Category: 10 Testing

Emulation of Interface Stiffness for Full Scale Tests on Wind Turbine Blade Bearings

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Summary

Typical damage modes of rotor blade bearings are mainly related to oscillating movements with small angles under high loads. Under these conditions, the lubrication film between rollers and raceways of the bearings tends to collapse, which causes metal to metal contact and surface induced damage. Since established calculation methods for lifetime prediction are unsuitable for wind turbine blade bearings, full-scale test rigs are developed to close this particular knowledge gap. To generate realistic boundary conditions for bearing tests, the stiffness of the interfaces such as blade and hub have to be investigated. Because of the disadvantages of using real components within these test rigs, a possible solution is to use substitute components which aim to emulate the interface behaviour of the real components. This paper presents a method to derive a substitute of a rotor hub based on a FE-model of the virtual IWT7.5-164 research turbine. The results indicate that it is feasible to match the interface stiffness of the considered rotor hub with a corresponding substitute with only minor limitations.

1. Blade bearing of wind turbines

Blade bearings of wind turbines are subjected to adverse operating conditions. They need to withstand high bending moments while standing still or oscillating at low speeds with small angles (Figure 1).

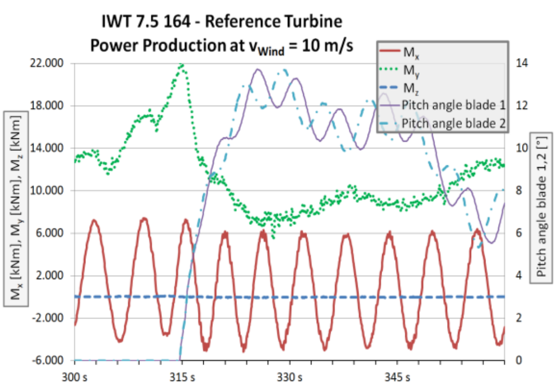

Figure 1: Pitch bearing loads using individual pitch control (IPC) of the IWT-7.5-164 reference turbine [1], [5]

Resulting damage modes from these operating conditions include roller contact fatigue, edge loading, core crushing, ring fractures, fretting and false brinelling. Fatigue is the most relevant damage mode for continuously rotating bearings with full film lubrication conditions, whereas surface induced damage modes like wear, fretting and false brinelling originate during standstill under dynamic loading or small oscillating movements, when the lubrication film between rollers and raceways collapses [1]. Established calculation methods such as ISO 281 are unsuitable for non-elasto-hydrodynamic lubrication conditions and do not allow a reliable prediction of rotor blade bearing lifetime [2]. At the current state of knowledge, it is not possible to predict the development of surface induced damages as it has been summarized by Schwack et al. [3].

Based on the described damage modes and their effects, blade bearing test rigs are designed and operated in order to reproduce realistic and repeatable loading conditions for the bearings. The loading of the bearing depends on the stiffness of the interface components, such as hub and blade. Therefore, it is necessary to emulate their stiffness behaviour. On one side, rotor blades which are made from composite materials have a relatively low stiffness compared to steel components. Due to their design with spar cabs and shear webs, unsymmetrical load distributions are applied to the bearing. On the other side, rotor hubs which are large hollow bodies of casted steel alloys have usually a relatively thin wall thickness compared to their size. Therefore, they are not able to serve as a rigid support for the bearing. As a rigid support would be needed to ensure a more symmetric load distribution in the bearing, rotor hubs react to different load cases with complex deformations. This directly affects the load distribution in the bearings provoking unfavourable operating conditions.

If the stiffness behaviour of hub and rotor blade interface is disregarded, the resulting load on the bearing is changed, which also leads to a different, non-realistic distribution of the rolling element contact forces.

Figure 2 shows an exemplary load situation for the rolling elements of a typical four-point roller bearing as it is the standard type for blade bearings in wind turbines. As it can be observed, there is a large variation in the resulting reaction force vectors, showing the non-symmetrical loading of the roller elements.

