Fatigue Analysis of a Wind Turbine Power Train

N. Ghareeb; Inst. of General Mechanics (IAM), Aachen University of Technology, RWTH Aachen
Y. Radovcic; SAMTECH Germany GmbH, Hamburg

EXTERNAL ARTICLE

ENGLISH

During operation, wind turbine power trains are subjected to a diverse spectrum of dynamic loads. The high number of load cycles and operating ranges during the life cycle of the wind turbine makes the fatigue conditions particularly important. This has to be considered for design parameters.
One of the components of the wind turbine that is subject to bending and torsion loads is the shaft. A special feature of the rotational-bending loading is that during a single revolution of the shaft, maximum tensile, as well as compressive stresses are observed on its surface.
Because of shaft rotation, a fatigue crack can be initiated at any point on the periphery of the shaft which will finally lead to its fracture.
In this paper, a brief introduction to fatigue analysis under these conditions is given. After that, a broken shaft of the wind turbine gearbox is analysed and investigated, in order to determine the features that resulted in the failure.
A macroscopic investigation of the fatigue-fracture surface is done with the help of relatively simple techniques. Consequently, a simple 3-D FEM analysis is carried out by modelling the shaft in order to get an idea of how the fatigue fracture began and propagated along the shaft.
Moreover, a theoretical analysis is made in order to illustrate the results.
Finally, a comparison is made between these results and the corresponding data from an FEM Simulation.

Introduction

When designing mechanical and structural components, two load contributions are of prime importance, the extreme loads and the fatigue loads. The extreme loads are the loads which could cause the structure to fail due to loads exceeding the yield strength or, possibly, ultimate strength of the material.
The fatigue loads are cyclic, each of which may be substantially below the nominal yield strength of the material, and which could lead to its fracture after a sufficient number of fluctuations. Note that in addition of the cyclic loads, there are loads due to non-stationary aerodynamic loading. Furthermore, some resonant frequencies might be excited due to a large wind operating range.
Generally, a fatigue crack is formed at a point or points of maximum local stress, and it propagates under applied cyclic stresses through the material until complete failure results.

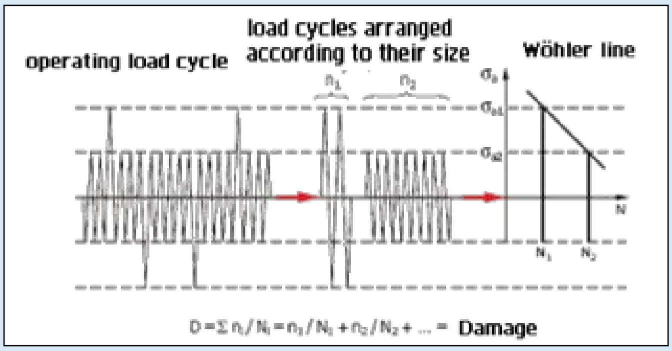

Fig. 1: Classification of load cycles according to Miner

Fig. 2: The broken shaft at the middle stage of the gearbox of a wind turbine

Fig. 3: Section of the shaft where failure occurred

The fatigue behaviour is influenced by many variables. They include the type of loading (uniaxial, bending, torsion), shape of loading curve, part size, part finish, operating temperature and atmosphere. For these reasons, the prediction of the fatigue life of a material is complicated, since it is very sensitive to small changes in these factors and loading conditions.

In order to calculate the life of a component subject to variable loading, a method is needed that relates constant-amplitude fatigue test data to a random stress history. The Palmgren-Miner cumulative damage rule provides a simplified approach to this problem (Fig. 1). This method assumes, that the damage in the mechanical component increases linearly, under fluctuating load, with the number of the load cycles, and it breaks when D = 1. (Eq. 1)

Moreover, in case of load cycles with different amplitudes, the fracture damage Di will be summed up for different stresses corresponding σ_{ai} .Thus,

$$D = \sum_i D_i = \sum_i \frac{n_i}{N_i} \quad \text{(Eq. 1)}$$

where n_i is the expected number of load cycles with load case σ_{ai}, N_i is the number of load cycles at σ_{ai} when failure occurs based on the Wöhler-line or Haigh-Diagram.

