Fretting fatigue induced surface cracks under shrink fitted main bearings in wind turbine rotor shafts

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Abstract

This paper deals with the phenomenon of fretting fatigue on the rotor shaft of a wind turbine at the shrink fit of the main bearing. The sensitivity of the rotor shaft for growing cracks that nucleate at the bearing seat is investigated and available approaches of fretting fatigue prediction for shaft-hub and shaft-bearing connections are presented. Their application to the rotor shaft of a wind turbine is assessed and a validation on a full scale and a 1:10 scale rotor shaft fatigue test rig is proposed.

Keywords: rotor shaft; wind turbine; fretting fatigue; wear; crack growth

1. Introduction

The rotor shaft is a highly safety relevant component of a wind turbine, since it carries the rotor of the turbine. Along with the trend of growing rotor diameters and hub heights, the loads on the rotor shaft and the main bearing of a wind turbine are continuously increasing. According to Herrmann et al. (2015), more compact drive train designs are being developed, with shorter and hollow rotor shafts. Due to this, relative movements between the rotor shaft and the shrink fitted inner ring of the main bearing occur. These relative movements can lead to the undesirable phenomenon of fretting fatigue. The contacting surfaces are deteriorating and the fatigue strength of the shaft is reduced unnoticed, see Fig. 1.
Today it is not required by any wind turbine design standard or guideline to assess the influence of fretting on the fatigue life of the rotor shaft of a wind turbine. Also no approach for a holistic estimation of the rotor shafts fatigue life, that incorporates fretting fatigue, is available.

At the same time a more light-weight design of the rotor shaft is desired. This has to be achieved, without reducing the safety of the component. Since wear marks can already be witnessed on the rotor shafts of today’s turbine assemblies, an examination of both the fatigue behavior and the emergence of fretting is necessary. Due to the high efforts and costs of experimentally investigating such a large component (8 t in the considered case, but up to 25 t for larger turbines), both aspects will have to be regarded in one test setup. For this reason, the joint research project BeBen XXL has been initiated. The project is carried out by the project partners Fraunhofer IWES, Suzlon and the University of Applied Sciences Hamburg. Within that project a full scale fatigue test bench for rotor shafts has been developed and set up. On the full scale test bench six shafts from forged steel will be tested. On a 1:10 scale version, shafts from normal and higher strength cast iron will also be tested. In addition, adequate numerical approaches to predict fretting fatigue shall be evaluated and preferably be verified by the test results.

2. Rotor shaft of a wind turbine

The rotor shaft is a standard component of geared wind turbines. In the considered case, the rotor shaft is a solid shaft with an outer diameter of 710 mm at the bearing seat and made from forged steel (42CrMo4). In Fig. 2b a typical arrangement of the drivetrain of a geared wind turbine is shown. The rotor shaft is connected to the hub which carries the three rotor blades. The rotor shaft is supported by the main bearing (fixed) and the gearbox (floating).

The main bearing of such an arrangement is typically a spherical roller bearing and the gearbox is elastically mounted. The main function of the rotor shaft is transmitting the torque from the rotor to the gearbox. In addition, it carries the weight of the rotor of the wind turbine and bending moments resulting from vertical wind shear as well as spatial turbulent wind. An illustration of the aerodynamic load of a wind turbine is shown in Fig. 2a. The turbulent...
wind field leads to resulting forces of different magnitude on the three rotor blades, applying a bending moment to the attached rotor shaft. Since the wind speed is usually higher at greater heights, the resulting forces on the rotor will most of the time be smaller at the bottom than at the top. The bending moment, resulting from this vertical wind shear is acting in the opposite direction than the one from the own weight of the rotor. This means that typically the bending moment at the rotor shaft resulting from the aerodynamic loads will reduce the bending moment resulting from the own weight of the rotor. In a special loading scenario however, a negative vertical wind shear will lead to a summation of both components of the bending moment.

Since the shaft is rotating during turbine operation, it is subjected to a rotating bending moment. In addition, a torsional moment, an axial force and cross forces are induced. An exemplary time series of the bending moment \( M_{Y, NR} \) (coordinate system rotates with the shaft, compare Fig. 2a) at the rotor shaft is shown in Fig. 3. The loads have been derived from a multi-body simulation of a wind turbine in a wind field with longitudinal and spatial turbulence in accordance with IEC 61400-1-edition 2. The software used was FLEX5. The blue sine wave shows the deterministic bending moment on the rotor shaft at the position of the main bearing resulting from the own weight of the rotor blades, the hub and the part of the shaft in front of the main bearing. The rotating bending moment from aerodynamic loads (green line) is, due to the random characteristics of the distribution of the wind speed, less predictable. The orange line shows the superposition of both bending moments on the rotor shaft at the position of the main bearing. The sections marked in grey highlight the occurrence of negative wind shear, here both components of the bending moment act in the same direction.

