# Physics based methodology for wind turbine failure detection, diagnostics & prognostics

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## Abstract

The prediction of the time to failure for components within a wind turbine is becoming more important as a consequence of enlargement of the wind turbines and placing them offshore. These developments bring higher replacement and downtime costs with it in case of failure. Current failure prediction models are data driven or based on statistics, however both approaches are not sufficient to predict the failure accurately. This paper focuses on the actual loads acting on the system by taking into account how the component will fail or in other words the physics of failure. A generic physics of failure based methodology has been proposed that gives a step-by-step plan in which forces and operational data are taken into account. The methodology is divided into three parts: detection, diagnostics and prognostics. In order to validate the physics based methodology, a case study has been set up for one component and failure. SCADA and CMS data from three operating wind turbines are used to complete the case study. In this way both SCADA and CMS data are used in one method, where usually either SCADA or CMS is used. The degradation pattern and prediction of the time to failure are obtained. The case study has been proven that the methodology is useful in practice and shows the high potential of using this approach.

**Key words:** wind turbine, physics of failure, detection, diagnostics, prognostics, SCADA, CMS

## 1. Introduction

The wind energy production grew enormously in the past several years. In order to achieve this wind power growth, the industry mainly focuses on the development of offshore farms and larger wind turbines [1]. These developments have a negative impact on the operations & maintenance costs in case of failure. Larger wind turbines have higher replacements costs and wind turbines placed offshore are difficult to get access to. These accessibility problems has led to operations & maintenance costs which are up to five times that of onshore [2]. Underlying to these costs are the relatively high failure rates [3, 4]. Even though current wind turbines are designed to survive at least 20 years of operating, many components do not fulfill this requirement.

As a result, various failure prediction methods are available to reduce the operations & maintenance costs. Generally, the prediction methods can be classified into three categories: statistics, data driven and physics of failure. In order to show the differences between these methods, the P-F curve is used as shown in figure 1. The curve shows the component degradation over time, where the actual failure is assigned with F. P stands for potential failure and is the point where the method can say that the component is failing. In this case the P of each prediction method is included.

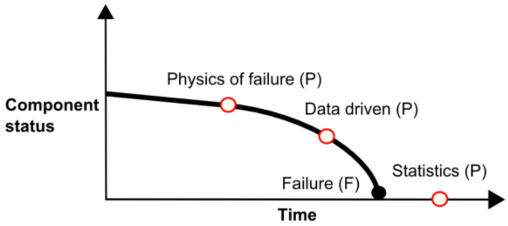

**Figure 1: P-F curve including prediction**

Statistics is a common used approach to relate the reliability gathered from experiences to the estimation of the next failure. A statistical method uses data not based on the actual condition of the component. It means that the prediction based on statistics can be anywhere in the graph and even can estimate the failure after the real failure. A data driven approach uses operational data like CMS and SCADA and monitors the behavior of several parameters. When a clear deviation of the parameters is noticed, an upcoming failure is detected. This often occurs just before failure. The limitation of this approach is that no link is made why and how the system fails. The physics of failure based method takes into account how the systems fails and determines the system condition using physical laws combined with operational data. This method can follow the curve most accurately and calculates the degradation from the time the system went operational.

The physics of failure based method has the advantage that potential damage can be prevented in an early stage and/or downtime costs of an upcoming failure can be reduced due to better planning of the component replacement. The lack of knowledge how far the method can reach and how it should be implemented in practice show the large potential for starting research. Gray et al. [5] en Stringer et al. [6] paid first attention to the physics based methodology, however no generally applicable method was presented and no detailed computations were included.

This paper presents a generally applicable physics of failure based methodology. All steps needed are explained and validated with a case study. Both SCADA and CMS data from operating wind turbines are included. In this way it can be investigated how two different monitoring systems can be combined in one method.

## 2. The physics of failure based methodology

A generally applicable physics of failure based methodology has been developed and shown schematically in figure 2. Two loads are considered in this methodology:

- Design load: failure mechanism load under designed operational circumstances.

- Additional load generator: additional load under undesigned operational circumstances.

When only the design load is taken into account, the component lifetime equals the design lifetime. For this reason, the additional load generator is needed to compute loads from external factors. The general idea behind this method is to relate the design load and additional load generator to the consumed and remaining useful lifetime for two cases: a reference case when only the design load is acting on the system and a failure case when the design load is influenced by the additional load generator. The final outcome is the remaining useful lifetime of a specific component. In order to achieve this, the methodology contains four key elements with the following meanings:

- Specification: Selection of the critical component and failure mechanism.