There are several ways to compute the number of life cycles. The easiest one, used in the analysis here is to use a "rainflow" count on the equivalent stress. Other methods include maximum shear plane determination. These methods use the full stress tensor.

Fatigue Analysis of Wind Turbines

Due to the turbulent nature of the wind, and due to the high number of load cycles which occur during the life of the turbine, fatigue considerations must be taken into account while designing wind turbines. The respective load spectrum, which is composed of the load amplitudes and the corresponding load cycles, depends mainly on the dynamic properties of all components of the wind turbine. As a result, the whole system must be modelled and simulated, in order to determine these loads.

Consequently, in order to perform the simulation, the tools and models to be used must fulfil high demands and requirements, since the results must be accurate and reliable in terms of stress estimation.

Although failure in the bearings is the main source of damage that occurs in the wind turbine, other components are also subject to failure.

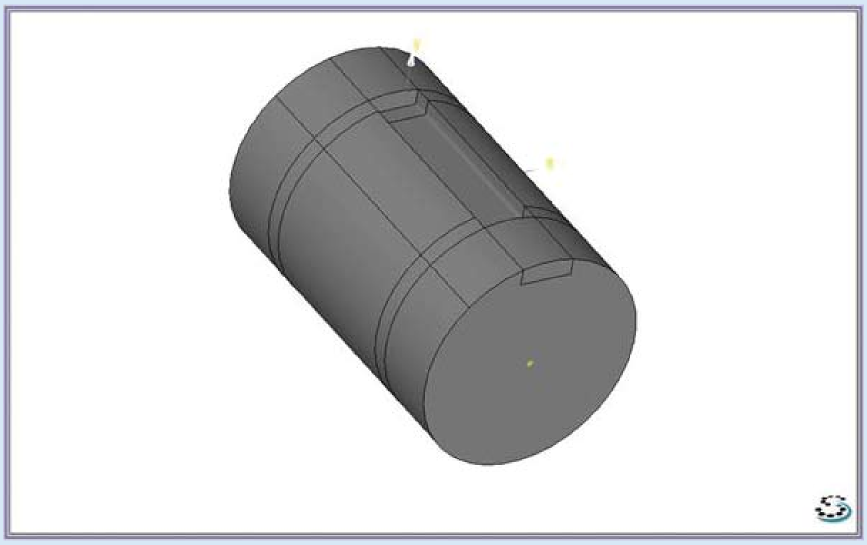

Fig. 4: Shaft geometric model

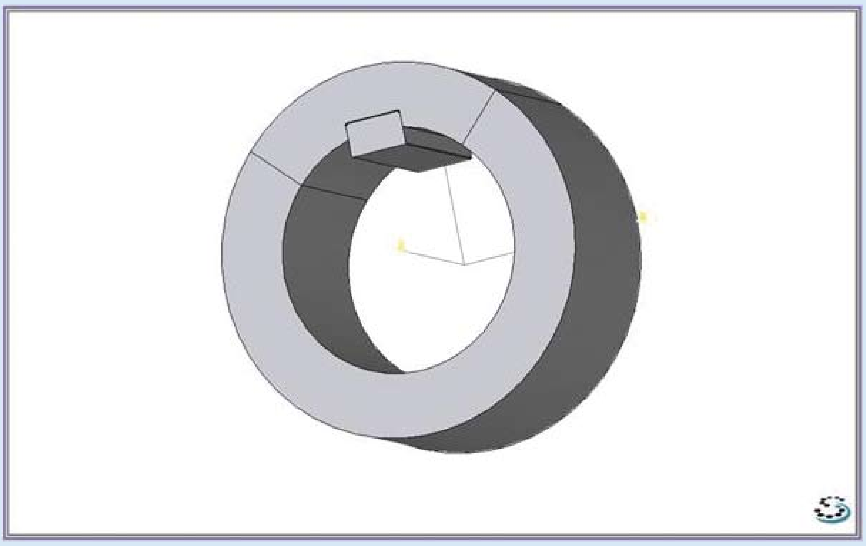

Fig. 5: Gear and key geometric model

For the analysis, a 660-KW wind turbine is used. The middle shaft in the gearbox was broken, and this caused the wind turbine to be out of action (Fig. 2).
This broken shaft is investigated macroscopically at the beginning and then it is modelled as a 3-D Model in order to search for the cause of its damage. Finally the life cycle is calculated as well.

