Impact of Carrier Position and Surrounding Stiffness on Planet Carrier Bearing Loads

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Abstract. Bearing failures in wind turbine gearboxes still occur which results in downtimes of wind turbines and hence increase the levelized costs of energy. One of these affected bearings is the generator sided planet carrier bearing as it is still subject to unexpected fatigue related failure modes. To reduce the occurrence of these failures a methodology is presented to determine the most influencing wind loads on this planet carrier bearing using the finite element method. Therefore a 1 MW wind turbine drive train model is used and the influence of several parameters influencing the load distribution of the bearing are investigated. These parameters are the surrounding stiffness of the bearing seat, the influence of the individual components of the wind loads and the circumferential position of the planet carrier. The results show that a realistic surrounding of the bearing seat leads to an unequal circumferential load distribution while the bearing itself is less loaded than using the generic surrounding of the bearing seat. The different components of the wind loads do not have much impact on the load distribution while the force acting on the rollers change. The different positions of the planet carrier also do not affect the circumferential load distribution.

1. Introduction and Objective

Wind turbines become more and more competitive because the levelized costs of energy (LCOE) decrease continuously [1] but it is still the aim to reduce the LCOE. Downtimes of wind turbines increase the LCOE and thus should be reduced as much as possible. One of the reasons for the downtime is the failure of the main gearbox in the mechanical drive train as it is used in 65 % of the erected wind turbines [2]. A large part of these gearbox failures result of bearing failures [3]. The high speed shaft bearings are affected by smearing and rehardening which result from high slipping conditions. These slipping conditions result from high gradients in the rotational speed and torque. The intermediate and low speed shafts are affected by wear related failure modes like fretting corrosion which is caused by ring creep and fatigue failures like axial cracking, surface flaking and pittings.

These failures can be ring creep which leads to ring fractures or fatigue failures for the planet carrier bearings or smearing and rehardening for the high speed shaft bearings [4, 5]. Fatigue related failure modes are e.g. axial cracking, surface flaking and pittings [6].

While the high speed shaft bearings can be replaced while the gearbox is still on the tower, a change of the planet carrier bearings is demanding and leads to an exchange of the gearbox. For this a spare gearbox, a crane and calm winds are necessary. Thus it leads to loss of AEP and increase of OPEX and subsequently increases the LCOE. A possible way to reduce the downtime is by using a more reliable load prediction of the planet carrier bearings by knowing the effect of the known wind loads on the bearing loads and its circumferential load distribution. The planet carrier bearing is selected as a result of a study among the project partners who represent companies among the wind industry.

One approach is, to use the finite element method to determine the influence of the surrounding stiffness of the bearing seat on the roller load distribution which subsequently allows to improve the design of the gearbox housing. The finite element method also allows to give a statement on the influence of different wind loads and different rotational positions of the carrier. By knowing the
individual roller load it is then possible to calculate the Hertzian pressure and subsequently the individual lifetime per position on the circumference using more detailed tools.

2. Modelling

The finite element model for the individual roller load calculation was built for a 1 MW wind turbine nacelle with four point suspension bearing concept (see Figure 1). It consists of the main shaft, the main bearing, the machine carrier and a three-stage gearbox. The gearbox is built up of one planetary stage and two helical gear stages (PSS).

![Figure 1. Finite Element Model of 1 MW wind turbine mechanical drive train](image1)

The main bearing is build using two spherical roller bearings. The gearbox is connected to the machine carrier using two torque arms with an elastomer and two ball joints. This allows the gearbox to move under gravity and under wind loads.

The planet carrier which is used in this gearbox is shown in Figure 2. The two tapered roller bearings supporting the planet carrier and the planet pin are shown. The load transfer from planet carrier to the ring gear is modelled using forces on both components.

The rollers of the planet carrier bearings are modelled using previously calculated nonlinear roller stiffness maps [7]. This allows to achieve the load acting on each roller separately. Subsequently the effect of the local surrounding stiffness can be determined. The different surroundings are shown in Figure 3.

![Figure 2. Planet carrier model](image2)

3. Investigated parameters

The different parameters that affect the bearing load and its circumferential load distribution are the surrounding stiffness of the bearings seat, the wind loads and the influence of different positions of the planet carrier.
3.1. Surrounding stiffness of the bearing seat
The first investigation focusses on the different circumferential load distribution due to different surrounding stiffness. For this investigation a generic and a realistic surrounding of the bearing seat are used (see Figure 3). In the following these stiffness are referred to as the surrounding stiffness.