By the means of a rainflow counting algorithm the load time series are transferred to a cumulative frequency distribution of loads, extrapolated to 20 years of turbine operation and afterwards analytically converted into a distribution of nominal bending stresses at the bearing seat. All stress values in the distribution are also converted to an R-ratio of -1 by considering the materials sensitivity to mean stress. The distribution is shown in Fig. 4. The shape of the cumulative frequency distribution of the stresses reflects the particular characteristics of the different load
components. The deterministic aerodynamic load contributes the triangular shape and the gravity load from the own weight has a rectangular shape, due to the constant load amplitude.

3. Fretting Fatigue

Fretting fatigue is a well-known phenomenon that occurs when two contacting surfaces perform oscillating relative movement under normal load. This can eventually lead to a disruption of the participating surfaces. Small particles are being separated from the surfaces which have a strong tendency to oxidize. These developed oxides are much harder than the surfaces in contact and generate further degradation. According to Babbick (2012), the oxide particles accumulate in the fitting gap and lead to a modified stress distribution, a change in dimension and position accuracy, surface pitting, the formation of grooves and cracks. Aurújo et al. (2007) state that fretting fatigue crack nucleation is driven by mixed-mode. Eventually the crack becomes long enough and the crack growth is dominated by the principle stress. The crack will then continue growing in a direction perpendicular to the contact zone surface. When a surface crack is induced by fretting in a region of high cyclic strain, a growing crack can lead to a complete failure of the component.

4. Crack growth at bearing seat

To get a first impression of the rotor shaft’s sensitivity for the growth of fretting induced cracks at the bearing seat, an analytical crack growth calculation is done. For this purpose and in terms of a simplified approach, the cumulative frequency distribution of the stresses from Fig. 4 is used.

[Fig. 5. Stress distribution on the rotor shaft and hot spot at edge of main bearing seat]

This seems reasonable when considering that cracks from fretting fatigue will at a certain length start growing under Mode I, dominated by the principle stress, in a direction perpendicular to the contact surface. In the case of the rotor shaft, the principle stress is set equal to the bending stress.

The distribution is reduced to a subsequence with a scope of 1/144 of the total distribution, to reduce the influence of the load sequence. 144 is the cycle number of the load level with the least cycles in the cumulative frequency

[Fig. 6. Microsections of fretting fatigue induced surfaces cracks on a rotor shaft at the bearing seat (pictures provided by DNV-GL)]
distribution. This subsequence is repeatedly applied to the rotor shaft, each time going from the lowest to the highest stress level.

This routine is repeated until the crack growth stops, unstable crack growth begins or the full spectrum (20 years) is completed.

According to Carter et al. (2012), the initial crack on the rotor shaft is expected to be at the edge of the contact, in this case the edge of the inner ring of the main bearing. In terms of a conservative approach, the main bearing is assumed to sit at the shaft shoulder. This is also the position of the highest stress concentration on the shaft, see Fig. 5. In the considered case the notch at the shaft shoulder leads to a stress concentration of $K_t = 3.7$. The cumulative frequency distribution from Fig. 4 is therefore multiplied by 3.7. In order to be able to define an initial crack on the surface of the rotor shaft, microsections of a rotor shaft with fretting fatigue induced surface cracks are taken into account, see Fig. 6. This specific rotor shaft completely failed after only 9200 hours (22 months) of turbine operation. Although the specific loading and exact geometry of that broken shaft is unfortunately not available to the authors, it is assumed that the turbine was subjected to loads during operation which were clearly above the design loads. The shaft did not break from pure fatigue, but from fretting fatigue. This emphasizes, that at least in this case, fretting fatigue was more critical for the component.

The cracks shown in Fig. 6 have a length of up to 1.8 mm. For an analytical investigation, different initial crack lengths are compared with regard to crack growth. Therefore, a semi-elliptical surface crack in a solid cylinder under bending loading is presumed and a linear-elastic cycle-by-cycle calculation based on the NASGRO-equation from FORMAN and METTU is done, see Fig. 7.

For the considered rotor shaft assembly and the cumulative frequency distribution of the stresses, the threshold value for crack growth is reached at a crack length of 1.3 mm. According to the calculations, any crack with that length or above will lead to a failure of the component within the 20-year life span. For a crack with a length of 1.3 mm the shaft will break after $9/10$th of the component’s life.