The Macroscopic Investigation

The broken shaft in the gearbox of the 660KW wind turbine was put under the microscope (Fig. 3). The beach marks were seen clearly on the surface from the beginning, and this indicates fatigue damage. The beach marks even show that the fatigue cracks started on the sides of the keyway, and not on its corners.
After their initiation, the cracks have propagated along the cross section of the shaft till it fractured. The beach marks are seen to be lying very near to each other, and this indicates that the shaft was not rotating continuously and thus the fatigue fracture has developed over a very long time.
The surface, where the fracture occurred, is covered by an oxidized layer that was formed due to the opening and closing of the cracks while the shaft was rotating. This is due to the cyclic loading.
Finally, the lower surface of the keyway was seen to be deformed. This means, the key was moving along the keyway and this has led to the initiation of the cracks.
Due to these observations, it could be seen that the shaft has broken due to fatigue loads resulting from a unidirectional bending with less nominal stresses.

The FE Analysis

A simple 3D finite element analysis was carried out to test the method. We modelled a part of the shaft including the keyway. Furthermore, the key and gear were modelled to apply boundary conditions (Fig. 4, 5).
Concerning the boundary conditions that were applied on the model, some assumptions were made. Thus the input face of the shaft was considered rigid and constrained through its center in the lateral direction. The output side