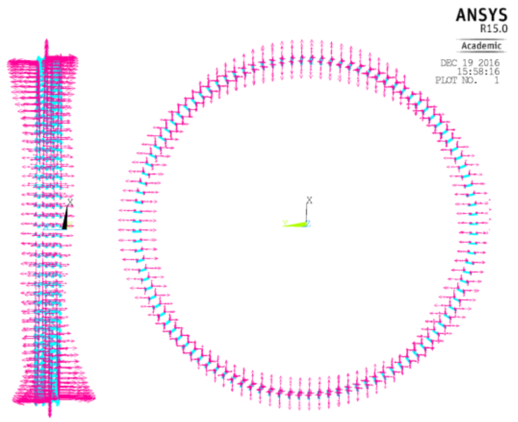

Figure 2: Exemplary load distribution for a 4-point roller bearing in ANSYS. The magenta coloured vectors illustrate the reaction forces of the rolling elements with respect to the principal axis.

To validate blade bearings and further develop suitable methods for lifetime prediction for oscillating bearings, test rigs need to emulate stiffness conditions for the hub and blade interfaces close to the real loading scenarios.
Fraunhofer IWES is going to build a test rig for blade bearings up to the 10MW wind turbine class. The objective is to include hub and blade substitutes to emulate the stiffness as accurately as possible. Additionally, a six DOF load application hexapod facilitates stressing the bearings with real time series of wind loads in a range of up to 50MNm bending moments.
Based on the presented approach, this paper shows how a rotor hub substitute for the mentioned test rig is derived from a model of a real rotor hub.

2. Design Approach of a Hub Substitute

Based on the deformations of the IWT 7.5-164 rotor hub FE-model, an approach for designing a hub substitute for the emulation of the real hub stiffness has been developed. The analysis is carried out based on the GL's blade coordinate system [4].
Since the stiffness of an object is the extent to which it resists deformation in response to an applied force, deformation values are used to characterize the stiffness properties of the hub.
The first step is the analysis and processing of the deformation data followed by the geometric parameterisation of the substitute to match the desired deformations. For the analysis, the extreme load case for the hub is considered which causes the maximum deformation. Therefore, the rotor hub is analysed via FEM to calculate the deformations of the blade bearing flange. A certain number of nodes on the outer contour of the flange are used to obtain the deformation values (Figure 3).

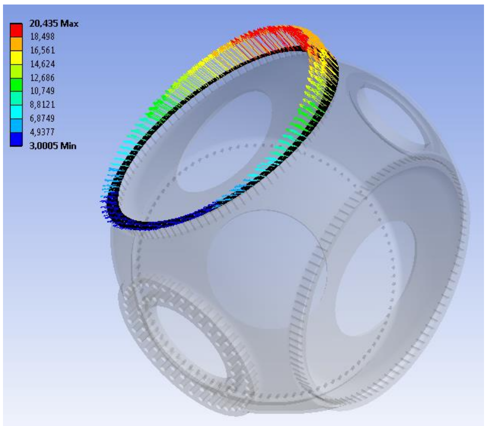

Figure 3: Exemplary FE-analysis with deformation vectors at the observed flange. Values in [mm]

The resulting deformation of the hub flange surface for different load cases has an influence on the loading of the bearing. However, the overall deformation of the hub is too large to be recreated with a substitute, which is desired to be less complex and smaller than the original hub. Therefore, it is necessary to separate this surface deformation from the overall deformations of the hub. Since the z-direction is the main direction of the force vectors caused by the flap-wise and edge-wise blade bending moments, which are the highest occurring loads, the deformation in the z-direction is of particular interest for this design approach.

The calculated deformations from the FEM are filtered in order to extract the amount of deformation which does not affect the surface deformation of the flange.
By creating a regression plane, it is possible to derive the curvature in form of residual deformations. This regression plane will later serve as the new reference plane, where the residual deformation corresponds to the total z-deformation of the substitute component. If the approximation of the real deformation with the regression plane is well chosen, it is possible to adjust a substitute component to this residual deformation. However, this requires the definition of a basic geometry for the substitute. As a first approach, a steel disc with variable cross sections serves as the origin for an analytical parameterisation process. The procedure is to split the disc in equally distributed sections and to attribute to each one a custom stiffness, which matches the desired deformation for a certain load case. These sections have geometric parameters, such as length, thickness, etc. which affect their area moment of inertia and bending behaviour under a load. Figure 4 shows a cross section with the geometric parameters and the corresponding force F.

