Improvement of the pitch bearing load distribution by shape-optimized stiffening plates

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Abstract. Due to a small cross-sectional dimension compared to their diameter, pitch bearings have a comparably low stiffness. Therefore, surrounding structures strongly influence the internal load distribution of the bearing. By the use of stiffening plates, the load situation of pitch bearings can be positively affected. With a gradient-based optimization procedure regarding contact force and contact angle distribution, an optimized plate shape is determined. This optimized shape provides at the same time a more equalized load distribution, while minimizing the effect of contact ellipse truncation.

1. Introduction and Motivation

As the central component of the pitch adjustment system, the pitch bearing is highly loaded by aerodynamic forces and bending moments. Due to the small cross-sectional dimensions in relation to the bearing diameter, pitch bearing rings have a low structural stiffness. Because of this, the surrounding structures have a major influence on the internal load distribution of the bearing [1]. A widely used bearing type for pitch bearings are four-point slewing bearings. The dependency of this bearing type to surrounding structures has been analyzed in several studies [2,3,4,5,6], which have shown that considering these adjacent structures results in an irregular contact force distribution but also to strongly varying contact angles. Thereby the combination of locally high contact forces and contact angles can lead to contact ellipse truncation effects [5] that damage the raceways’ edges and lead to raceway wear and tear [6]. Standard design guidelines [7,8] do not take these relationships into account, what in the case of pitch bearings has to be seen critical. As over the last decades the rotor size and with that the pitch bearing loads constantly increase [9], it can be assumed that the risk of pitch bearing failure rises if these relationships are not considered in the pitch bearing design process.

One solution to overcome this problem, is the use of so called stiffening plates. Previous studies have shown that stiffening plates, which are mounted between the pitch bearing and the rotor blade can positively influence the contact force and contact angle distribution [10,11]. Even though it is common knowledge that the load distribution can be adjusted with this component, this is not done in the sense of finding a stiffening plate design, which provides an optimum load distribution. In this work, an optimization algorithm is presented, which is used to develop a shape-optimized stiffening plate for a double-row four-point slewing bearing. The stiffening plate’s design ensures a balanced inner load distribution of the pitch bearing and at the same time minimizes the risk of contact ellipse truncation.
2. Methodology

The development of a shape-optimized stiffening plate that improves the load distribution of a pitch bearing initially presuppose a suitable analysis model. The requirements for such an analysis model include the calculation of the internal load distribution of the rotor blade bearing based on given shape parameters of the stiffening plate and a given load collective. Additionally, the computational cost of such an analysis model has to be kept to a minimum, since an optimization generally involves several iteration loops. In further proceeding manageable performance indicators are derived that adequately describe the internal load situation of the bearing. These performance indicators are used to formulate the actual cost function of the optimization problem. To find an optimum set of shape parameters, a gradient based optimization algorithm is used.

2.1. Analysis model

In former studies a Finite Element Method (FEM) model of a pitch bearing arrangement has been developed (Figure 1a) [11]. As already described, the consideration of surrounding structures is a prerequisite for the calculation of a realistic load distribution. Therefore FEM is a suitable method for modelling the pitch bearing arrangement. Unfortunately, in comparison to analytic models, higher computational costs are inherent to FEM models. Surrogate models of the actual ball contacts are used to reduce these computational costs (87.3 million to 1.2 million elements). Within this surrogate model, contact pairs of the pitch bearing has been replaced by nonlinear traction spring elements [2]. Even though this surrogate model is used instead of a computationally intensive contact simulation, because of the special geometrical design of the surrogate model, it is still possible to evaluate the contact force and contact angle of the replaced contact. The developed FEM model of the pitch bearing arrangement was extended by a parameterized stiffening plate (Figure 1b)

![Figure 1](image.png)

**Figure 1** FEM model of pitch bearing arrangement: a) Resulting load distribution of loaded pitch bearing [11] b) Shape parameters of stiffening plate design

The parameters used to define the shape of the stiffening plate include the positions of radial support points \((d_1, \ldots, d_8)\) from which the inner contour of the plate is derived. This allows a local radial stiffening of the bearing ring and compensates for irregularities in the bearing load distribution. Also
the plate thickness is varied in order to adjust the stiffness not only locally but also over the full circumference. The stiffening plate’s shape thereby is defined by the vector

$$\vec{x} = \begin{pmatrix} \vec{d} \\ t \end{pmatrix} = \begin{pmatrix} d_1 \\ \vdots \\ d_8 \\ t \end{pmatrix},$$  \hspace{1cm} (1)

$$d_i \in [d_{min}, d_{max}], \ t \in [t_{min}, t_{max}]$$

Load cases are defined by cutting forces and cutting moments, which are applied to a reference point, which is coupled to the cut surface of the rotor blade root. A single static load case is given by the vector

$$\vec{P} = \begin{pmatrix} \vec{F} \\ \vec{M} \end{pmatrix} = \begin{pmatrix} F_x, F_y, F_z, M_x, M_y, M_z \end{pmatrix}^T$$  \hspace{1cm} (2)

and a load collective is described by the set

$$L = \{ \vec{P}_1, \vec{P}_2, ..., \vec{P}_p \}$$  \hspace{1cm} (3)