- Detection: Detection and quantification of failure sensitive parameters.

- Diagnostics: Establishment of consumed lifetime based on failure parameters from detection in combination with loads obtained from physical relations.

- Prognostics: Prediction of remaining useful lifetime based on diagnostics outcome.

The generic character is obtained by presenting the methodology in such a way that every step can be reproduced for every component and failure.

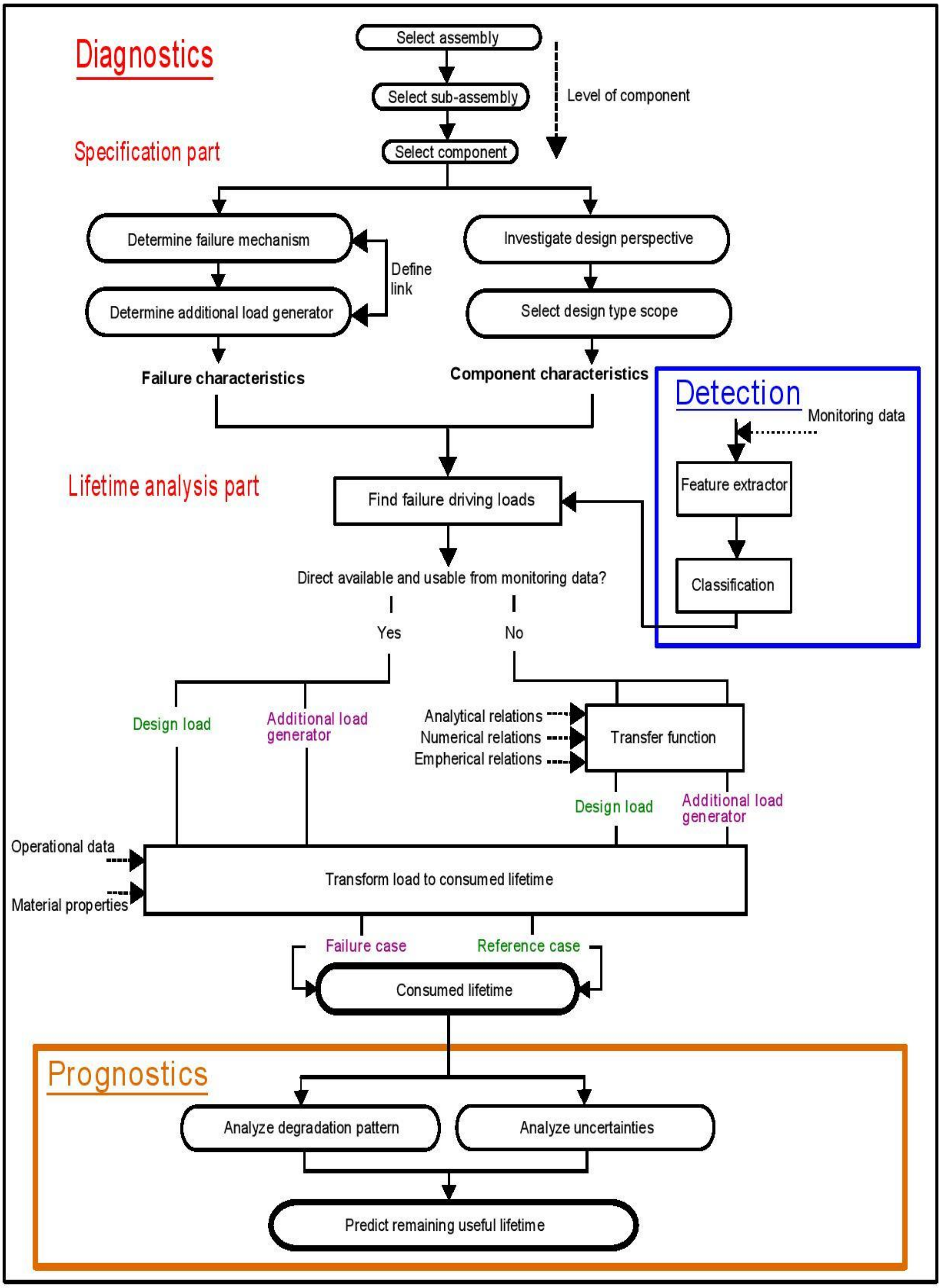


**Figure 2: Flowchart physics of failure based methodology.**

Moreover, the method is presented at tactical level, by which the user can proceed to a sufficient detailed level. All the steps in this methodology are explained by means of a case study. The most important part of the methodology is the transfer function, in which the loads are computed by means of physical relations.