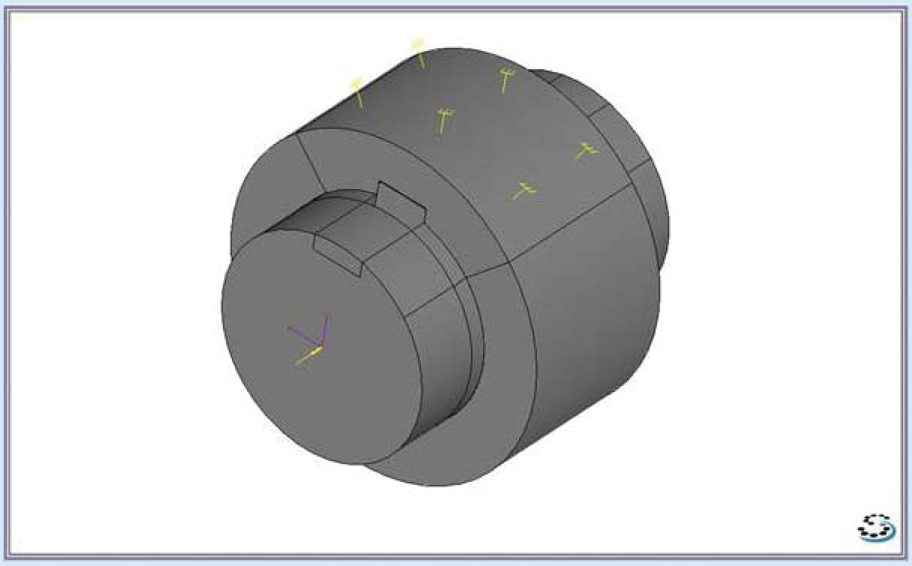

Fig. 6: Torque, reaction and input side support

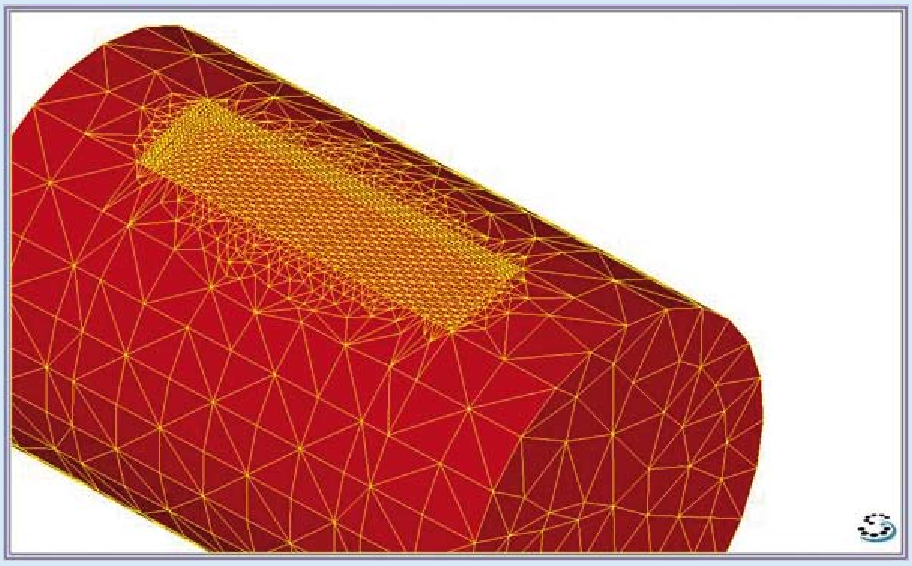

Fig. 7: Mesh on the shaft and on the keyway

face was also used as a rigid element and it was constrained in all directions. Furthermore, the rotations around vertical and lateral axes were blocked in order to make a simulation of a ball bearing. One of the faces of the gear was also blocked in the tangential direction (Fig. 6).
The gear and the key were meshed as one solid (Fig. 7). Contact conditions were imposed between the key and the keyway as well. The torque was imposed on the input part of the shaft.
The simulation was carried out using the FEM Program SAMCEF from SAMTECH, and the results from the stress analysis were read (Fig. 8).
Based on the results from the analysis, it was noticed that the maximum stress has exceeded the yield stress in some regions of the shaft. This could illustrate the reason why the shaft broke at the end.

Fatigue Procedures

In order to be able to calculate the lifetime, a history of the stress tensor at every node on the shaft must be obtained.
In this simple presentation, it was supposed that the stress due to torque varies according to a given time signal (Fig. 9). The bending stress varies with respect to the same time signal and it alternates with every rotation. Both stresses were supposed to act on the same material face, thus perpendicular to the shaft axis.
Based on the FE analysis, the main stresses were computed as two different load cases. After that, and based on the equation:

$$\sigma_{eq} = \sqrt{\sigma^2 + 3\tau^2}$$

The main stress was calculated for every time step as well. The time signal used for the simulation was 600 seconds. This has led to an equivalent stress curve with respect to time. After that, a rainflow procedure was used on this curve to obtain the equivalent cycles. For each cycle, the fracture damage for every load cycle was computed (Eq. 1).
The number of load cycles was found out by using either the Wöhler or the S-N curve (also known as Haigh-Diagram) for the material.