![Figure 3. View of a) generic and b) realistic surrounding of generator sided planet carrier bearing seat](image-url)

The generic gearbox design has a very equally distributed stiffness around the bearing seat due to its solid plane. The realistic surrounding uses several columns to support the bearing seat. By this it allows a better flow of oil and saves weight but leads to an unequally distributed stiffness around the bearing seat and thus might affect the load distribution on the bearings' rollers.

The planet carrier and its bearings and the planet pins are shown in Figure 2. The planet carrier is supported using two preloaded tapered roller bearings. The preload is applied at the rotor sided bearing, shifts the planet carrier slightly to the back and hence preloads also the generator sided bearing.

3.2. Wind loads
The wind loads are torque, which is used to generate power, bending moments and thrust and lateral forces at the hub of a wind turbine. They act on the main bearing and subsequently the torque arms and the planet carrier bearings.

The acting loads are nominal loads for this 1 MW drive train for a wind speed of 16 m/s and 16 % turbulence. These values were chosen as 16 m/s is the nominal wind speed for the turbine and as it is IEC class Ia 16 % turbulence is a possibility. [8] The loads acting on the main shaft result from a multi-body simulation of the turbine including rotors and tower with a generated windfield. The loads are shown in Table 1.

|                  | Thrust $F_x$ | Lateral Force $F_y$ | Lateral Force $F_z$ | Torque $M_x$ | Bending Moment $M_y$ | Bending Moment $M_z$ |
|------------------|-------------|---------------------|---------------------|--------------|----------------------|----------------------|
|                  | 79.5 kN     | -0.813 kN          | -85 kNm             | 304 kNm      | -41 kNm              | 9.8 kNm              |

3.3. Different circumferential positions of the planet carrier
Another parameter which affects the load distribution in the bearing is the position of the teeth contact between the three planets and the ring gear which leads to a deformation of the housing and thus to a change in the surrounding stiffness of the bearing seat. Hence the effect of different positions of the planet carrier is investigated using a rotation of 40 degrees and 80 degrees as this allows a statement on the load distribution per evolution of the planet carrier. The positions are displayed in Figure 4.
4. Results

The initial load distribution of the bearing is influenced by the bespoke parameters: The surrounding stiffness of the bearing seat, the assembly situation which affects results in preload and gravitational loads. When the turbine is generating electrical power the bearing is also influenced by the wind loads and the circumferential position of the planet carrier.

4.1. Influence of the surrounding stiffness of the bearing seat

Hence the first investigation focuses on the difference in the load distribution due to the surrounding stiffness. Therefore a generic geometry of the planetary bearing seat was designed. In comparison a realistic geometry was used. Both of these geometries are shown in Figure 3. The resulting bearing load distribution is shown in Figure 5.

The bearing in the generic gearbox is higher loaded than the bearing with the realistic gearbox design. Thus the cumulative load for the generic gearbox is also higher. The bearing in the generic design is loaded with 61 kN while the bearing load of the realistic design is only 52.6 kN. This reduction of 14% increases the lifetime of the bearing by 50% according to ISO TS 16281 [9] (see equation 1).

\[ L_{10r} = a_1 \cdot a_{ISO} \cdot \left( \frac{C}{P} \right)^p \]

This equation calculates the lifetime with a ten percent probability of default \( L_{10r} \) using the lifetime coefficients \( a_1 \) and \( a_{ISO} \). \( a_1 \) is dependent on the demanded probability of failure and equal to 1 for ten percent probability of failure. \( a_{ISO} \) is dependent on the viscosity and the contamination of the oil and
the fatigue load limit of the bearing. $a_1$ and $a_{ISO}$ are the same for both calculations. $C$ describes the basic dynamic load rating and is bearing specific while $P$ describes the equivalent dynamic bearing load. $p$ is the life exponent which is 10/3 for a tapered roller bearing. [9] The calculated load difference per roller is shown in Figure 6.

![Figure 6](image6.png)

**Figure 6.** Difference in roller load due to different surrounding stiffness

The difference in the roller loads varies between +1 kN and −0.39 kN per roller.

While the generic geometry leads to very equally distributed roller loads, the realistic geometry leads to peaks where the bearing seat is surrounded by cast iron. This is shown in Figure 7.

