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Free Contact Angles in Pitch Bearings and their Impact on Contact and Stress Conditions

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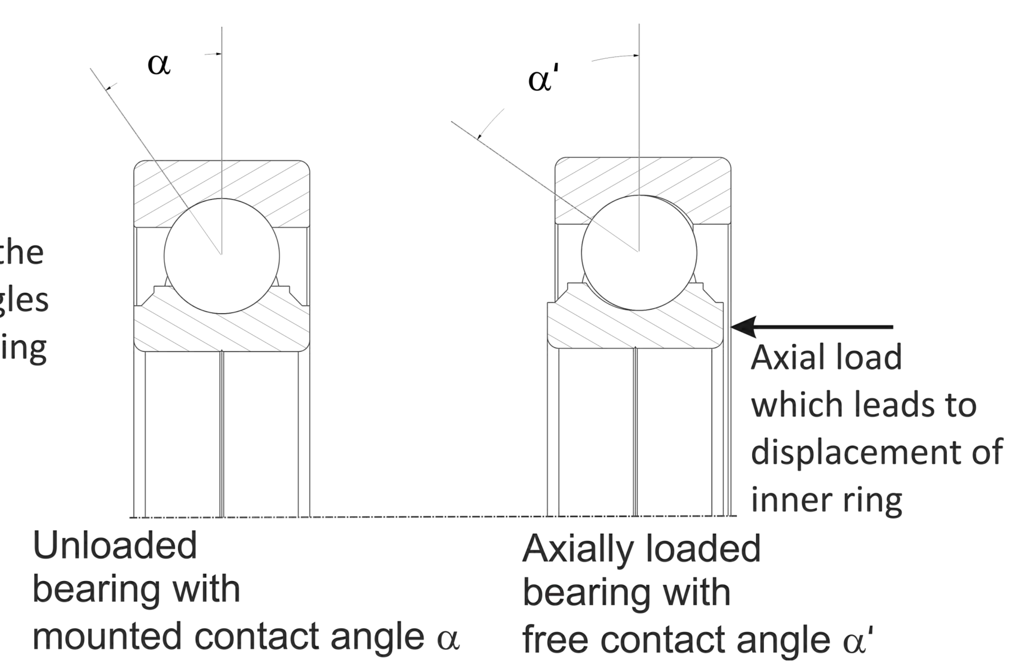
Abstract

The pitch bearing which connects hub and blades, allows the required oscillating movements of the blade, to reduce the lift and drag coefficient and thus control the power and load of the wind turbine. During the service life of about 20 years of a turbine, the pitch bearing is loaded dynamically. In a double row four-point contact bearing which is often used for this application, the described loads lead to radial and axial displacements between the inner and outer ring. These displacements lead to contact angles which vary from the mounted contact angle which is given by the supplier of the bearing. These divergent angles are called free contact angles. Deviations from the mounted contact angle can lead to reduced fatigue life of the bearing and increased wear arising from the modified contact behavior. Under axial loads and bending moments the contact area shifts perpendicular to the rolling direction. In the worst case the contact area is truncated by the geometry of the bearing which leads to concentrated and comparatively high stresses. There are actually no published results how the free contact angles differ from the mounted contact angle in a pitch bearing of a modern wind turbine.

Schematic representation of the occurence of free contact angles with an axial loaded ball bearing



Objectives

This analysis shows how the free contact angles of a double row four-point contact bearing with a mounted contact angle of 45 degree of a modern wind turbine behave during the service life and the effects of contact angle variations on the stress distribution.

For the analysis the bearing geometry and loads of a state-of-the-art reference wind turbine IWT 7.5 MW are used [1].

Methods

In the following the data set which was used for the analysis is presented. This includes the bearing geometry, loads and assumptions of the FE-Model.

Bearing, hub, blade, and loads

The following table gives an overview about the dimensions. Furthermore the hub was designed in detail to consider the stiffness of the component.

Bearing Geometry

Parameter	*Size*
Pitch diameter	4690 mm
Ball diameter	80 mm
Contact angle	45°
Number of balls per row	156
Number of rows	2

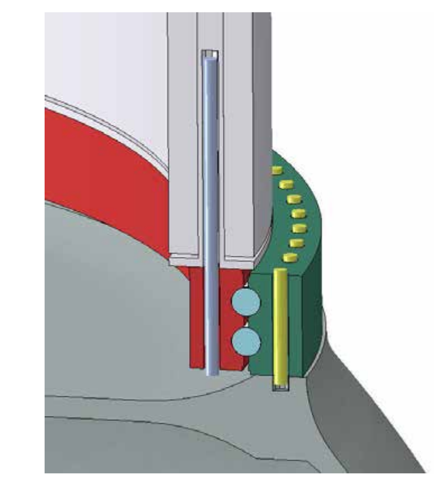
CAD-Model of the hub

Equivalent load $P = 0.75 \cdot F_{radial} + F_{axial} + \frac{\sqrt{M_{flap}^2 + M_{edge}^2}}{d_{pitch}/2}$