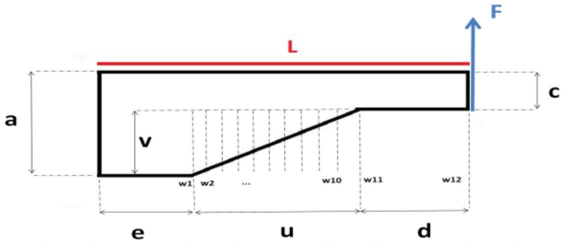

Figure 4: Sketch of a single developed substitute cross section with parameters and corresponding load

To define the corresponding stiffness for the cross sections, a MATLAB program was written, which searches possible combinations of parameters. Afterwards a number of single parameterized cross sections are used as sampling points to match the deformation. Subsequently these sections are combined to form a body using cubic splines. This method provides a smooth transition between the cross sections. Finally, the analytical and the FEM deformation of the substitute are compared to the deformation calculated using the original hub FEM model.

3. Results

The mathematical regression plane has been calculated to derive the residual deformation, which is the deviation from this encountered plane. Figure 5 shows exemplarily how the regression plane for an accurate approximation fits into the deformation values of the outer blade bearing flange contour of the hub.

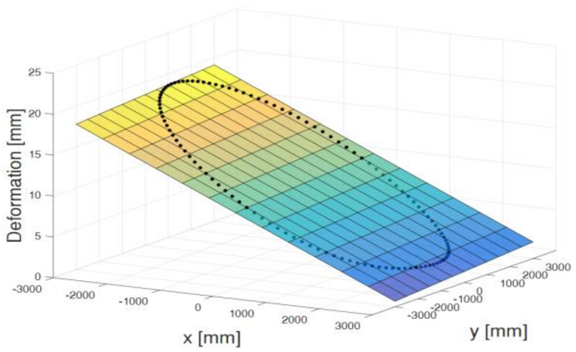

Figure 5 Exemplary regression plane and residual deformation

The deviation of the deformation values from the regression plane are small compared to the initial deformation, but they exhibit an irregular behaviour, which does not correspond to the usual deformation caused by a bending moment, as it can be observed in Figure 6.
The reason for this irregular behaviour is that the plane regression has distorted the deformation characteristic of the hub flange. This is caused by the high grade of approximation. To solve that problem, it is necessary to rotate the regression plane in the two angles so that the deformation behaviour matches the characteristic of a bending moment in a better way (Figure 6).

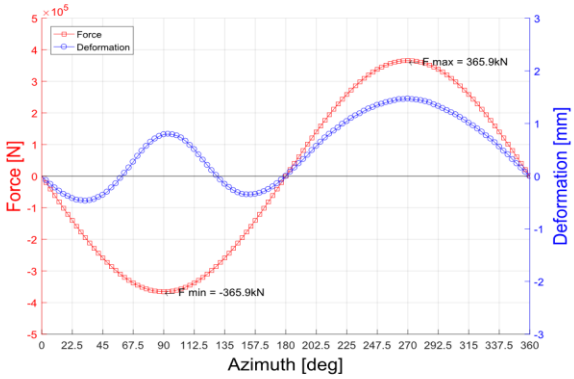

Figure 6: Exemplary residual z-deformation and corresponding force

With the described custom regression plane, it is possible to restore the deformation characteristic due to bending. Bending is characterised by a neutral axis separating compression and tension zones as shown in Figure 7.
The progression of the negative deformation results from the high stiffness area near the main shaft flange interface, whereas the maximum positive deformation occurs close to the top, where the hub has the lowest stiffness.

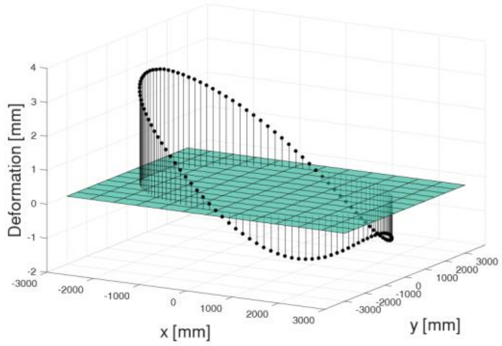

Figure 7: Residual deformations from custom regression plane

Since the deformation obtained from the regression is relatively small and lies within the elastic zone of steel, it is possible to find a suitable substitute component. This deformation is the input for the parameterisation of the substitute. Based on this, the cross sections of the substitute can be adapted to the input deformation by geometric parameterisation.
The result of this fitting process is shown in Figure 9. As it can be observed, the theoretic deformation behaviour caused by the M_y-bending moment (in flap-wise direction) fits the input deformation with high accuracy.
Based on the analytic parameterisation, a corresponding geometry is created using CAD, as it can be seen in Figure 8.