A shaft that exhibits a crack with 1.5 mm in length will break after $1/2$ and with 2.0 mm after $1/7$th of the expected life time. These results show that fretting fatigue at the bearing seat can be critical for the life of the rotor shaft, when high stress concentration is acting on the fretted surface. This perception also corresponds to the fact that the specific rotor shaft came to such an early failure.

5. Estimation of fretting fatigue

The crack growth calculations and the case of shaft failure presented above are pointing out the challenges that go along with shrink fitted main bearings on wind turbine rotor shafts. This holds true especially against the background of continuously growing rotor sizes and loads and a request for lighter designs. This enforces the need for a feasible approach of assessing the influence of fretting fatigue on the rotor shafts life. However, the crack growth calculation
made above has the problem of completely neglecting the phases of fretting fatigue crack nucleation and short crack growth.

In the past, numerous attempts have been made to predict the impact of fretting fatigue. Yet, none of them have been adopted to the rotor shaft-bearing connection of a wind turbine, nor have they been validated for such an application. Extensive surveys on available concepts can be found in Zeise et al. (2014), Talemi (2014), Carter (2012) and Vidner et al. (2007). An overview is presented in the following section.

5.1. Notch factor model

In design guidelines like FKM (2012), the dimensioning of certain distinguished shaft-hub connections is specified. Stress concentration factors are defined in order to take into account the reduced fatigue strength of the component. Although there is good compliance with the specifically mentioned applications, a transfer to other designs is limited.

5.2. Energy based models

One of the first approaches towards fretting fatigue assessment has been developed by Funk (1968). The Funk criterion corresponds to a specific friction energy:

\[ p_f \cdot s \leq (p_f \cdot s)_{crit} \] (1)

The process of wear will start to set in, when the product of contact pressure \( p_f \) and slip \( s \) exceeds a critical limit. This limit has to be determined experimentally for each application.

Ruiz et al. (1986) found out that the location of crack initiation can only be predicted safely, when there are sufficiently high tension stresses in the area of high specific friction energy. They defined the fretting-fatigue-damage-parameter (FFDP):

\[ FFDP = \sigma_f \cdot \tau \cdot s \] (2)

It states that the crack will initiate, where the product of the frictional shear stress \( \tau \), the tangential tensile stress \( \sigma_f \) and the slip \( s \) reaches its maximum. Unfortunately, critical values for the FFDP do not exist. It can therefore only be used to predict the likely location of crack initiation. Further developments have been made by Ding et al. (2007) and Vidner et al. (2007).

5.3. Multiaxial models

Multiaxial models try to take into consideration the time-variant and non-proportional nature of contact stresses. To account for this, critical plane approaches are used. The objective of these approaches is that a crack initiates in a plane of maximum strain. This critical plane is found by calculating a value from a distinct combination of stresses and strains, systematically for varying plane orientations. This value then has to be compared to an appropriate parameter like the Smith-Watson-Topper parameter (SWT) proposed by Smith et al. (1970):

\[ SWT = \sigma_{n, max} \frac{\Delta \varepsilon_1}{2} = \sigma_f \left( \frac{2 N_f}{} \right)^{2b} + \varepsilon_f \left( 2 N_f \right)^{6c} \] (3)

with \( \sigma_{n, max} \) being the maximum principal stress and \( \Delta \varepsilon_1 \) the normal strain amplitude on the critical plane. \( E \) is the Young’s modulus, \( \sigma_f \) and \( \varepsilon_f \) are material strength parameters and \( b \) and \( c \) are material fitting parameters.

Another parameter has been developed by Fatemi et al. (1988), the Fatemi-Socie (FS) parameter:

\[ FS = \frac{\Delta \nu_{max}}{2} \left( 1 + k \frac{\sigma_{n, max}}{\sigma_{y, yield}} \right) = \frac{1 + \nu}{E} \left( 2 N_f \right)^b + \left( 1 + \frac{1}{2} \left( 2 N_f \right)^b \right) \] (4)
where $\Delta \gamma_{\text{max}}$ is the maximum range of shear strain and $\sigma_{n\text{max}}$ the maximum normal stress perpendicular to the critical plane. $\sigma_{\text{yield}}$ is the yield strength and $\nu$ the Poisson ratio. Further parameters have been developed for example by Findley (1957) and McDiarmid (1994).

In general, multiaxial models predict the number of cycles until crack initiation and also the location and orientation of the crack. Yet, according to Zeise et al. (2014) all these concepts have the disadvantage that they neither take into account the increasing friction coefficient, nor the change in local stress distribution.