The output values of the analysis model, which are used for further processing, consist of resulting contact forces

$$F(\vec{x}, \vec{P}) = \{ f_1, f_2, ..., f_N \},$$  \hspace{1cm} (4)

and corresponding contact angles

$$A(\vec{x}, \vec{P}) = \{ \alpha_1, \alpha_2, ..., \alpha_N \}$$  \hspace{1cm} (5)

2.2. Contact ellipse truncation

Since contact ellipse truncation can drastically reduce the lifetime of four-point slewing bearings, the occurrence of this effect has to be investigated during post-processing. For this purpose, relationships have been described by Krynke [6], which specify the necessary conditions for the occurrence of this effect (Figure 2).
According to Krynke [6], contact ellipse truncation occurs when the condition
\[ \alpha + \alpha_{\text{ellipse}} \geq 90^\circ - \alpha_{\text{edge}} \] (6)
is met, where \( \alpha_{\text{ellipse}} \) can be approximated as
\[ \alpha_{\text{ellipse}} \approx \frac{2a}{d_{\text{ball}}} \] (7)
and the semimajor axis \( a \) of the contact ellipse is calculated as
\[ a = \frac{3}{\sqrt{E\sum \rho}} \xi^3 F, \]
\[ \xi^3 (1 - \nu^2) \]
\[ \sum \rho = \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22}, \] (8)
whereby the variables \( E \) and \( \nu \) describe the Young modulus and the Poisson's ratio of the raceway and ball material and the coefficient \( \xi \) is interpolated from look up table data [12]. Using these formulations given by Krynke [6] the following equation is derived
\[ f_{\text{truncation}} = \frac{s(\alpha)^3 E \sum \rho}{3 \xi^3 (1 - \nu^2)}, \quad s(\alpha) = \frac{d_{\text{ball}}}{2} \cdot \left[ (90^\circ - \alpha_{\text{edge}}) - \alpha \right] \] (9)
Equation (9) indicates the minimum force required to cause contact ellipse truncation for a given contact angle. E.g. for forces exceeding this threshold, the semimajor axis of the contact ellipse increases so much that it extends beyond the edge of the raceway. Figure 3 visualizes this relationship.

Figure 3 Occurrence of contact ellipse truncation as a function of contact force and contact angle
To quantify the severity of this effect, the contact ellipse truncation factor

\[ \mu_i = \frac{f_i}{f_{\text{truncation}}(\alpha_i)} \] (10)

is derived. As the ratio of the actual contact force and the minimum force to cause contact ellipse truncation, this parameter can be used to consider the risk of truncation effects into the optimization procedure. For truncation factors smaller than 1 contact ellipse truncation does not occur, whereas for values greater than 1 the conditions for this effect are met. To consider this relationship within the optimization procedure, the truncation factor is added to the cost function, which is basically defined as the standard deviation of the contact forces. Thereby, for a single load case, the cost function is defined as

\[ f_{\text{cost}}(\mathbf{x}, \mathbf{P}) = \sqrt{\frac{1}{N} \cdot \sum_{i=1}^{N} |f_i - \bar{F}|^2} + a \cdot \left[ \max (\{ \mu_i | i \in [1, N] \}) \right]^b \] (11)

while \( a \) and \( b \) describe scaling parameters to adjust the weighting of the truncation factor. With that formulation, the optimization procedure aims at an equalized load distribution, while considering the severity of occurring truncation effects. On the basis of a load collective, the cost function is defined as

\[ F_{\text{cost}}(\mathbf{x}, L) = \sqrt{\sum_{j=1}^{p} f_{\text{cost}}(\mathbf{x}, \mathbf{P}_j)^2} \] (12)

2.3. Optimization procedure

The optimization algorithm uses a gradient based procedure. Based on local gradients, searching directions are calculated. Thereby the shape parameters, which define the inner contour and the thickness of the stiffening plate, are optimized in different loops. This is done because distance parameters can take continuous values between an upper and lower limit, whereas only discrete values are permissible for the plate thickness. Therefore different formulations are necessary for an iterative search for solutions of these parameters. This approach is illustrated in figure 4.
The calculation of the search directions for the inner contour of the plate is done using the formalism

$$
\hat{S}_d^n = \frac{-\hat{\nabla} F_{\text{cost}}(\hat{d}^n)}{\|\hat{\nabla} F_{\text{cost}}(\hat{d}^n)\|}, \quad \hat{V} F_{\text{cost}}(\hat{d}^n) = \left( \frac{\partial F_{\text{cost}}(d_i^n)}{\partial d_i^n}, \ldots, \frac{\partial F_{\text{cost}}(d_k^n)}{\partial d_k^n} \right) \quad (13)
$$