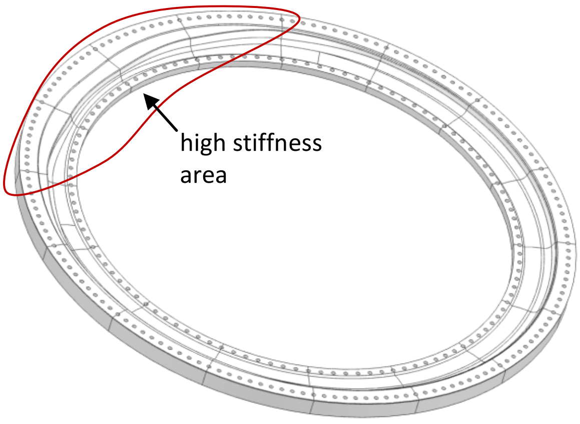

Figure 8: Resulting geometry of the rotor hub substitute. The red marked high stiffness section corresponds to the hub flange area near the rotor hub flange.

Figure 9 also shows that the FEM calculated deformations for the CAD-geometry match the analytically estimated deformation with good accuracy. The maximum deviation is located around 90° where the derived substitute is not stiff enough to resist against the highest occurring compression forces. Since the deviation of the FEM-deformations to the input deformation is less than 5%, it is considered to be sufficient for this stage of design.

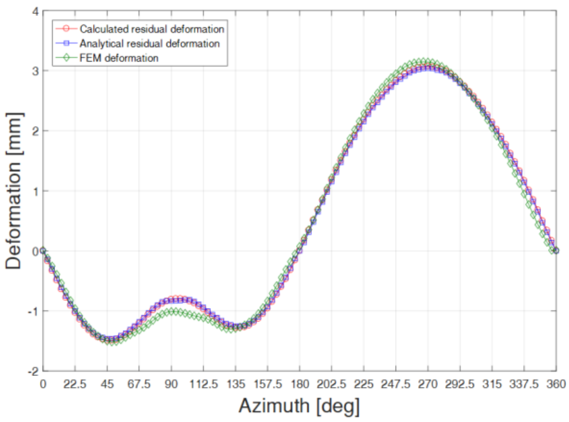

Figure 9: Comparison of input, calculated and FEM-deformations of the hub-substituting component for the maximum flap wise bending moment

4. Conclusions

In this paper, a method has been presented to create a rotor-hub substitute which aims to emulate the interface stiffness of a real rotor hub in a blade-bearing test. It has been shown that the input deformations of the real hub can be processed by using a mathematical regression to separate the surface deformation of the flange from irrelevant deformation of the hub. The processed deformations have been used to parameterize a steel disc as origin for a first design approach, which worked successfully. The analytical approach for this substitute geometry has been verified using FEM, finding that the fitting is within +/- 5% and therefore very accurate.

However, a minor discrepancy between the analytic and the FE calculated results for the maximum stiffness area of the substitute has been discovered, showing that there is potential for further optimisation.

The iteration of adapting the substitute to the desired deformation without exceeding the maximal tensile stress limit turned out as challenging.

It was discovered that the disc shape of the substitute tends to have an angular inclination of the surface under loading, which results from the one-sided suspension of the disc. This causes a torsional effect to the bearing, which is to reduce with further design optimisation of the substitute component.

Furthermore, the ovalisation occurring in the x-y-plane has not been emulated so far, but is subject for the ongoing work.

In summary, the present work has shown that the geometric parameterisation of the hub substitute to one load direction determines the possibility to adapt it to another load direction because there can be only one stiffness response for each single corresponding force. This makes necessary to choose which particular deformation characteristic is to be emulated by the substitute with highest priority for a blade bearing test.

5. Acknowledgements

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6. References

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