## 3. Case study

The presented methodology is validated with a case study. SCADA and CMS data from three operating wind turbines within the same wind farm are used for this purpose. One component and one failure type are selected and run through all steps needed to end up with the remaining useful lifetime.

## 3.1 Specification

The specification part starts with the selection of the wind turbine component. A failure starts always on component level, therefore a component must be analyzed. Before knowing the component, an assembly is selected. A method proposed by Lee et al. [7] was used to identify a critical assembly.

The average failure frequency and downtime from several assemblies are shown in figure 3. This failure data is gathered from available researches: Land Wirtschafts Kammer (LWK) and Wind Measurement and Evaluation Program (WMEP) [8]. The graph is divided into four areas, each with its own maintenance strategy. The assemblies in Q4 are defined as critical due to the large downtime per occurrence. Predicting the components in this area can save many costs.

The gearbox scores relatively high for both researches and is selected as assembly. Subsequently, the helical gear is a failure sensitive component within the gearbox and for this reason selected as component. Bending fatigue of the gear tooth is selected as failure mechanism. Next, the additional load generator must be determined. The fatigue lifetime is very sensitive for changing contact area. One of the causes of this is misalignment of the generator shaft with respect to the gearbox shaft. Another contact pattern of the gear teeth results in different bending stresses. This means that the following load cases are considered:

- Reference case: load from bending fatigue during designed operational circumstances.
- Failure case: load from bending fatigue during misalignment.

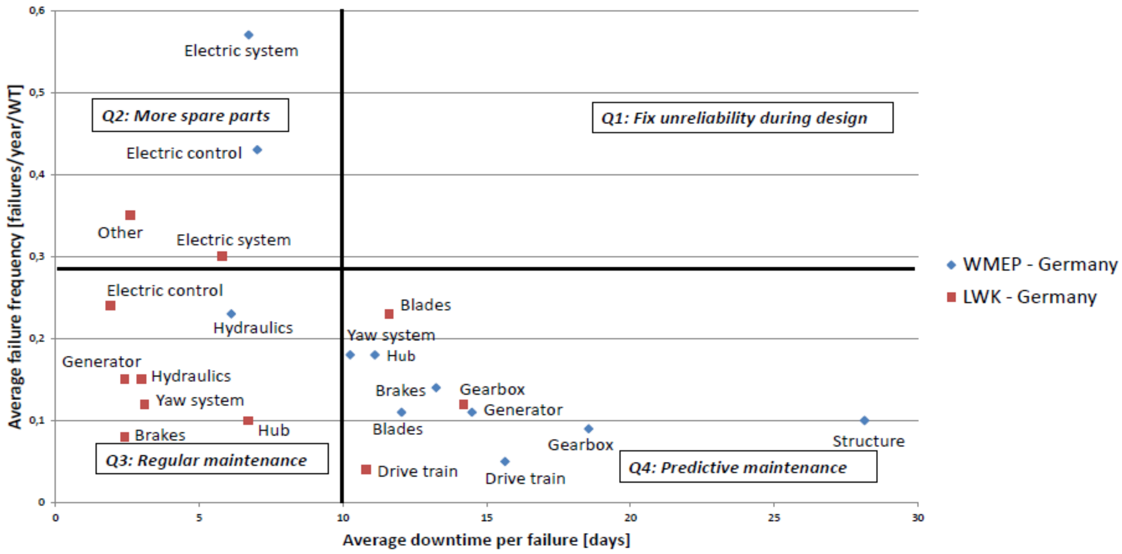


**Figure 3: Four quadrant chart. WMEP [8]: about 1500 WTs, majority < 1MW, 1991-2006, all onshore. LWK [8]: about 5800 turbine years, 1993-2004, all onshore.**

Reverse engineering is often applied to find out the design parameters. For this reason it is important to know from what perspective the component was designed. Another aspect is the design type scope, the question must be answered for which component types the methodology hold.