![Figure 7](image7.png)

**Figure 7.** Uneven load distribution due to elastic surrounding

The generic geometry leads to generally higher bearing loads, because the geometry is much stiffer than the realistic surrounding.

On one hand the realistic surrounding of the bearing leads to a longer lifetime of the investigated planet carrier due to lower loads. On the other hand the unequal load distribution may lead to ring creep when the pretension is reduced over the lifetime.

4.2. Loads acting on the gearbox

The second part of this investigation focusses on the effect of the different components of the wind loads acting on the bearing.

The first loads acting on the gearbox over its lifetime are preload and gravity. The effect of the preload on the bearing is an ideally equally distributed load for each roller to prevent it from slipping which leads to smearing and rehardening. [5] The load distribution on the planet carrier bearing after preload and gravity is shown in Figure 8.
The load is unequally distributed due to the different surrounding stiffness as discussed previously. The shift in the bearing load to the first quadrant results from the tilting of the gearbox under gravity (see Figure 9).

**Figure 9.** Tilting of Gearbox due to Gravity

When the turbine is producing electrical power the wind loads act on the turbine. Thus the effect of the different components of the wind loads on the bearing load distribution is investigated by applying the different components separately at the hub of the turbine. The resulting bearing load distribution is shown in **Figure 10**.
The thrust load $F_x$ has the highest influence on the generator sided planet carrier bearing. In this case the rotor sided bearing is less loaded as it is not able to take the thrust load due to the arrangement. $F_x$ has the most influence on the resulting load distribution in comparison to the other wind load components because of the four point suspension that is used in this drive train.

4.3. **Influence of different circumferential positions of the planet carrier**

The planet carrier rotates with the same speed as the rotor. The planets are in contact with the ring gear which also changes the stiffness over the circumference. Hence this effect is investigated for different positions of the planet carrier. For this investigation the wind loads from Table 1 are used.

The resulting loads differ between 1.5 % and 9.6 % in dependency of the applied wind loads between the 40° and the 0° position. The roller loads between 80° and 0° position differ up to 13.5 %. These differences are a result of the different stiffness of the ring gear with housing and planet carrier around the bearing seat. When the teeth contact is at the 0° position the surrounding deforms very equally, while the other positions on the circumference lead to more unequal deformations of the housing which can also be seen in the load distribution (see Figure 11).

**Figure 10.** Resulting load distributions due to different wind loads

**Figure 11.** Bearing load due to different circumferential positions of the planet carrier

Figure 12 shows the mentioned deformations under different positions of the teeth contact between planet and ring gear.
Nevertheless the reason for the generally higher loads in the 40 and 80 degrees positions still need to be investigated.

5. Conclusion and Outlook

The generator sided planet carrier bearing load is affected by different parameters which have an effect on the load distribution as well as on the height of roller loads.

The main roller load of the generator sided planetary bearing comes from the preload. The bearing is also subject of wind loads as are the other components in the gearbox. In this four-point suspension system the gearbox input loads are much lower than the acting wind loads which explains the limited influence of the wind loads on the bearing loads.

The surrounding stiffness of the bearing seat has a big influence on the load distribution in the bearing which is up to 12\% of the load peak. But while the roller load changes over the circumference, the cumulative load is slightly lower for the realistic surrounding of the bearing seat which results in a 50\% higher calculated lifetime. Knowing this may allow to use smaller planet carrier bearings and thus save money.

The circumferential position of the planet carrier leads to changes in the peak loads of up to 13\% which is also a result of the surrounding stiffness due to different deformations.

This more unequal load distribution leads to higher local stresses in the bearing ring and might affect fatigue related failure modes. This will be investigated using more sophisticated tools which give a statement on the lifetime of the locally stressed area. These investigations will be performed for a three point suspension drive train where higher loads are expected at the planet carrier bearings as the gearbox input loads increase using the shown method.

Additionally these investigations for the four and the three-point suspension will be validated using the 4 MW test bench at CWD in Aachen with the shown drive train. The roller loads will be measured using strain gauges on the circumference of the outer ring of bespoke bearing. The effect of different loads will be measurable as the loads can be applied separately.

It is expected that the loads on the bearing will be higher in the three-point suspension.
6. Acknowledgements
The authors would like to thank the Federal Ministry for Economic Affairs and Energy for the support. They also thank their project partners, who provided the equipment, insight and expertise which contributed greatly to this successful joint research.

Supported by:

[Image: Federal Ministry for Economic Affairs and Energy on the basis of a decision by the German Bundestag]

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