FE-Model

The respective FE model includes the blade bearing, the hub and one simplified blade root. The parts are connected by bolted joints which are modelled as well. A detailed evaluation of the rolling element loads and contact angles under special load cases necessitates an accurate consideration of the raceway kinematics in the model. Hence, the rolling elements are not simplified at all and consist of hexahedral, deformable elements. These show a surface-to-surface contact to the raceways and show all necessary degrees of freedom to represent the behavior of the real slewing ring. The contact angles and rolling element loads adjust themselves, until the equilibrium of the whole system is fulfilled. Even the truncation of the pressure ellipse can be recognized by the simulation due to this detailed modeling strategy. A submodel of the global system is used to evaluate the Hertzian pressure, the sub-surface stress and the deformation of the rolling element contacts. This model is based on a plastic material behavior and contains different material definitions for the hardened layer and the core material.

Overview of the FE-Model



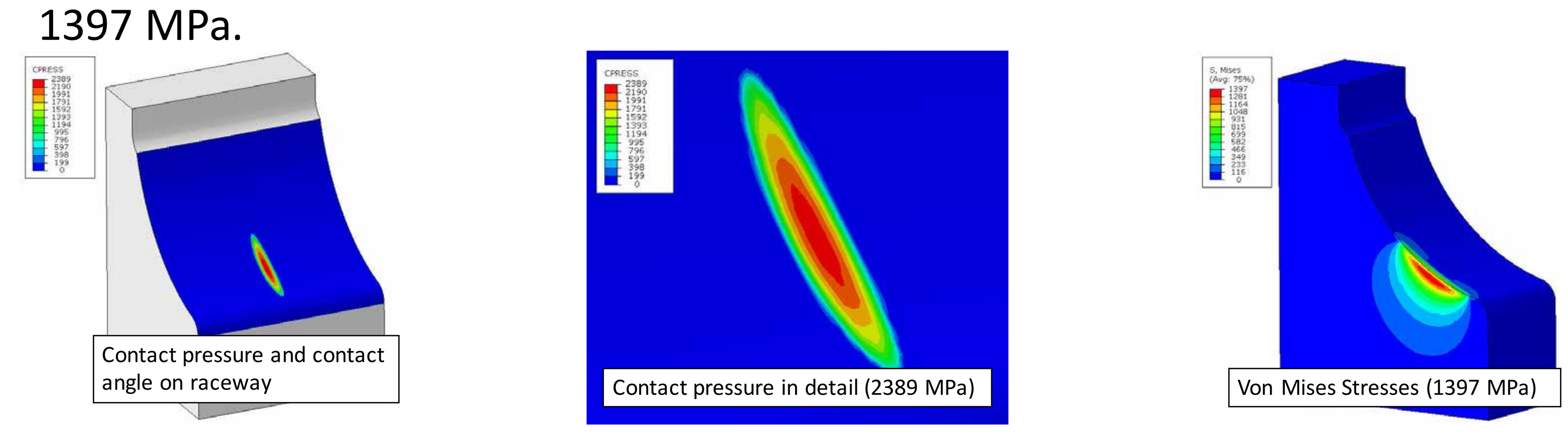
Adjacent structure in FE-Model

Results

On this poster, the results of one load case for normal operating conditions and one for extreme conditions will be presented. A more detailed presentation of the results will be given in the full paper.

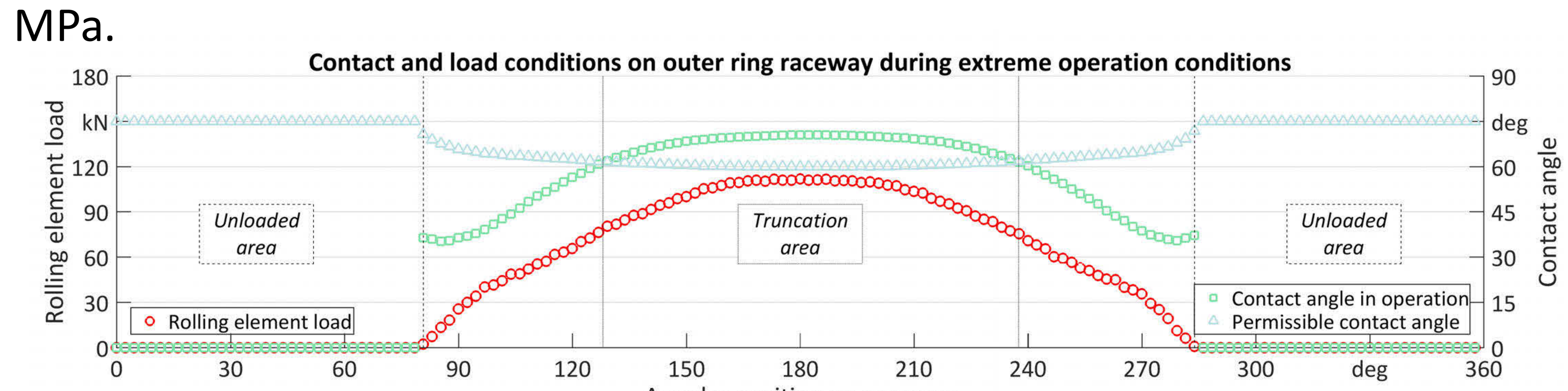
Operating conditions $(v_{wind} = 11\,\frac{\text{m}}{\text{s}}; P = 9150\ \text{kN})$

