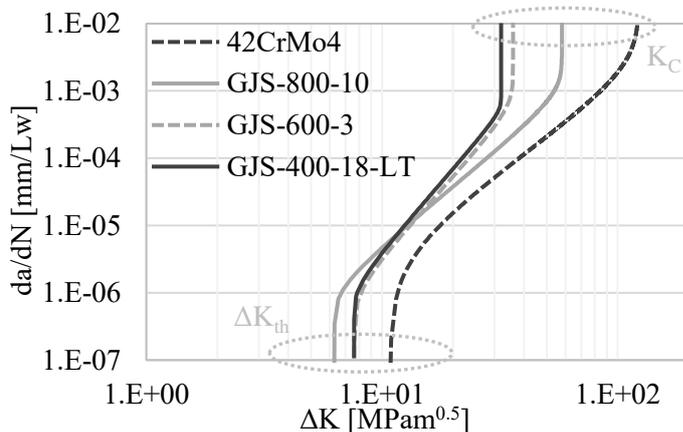


Figure 12. Crack propagation curve of different materials (C , n , p , q as Forman/Mettu parameters)

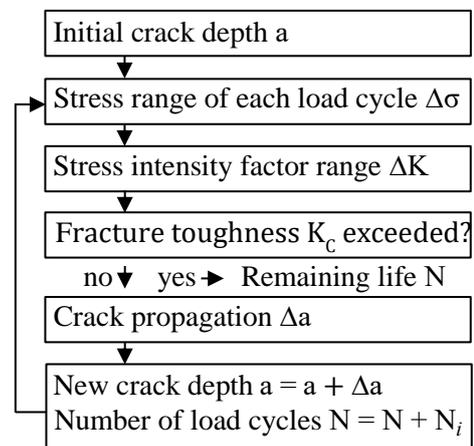


Figure 13. Iterative procedure to determine the remaining life

For the following remaining life calculation for the rotor shaft made of different materials, conservatively a semi-elliptical surface crack with a depth of 25 mm in the shaft hotspot is assumed. While the fatigue crack propagates, the variable operational loads out of figure 2 are expected. For a more realistically distribution of the load sequences, the load cycles of the spectra are greatly reduced (to around 3 month) and repetitively run through till all load cycles of the whole spectra are completed (20 years).

The numerical crack simulation, which is done in FRANC3D, is presented in figure 14. Therefore, a submodel including the hotspot, the expected location of the initial crack, is extracted out of the described FE-model of the rotor shaft. This submodel is implemented in FRANC3D to fit in a flaw and to re-mesh the crack-contained part. Afterwards the model is delivered back to ANSYS. With the input of the FE-results FRANC3D calculates the stress intensity factors of the crack front at different crack depths. From the differences in stress intensity from one crack depth to the next, the required number of load cycles for the crack propagation can be determined for the different materials. In figure 15 the crack growth in the shafts made of different materials is plotted as a function of the load cycles.

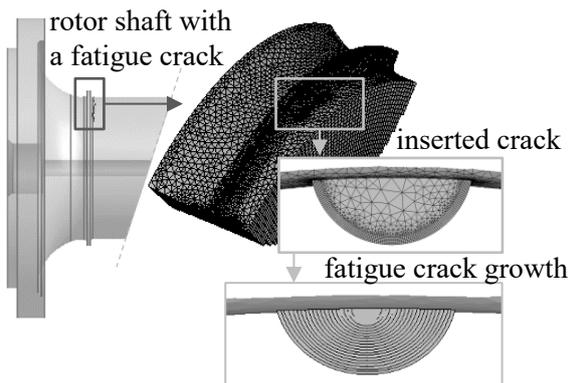


Figure 14. Numerical simulation of fatigue crack propagation in the hotspot of the rotor shaft

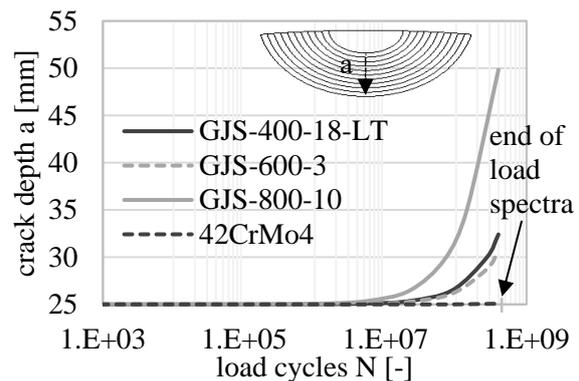


Figure 15. Crack propagation in a wind turbine rotor shaft made of different materials

The crack propagation is presented for the turbine life of 20 years. It has to be mentioned that in none of the considered shafts, despite of the conservative assumption of a 50 mm wide and 25 mm deep initial fatigue crack, unstable crack growth occur. In the forged shaft the crack increment is the smallest, followed by the crack propagation in the high strength cast shaft and then in the normal-strength cast rotor shaft. Initially, in the main shaft made of austempered ductile iron the crack grows the fastest, which is also apparent from figure 12. A consideration of more than 20 years, of higher load levels or lower ambient temperatures would lead to another order.

4. Conclusions

In order to reduce the weight of the rotor shaft two essential examinations are necessary. One of them is getting a better comprehension of the fatigue behaviour of this component by executing a full scale test. The other one is to identify more efficient materials for the rotor shaft.

The deviations for the upcoming experimental full scale fatigue test from turbine conditions have been explained and quantified and the limitations for the acceleration and an appropriate load level range have been defined. The upper limit for the loads is given by different influences, like a change of stress state in the hotspot, fretting fatigue in the interference fit of the main bearing and the temperature elevation. The lower limit for the load is defined by the endurance limit. For the acceleration of the test, the loading frequency of the rotor shaft is elevated from around 0.23 Hz to 1 Hz.

Further on, advantages and disadvantages of cast and forged shafts have been discussed based on a literature research of the manufacturing process, the material properties and of the application of cast shafts and axles in different wind turbine drive trains. In addition, numerical simulations have been done for fatigue life and crack propagation calculations. It is shown, that from fatigue perspective it is valuable to think about a material exchange for the rotor shaft. At operational loading the high strength cast shaft (GJS-600-3) and the shaft made of austempered ductile iron (GJS-800-10) are more suitable against fatigue as the forged shaft (42CrMo4). Furthermore, a risk assessment of the brittle cast iron has been done to show, that despite of a very conservative assumption of a potential defect size and position, there is no unstable crack growth in the rotor shaft during a service life of 20 years.

Acknowledgments

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