## 3.2 Detection

Detection is needed to monitor the misalignment size continuously and use this as input parameter for the lifetime calculations in the diagnostics part. Currently, once a year or less a laser measurement will take place to measure the misalignment size. This measurement gives not sufficient information when calculating the lifetime continuously. The damage sensitive parameter (feature extractor) and the transformation of this parameter to the size of misalignment (classification) are needed.

In order to find the feature extractor of misalignment a rotor dynamic model is reproduced from Redmond [9]. The rotor dynamic model includes the gear shaft, flexible coupling and generator shaft. By introducing a misalignment in the flexible coupling, a dynamic response was noticed. It turned out that the amplitude of the first and second excitation frequency get larger for larger misalignment values. This behavior is marked as feature extractor.

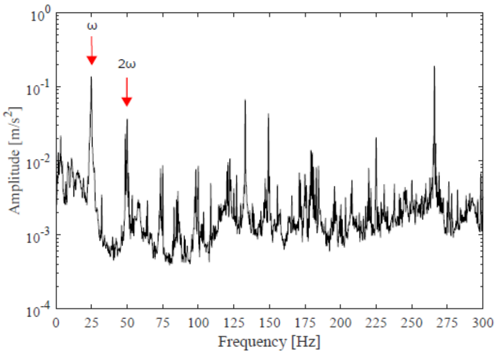


**Figure 4: Frequency spectrum.**

The found feature extractor is validated with a vibration condition monitoring system (CMS) data measured at the bearing besides the helical gear. The frequency spectrum gives the feature extractor. An example of a frequency spectrum obtained by means of a fast fourier transformation (FFT) of one measurement is shown in figure 4. In this case the generator shaft rotates at a nominal speed of 1500 RPM, which is equivalent to 25 Hz (ω) for the first excitation frequency and 50 Hz (2ω) for the second excitation frequency.

During 2012 up to and including 2015 every week the amplitudes of one FFT are plot as shown in figure 5 and 6. Every year the wind turbines undergo a maintenance inspection. During this inspection the misalignment is measured with a laser device and after that the misalignment is adjusted to an acceptable size. All the misalignment values from all measured wind turbines are classified into low, medium and high.

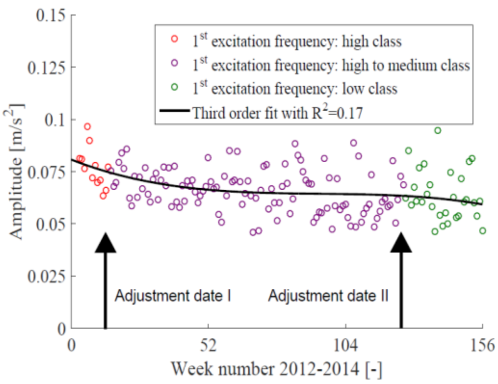


**Figure 5: First excitation amplitude pattern 2012-2014.**

During the adjustment dates an amplitude jump should be expected. However, no clear deviations can be seen. This means that the feature extractor used cannot detect misalignment, thus also no classification is possible. Probably another phenomenon has a higher impact on the first and second excitation frequency than misalignment. The misalignment values obtained from laser measurement are used for further calculations.

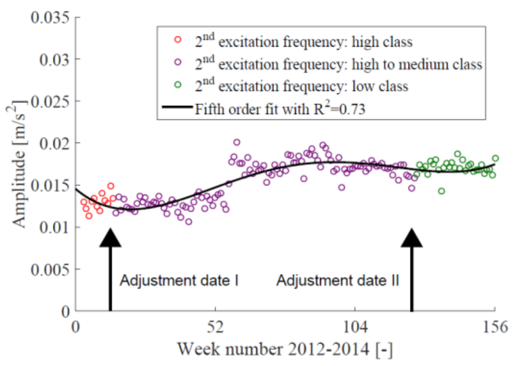


**Figure 6: Second excitation amplitude pattern 2012-2014.**

### 3.3 Diagnostics

The diagnostics part begins with finding the failure driving loads. For the reference case, the force acting on the tooth and the gear geometry are the load parameters which result in a bending stress. The International Standardization Organization (ISO) [10, 11] provides a widely used design standard to calculate the bending stress by taking into account the force and dimensions of the gear tooth. For the failure case the contact area of the interacting gear teeth changes, which results in an additional bending stress. It is not possible to calculate this in an analytical way, therefore a finite element method (FEM) is used to compute the influence of misalignment. In summary the loads are calculated as follows:

- Reference case: ISO standard
- Failure case: ISO standard + FEM

The transformation to the acting loads on the systems is called transfer function. The ISO equation for the gear root bending stress S is given as:

$$S = \frac{F_t}{bm_n} Y_F Y_S Y_\beta Y_B Y_{DT} Y_\varepsilon K_A K_V K_{F\beta} K_{F\alpha} S_F \qquad (1)$$

where
$F_t$ - transmitted force [N].
$m_n$ - normal module. This module is defined as the gear pitch diameter divided by the number of teeth [m].
b - gear face width [m].
$Y_F$ - form factor, which takes into account the influence of the tooth shape [-].
$Y_S$ - stress correction factor, which takes into account the stress amplifying effect of the root fillet [-].
$Y_\beta$ - helix angle factor, which compensates that the bending moment of a helical gear at the tooth root is less than the corresponding values for spur gears [-].
$Y_{DT}$ - deep tooth factor, which adjust the tooth root stress for high precision gears [-].
$Y_\varepsilon$ - contact ratio factor, which takes into account that the load is distributed over multiple teeth [-].
$K_A$ - application factor, which adjust the load for incremental gear loads from external sources [-].
$K_V$ - dynamic factor, which compensates load increments due to internal dynamic effects [-].
$K_{F\beta}$ - face load factor, which takes into account the effect of uneven load distribution due to misalignment or deflections. This factor is 1 for the reference case and will be calculated for the failure case [-].
$K_{F\alpha}$ - transverse load factor, takes into account non-uniform distribution of transverse load due to profile modifications, manufacturing accuracy and running-in [-].
$S_F$ – safety factor, factor according to guidelines of international wind turbine gearbox standards [-].

A FEM model is built which include the gear geometry and misalignment value. Since the misalignment value cannot be detected by means of CMS, a three year average misalignment value obtained from laser measured is used.

It is possible that the data systems provide the correct failure parameters directly. In this case SCADA is used and does not deliver the failure parameters directly. In order to calculate the bending stress, the force is the only parameter that changes during operation. This force can be calculated by means of the power relation:

$$P = T \bullet \omega \qquad (2)$$

where
P - power, measured from 10 minute averages SCADA [W].
T - torque, dividing this parameter by the gear pitch diameter then the force is obtained [Nm].
ω - generator speed [rad/s].

The number of cycles gives the number of gear mesh interactions and is calculated as:

$$n_i = \omega_i \bullet t \qquad (3)$$

where
$n_i$ – number of cycles [-].
$\omega_i$ – generator speed [RPM].
t – time span [min].

Not every wind speed is suitable to take into account for the force calculations. As shown in figure 7, very low wind speeds result in the fact that the energy consumption of the wind turbine itself is higher than the production. The loads on the blades become too high for very high wind speeds and then the blades are turned in such a position that it does not produce energy anymore. This means that only the range between 2 m/s and 24 m/s is taken into account.

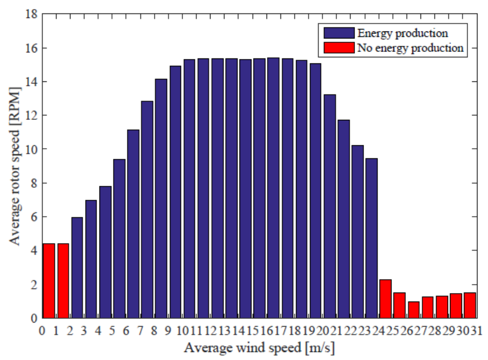


**Figure 7: Average wind speed and rotor speed.**

As a consequence, the load spectrum can be obtained as shown in figure 8. The wide spread attracts attention, which is caused by the daily wind speed variation. The large peak around 550 MPa is because then the maximum power and maximum generator speed is reached, which is the favorable situation of the wind turbine. For the failure case, the obtained misalignment value results in a bending stress increase of 6%.

All the loads acting on the system are known, so the fatigue resistance of the material is needed. Again an ISO standard [12] is used to calculate the fatigue resistance which is expressed through a S-N curve. In this case, case hardened steed is used, which is a common wind turbine gearbox material.