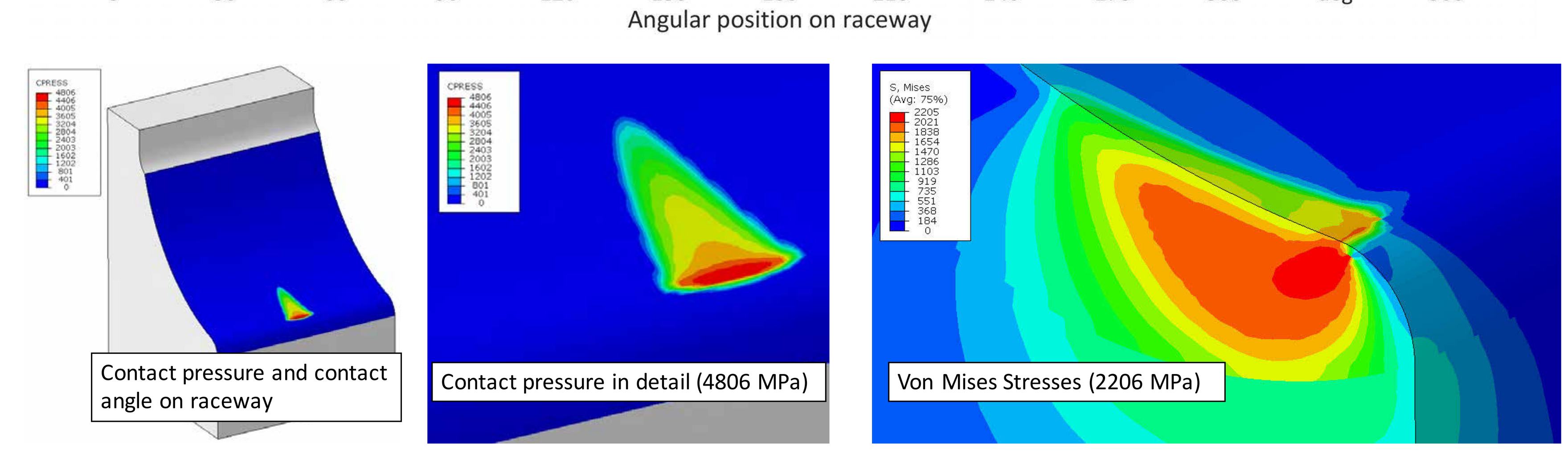
During operating conditions no truncation effects can be seen. Anyhow, the contact angle arise up to 65 degree, what can be seen in the figure below. The surface pressure is 2389 MPa. The von Mises stresses for this load case are 1397 MPa.



Extreme load conditions $(v_{wind} = 50\,\frac{\text{m}}{\text{s}}; P = 21340\ \text{kN})$

The graph below shows the rolling element load, the contact angle in operation during extreme loads and the permissible contact angle over the peripheral angle of the raceway of the outer ring. The permissible contact angle was calculated using the bearing geometry and the element load and represents the contact angle at which truncation occurs. It can be seen that truncation occurs in this load case. The truncated pressure ellipse is illustrated in the figures below. Thus, the von Mises stresses arise up to 2205 MPa.





Conclusions

The analysis showed that the contact angle which occurs in real applications, called free contact angle, is in value most of the time greater than the mounted contact angle which is given by the bearing supplier. Furthermore, under extreme loads, the contact angle grows so sharply that the Hertzian contact area is truncated by the bearing geometry. The analysis has also shown that these truncated pressure lead to stress spikes which can reduce the service life of the bearing significantly.

The free contact angle is not considered in most guidelines and standards. For a more accurate product design, with view on service life, it could be useful to consider the free contact angle. Furthermore, it could be meaningful to change the bearing design in large-bearing applications, like the given example.

References

1. IWES Wind Turbine IWT -7.5-164, *Specification*