5.4. Iterative fretting wear model

An iterative model to predict the fatigue strength of a component subjected to fretting wear has first been developed by Paysan (2000). Further improvements have been made by Njinkeu (2009) and Zeise (2015). This concept is incorporating the buildup of wear debris between the surfaces in contact, as well as the transport processes of that debris. Thus, it is possible to consider the changing coefficients of friction and stress distribution within the interference fit. According to Zeise (2015) the endurance limit of an assembly subjected to fretting is predicted by excluding crack initiation. Therefore, a critical state of crack initiation is identified by monitoring the local frictional shear stresses during the simulation of fretting development. A prediction of finite life is not intended. Although the concept has been proven to be valid for different geometries, Zeise (2015) states that it is lacking experimental validation for large-area closed contacts.

5.5. Approach of critical bearing load

In terms of shaft hub connections, a shrink fitted bearing on a shaft is a special case. This is mostly due to the complex stress distribution in the contact zone. According to Maiwald (2013), bearings that are subjected to large bending moments and axial loads will experience not only an axial slip between shaft and inner ring of the main bearing, but also a tangential creeping movement of the inner ring on the shaft. The reason for this is a wave-like deformation of the inner ring that results from a normal force acting on the rollers. Under rotation the rollers will introduce a tangential force onto the wave-shaped inner ring, forcing it to rotate on the shaft. Further numerical and experimental investigations on the critical bearing load that will lead to relative movements and possibly crack initiating fretting wear, have been done by Babbick (2012) and Aul (2008). This model is also lacking validation for large bearings.

5.6. Application to the rotor shaft of wind turbine

Basically all approaches of fretting fatigue prediction stated above, can be applied to the rotor shaft. Especially the iterative fretting wear model seems promising, because of its good accordance with the experiments. However, this concept and also the approach of critical bearing load, aim on excluding an initial crack respectively critical relative movements. In practice this could likely result in the need for an interference fit which would be difficult to realize. Hence, an application of a multiaxial model would provide the opportunity of identifying the moment of crack initiation, in order to perform a subsequent crack growth investigation.

6. Experimental setups

In order to assess the fatigue strength of rotor shafts and the emergence of fretting wear, two test benches have been designed. One setup is in full scale, the other one in 1:10 scale. In both test setups the rotor shaft is loaded with a rotating bending moment.

The full scale test bench has been developed and constructed by Fraunhofer IWES in Bremerhaven, see Fig. 8b. The support of the shaft is similar to the situation in the real wind turbine. The original main bearing is used and the second support has the same degrees of freedom as the gearbox. The load is introduced to the shaft by a load lever with 5.5 m in length. A cross force is applied to the end of the lever. In terms of a classical S/N test the load is kept constant and the shaft is rotated until a crack initiation is detected. During the test, the shaft’s condition is monitored.
by a large number of strain gages. In addition, displacements and temperatures are logged. In total, six shafts made from forged steel (42CrMo4) will be tested at different load levels.

The same test procedure will be carried out for the 1:10 scale version of the test bench. This one has been developed and assembled by the University of Applied Sciences in Hamburg. The major difference is the load application, see Fig. 8a. At the small test bench the bending moment is applied by two cross forces (blue hydraulic cylinders).

This way, resulting cross forces in the shaft of variable magnitude can be achieved. In addition to the six shafts of 42CrMo4, six shafts of EN-GJS-400 and six of EN-GJS-700 will be tested.

After the test, the surfaces of the shafts under the main bearing will be investigated in detail. This way the prediction accuracy of the fretting fatigue estimation models can be determined.

7. Conclusion

The rotor shaft of a wind turbine and its typical loads are clarified. The basic problem of relative movement between the main bearing inner ring and the rotor shaft and the resulting fretting fatigue phenomenon are explained.

To give an impression of the effect of fretting fatigue on the rotor shaft, a case of early shaft failure is presented. According to the specific cracks that can be seen on microsections of the shaft, a crack growth calculation is done, confirming the possibility of a complete failure in the presence of high local strain. This emphasizes the need for a feasible approach for the assessment of the rotor shafts fatigue life, incorporating fretting fatigue. Thus, the range of available approaches is presented. The next step will be to implement those models and carry out a numerical investigation of fretting fatigue crack nucleation and growth. Accompanying full scale and 1:10 scale tests of the rotor shaft in a realistic setting are conducted, offering the possibility of validating the fretting fatigue simulation models.

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References