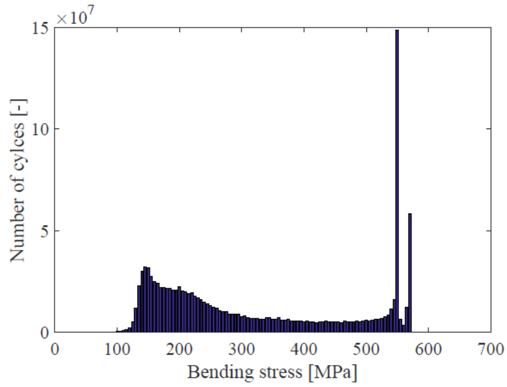


**Figure 8: Load spectrum.**

The final step is to calculate the damage of the gear tooth. This is done by applying the Palmgren-Miner rule [12]:

$$D = \sum_{i=1}^{p} \frac{n_i}{N_i} \qquad (4)$$

where
D - damage index, failure will occur when D = 1 is reached [-].
P - blocks of constant amplitudes.
$n_i$ - actual number of cycles for constant bending stress amplitude i [-].
$N_i$ - maximum number of cycles to failure for constant bending stress amplitude i [-].

The consumed gear tooth bending fatigue lifetime is presented in table 1.

**Table 1: Consumed lifetime.**

| Wind turbine | Reference case (D) | Failure case (D) |
|---|---|---|
| 1 | 0.106 | >1 |
| 2 | 0.132 | >1 |
| 3 | 0.089 | 0.917 |

## 3.4 Prognostics

The prognostics part analyzes the degradation pattern and want to answer whether this pattern is representative for future degradation. The degradation pattern over three years of the reference case is shown in figure 9. Remarkable is the large difference between the wind turbines. The main reason for this is the difference in wind speed. Wind turbine 1 experienced in average 10% higher wind speeds than wind turbine 3. Also the difference between the reference and failure case is significant, which shows the essence of taking into account the additional load generator.

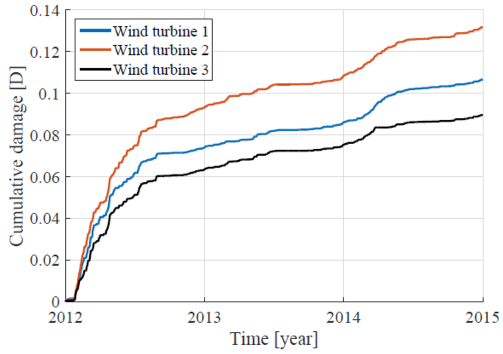

**Figure 9: Degradation pattern**

Even though the degradation pattern does not behave linearly, the total degradation over three years is used to calculate the remaining useful lifetime. The prognostic part could be added with weather forecast model to improve the uncertainty of the prediction.

**Table 2: Remaining useful lifetime.**

| Wind turbine | Remaining useful lifetime reference case (years) | Remaining useful lifetime failure case (years) |
|---|---|---|
| 1 | 25.3 | 0 |
| 2 | 19.7 | 0 |
| 3 | 30.4 | 0.3 |

## 4. Discussion

The presented methodology is a comprehensive tool and needs much information about the design and operation. This means that much effort is required for the user, who often is not the designer, and is dependent on the availability and usefulness of the information required. Since this is a very detailed method, the parameters must be known very accurately, otherwise large deviations will be seen.

The time at which the critical failure can be determined and the physical methodology can be started is often different for the design load and additional load generator(s). The critical components are known from experience and the design loads can be calculated from the point that the component went operational. This in contrast to the additional load generator, this load is often known after a certain period of operation.

The focus of this study was on the difference between failure and reference case. The results show large differences between the wind turbines itself and between the reference and failure case. Assumptions are made to take into account the misalignment, which need more attention to come with better results. However, the potential of using the additional load generator has been shown.

## 5. Conclusions

A generic physics based methodology has been proposed by dividing the method into a detection, diagnostics and prognostics part. The influence of loads on a component for a specific failure can be examined. The practical implementation has been proven by the failure of the helical gear tooth due to bending fatigue.

The application showed that the detection part needs a data type like CMS whereas the diagnostics part needs SCADA. Even though the data types are used separately, the outcome can be combined to improve the failure predictability.

Even though only one component and failure are used as application and it was not possible to detect misalignment continuously, the large potential of using this method has been shown.

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