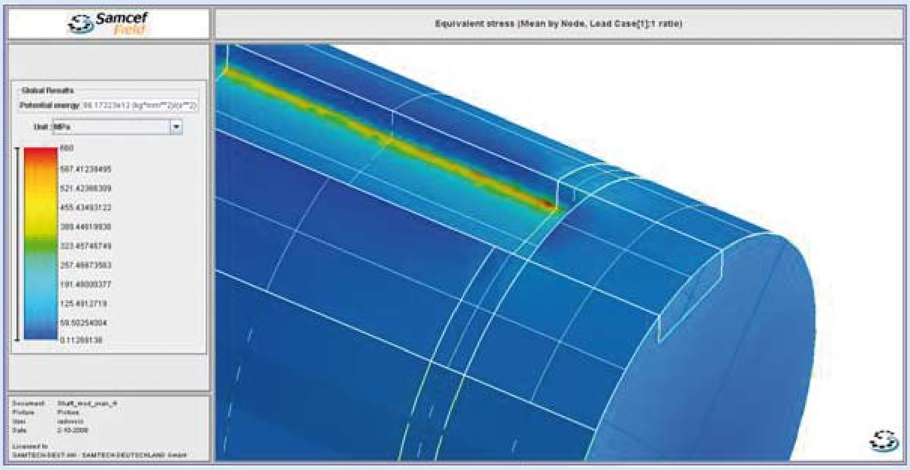

Fig. 8: Stress results

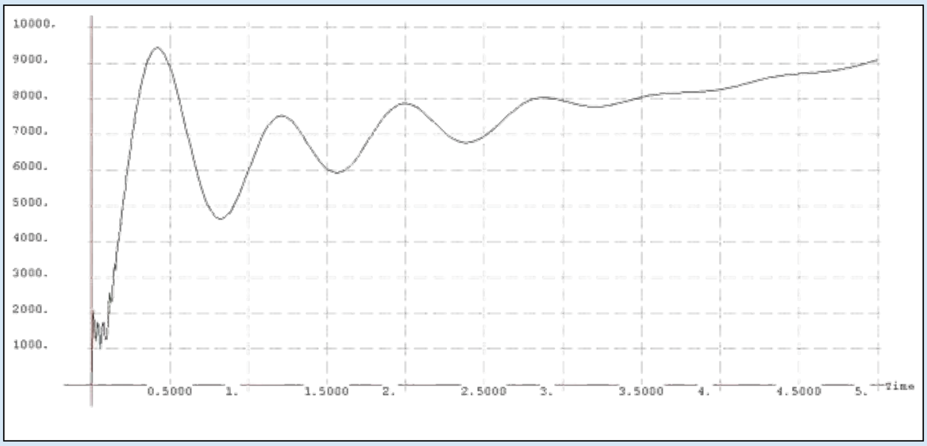

Fig. 9: Example of torque depending on time (SAMCEF Mecano).

Damages were then added together in order to obtain the total damage within the time signal used. Finally, the life time of the given part was predicted.

For the simple example of the shaft, the total damage was found to be 9.92e-6, which has led to **100792 cycles of 600s**. It must be emphasized that the number of cycles is inversely proportional to the damage (N =1/D). Since fracture occurs at D = 1, this means that a time of **three and a half year** will pass before a turbine will break down. Another assumption was made, stating that the turbine is running 200 days a year under the given cycle!

Of course, the simplifications in the small example have led to a too conservative value. Firstly, the friction should have taken a part in the torque load. Secondly, the use of S-N curves is conservative. Thirdly, a constant wind speed was assumed (16 m/s), while a wind speed ranging from 3 m/s to 25 m/s should be used. The time distribution of these series should be taken into account as well.

Some important remarks should be made. The plastic stress should lead to the use of Neuber plastic correction in order to predict strain. Thus ε-N curves should be used to obtain the damage.

Conclusion

This article shows that standard FE analyses are suitable to predict the life of a component. The limited analysis that was mentioned will be carried out in the future using full stress tensor combined with maximal shear plane method to predict the fatigue life of the whole component. The use of ε-N curves is strongly recommended as local stress might be plastic. Furthermore, a complete wind analysis must be carried out in order to estimate the total lifetime of the wind turbine.

Literature

[1] ASTM E 1150-1987, Standard Definitions of Fatigue, 1995 Annual Book of Standards, ASTM, 1995,

[2] ASM Metals Handbook Volume 11 – Failure Analysis and Prevention, 2002

[3] SAMTECH Deutschland, www.samcef.com

[4] ASM Metals Handbook Volume 11 – Fatigue and Fracture, 1996

[5] "Schadensuntersuchung an einer gebrochenen Getriebewelle", Bericht Nr. 967 aus dem Institut für Werkstoffanwendungen im Maschinenbau der RWTH Aachen, 2008

[6] Heege A., Radovcic Y., and Betran J. Fatigue load computation of wind turbine gearboxes by coupled structural, mechanism and aerodynamic analysis. DEWI Magazin, 28:60–68, 2006a.

Netzanbindung von WEA und Windparks
9 Sep. 2009, Bremen – Germany

Grundlagen der Windenergienutzung
1 Oct. 2009, Bremen – Germany

Wind Farm Planning and Risk Assessment
27 Oct. 2009, Istanbul – Turkey

KNOWLEDGE

DEWI's world-wide expert seminars are an excellent opportunity for companies that are involved in wind energy business to have their newly hired staff trained. Background knowledge and long-term practical experience of DEWI experts help to understand the complex contexts of wind turbine and wind farm layouts. Much more than isolated facts.

As one of the leading international consultants in the field of wind energy, DEWI offers all kinds of wind energy related measurement services, energy analysis and studies, further education, technological, economical and political consultancy for industry, wind farm developers, banks, governments and public administrations. DEWI is accredited to EN ISO/IEC 17025 and MEASNET for certain measurement and is recognised as an independent institution in various measurement and expertise fields.

DEWI
Quality by Know-how.

www.dewi.de