For the current parameter set, the partial derivative is calculated for each parameter and the resulting vector is normalized to get the new search direction. Thereby the partial derivatives are calculated by the finite difference approximation

$$
\frac{\partial F_{\text{cost}}(d_i^n)}{\partial d_i^n} := \frac{F_{\text{cost}}(d_i^n + \Delta d) - F_{\text{cost}}(d_i^n)}{\Delta d} \quad (14)
$$

In Case of the plate thickness, as there is only one parameter to consider, the calculation of the search direction can be simplified to

$$
S_t^m = \text{sgn} \left( - \frac{\partial F_{\text{cost}}(t^m)}{\partial t^m} \right) := \text{sgn} \left( - \frac{F_{\text{cost}}(t^m + \Delta t) - F_{\text{cost}}(t^m)}{\Delta t} \right) \quad (15)
$$

After finding a search direction, a set of cost function values is calculated, which correspond to multiples of a step size $\Delta d/\Delta t$ in this direction, starting from the current parameter values. Possible solutions for a better performing parameter set are calculated as

**Figure 4** Optimization Algorithm: a) Flow chart of optimization algorithm b) Intermediate results of optimization procedure
\[ D^n = \{ \tilde{d}^n + S^i_{\tilde{d}} \cdot i \cdot \Delta d | i \in [0,1,...,k] \} \]  
\[ T^m = \{ t^m + S^j_{t} \cdot j \cdot \Delta t | j \in [0,1,...,l] \} \]  

(16)  

(17)  

The best performing parameter set is identified by

\[
\tilde{d}^{n+1} = \arg \min_{\tilde{d} \in D^n} F_{cost}(\tilde{d})|_{t=t^m}
\]  

(18)  

\[
t^{m+1} = \arg \min_{t \in T^m} F_{cost}(t)|_{d=\tilde{d}^n}
\]  

(19)  

and accepted as the solution of the iteration loop. The iteration is repeated as long as the termination condition

\[
\varepsilon_{n/m} = \left( \frac{F_{cost}(d^n, t^m) - F_{cost}(d^{n+1/n}, t^{m/m+1})}{F_{cost}(d^n, t^m)} \right) \leq 0.005
\]  

(20)  

is met. The optimization procedure ends if no significant improvement can be achieved, whether the thickness or the inner contour of the stiffening plate is changed.

3. Results

Although the computational cost of the FEM model has already been significantly reduced, the implementation of the optimization algorithm requires a relatively large number of calculations. Therefore, the optimization procedure has been carried out for a simplified load collective consisting of three single load cases derived from multibody simulations of the corresponding wind turbine. The underlying loads are listed in table 1.

| \( L \) | \( F_x \) | \( F_y \) | \( F_z \) | \( M_x \) | \( M_y \) | \( M_z \) |
|---|---|---|---|---|---|---|
| \( \tilde{p}_1 \) | 0 | 0 | 0 | 0 | 10 MNm | 0 |
| \( \tilde{p}_2 \) | 0 | 0 | 0 | 7.07 MNm | 7.07 MNm | 0 |
| \( \tilde{p}_3 \) | 0 | 0 | 500 kN | 0 | 7 MNm | 0 |

Table 1 Load collective of optimization procedure

Two load cases contain pure bending moments that differ in their orientation but have the same magnitude as the maximum bending moment from the load calculations carried out. A third one contains an acting axial force, which is equal to the maximum axial force of the load calculations from which the simultaneously acting bending moment was also taken. The simplified load collective is therefore to be regarded as being representative for maximum loads, but certainly do not reflect the complexity of the load combinations that could potentially be acting.

The intermediate results of the optimization can be taken from figure 4b). In total eight iterations were needed to find a solution for the optimization procedure, in which six were necessary to optimize the inner contour and two to optimize the thickness of the stiffening plate. Comparing the contact force distribution and contact ellipse truncation parameter over the circumstance for the starting design and the end design, it is clarified that the optimization procedure positively changes the load distribution of the bearing. The corresponding diagrams are shown in figure 5 and figure 6.
Figure 5 Improvement of load distribution through shape optimization

With regard to the forces, it can be stated that the loads are distributed more evenly between the two bearing rows. This reduces the maximum rolling element force for every load case under consideration. The improvement of the shape optimization is particularly significant for the contact ellipse truncation factor, which, for example, has fallen from a maximum value of 3.35 to 1.04 for load case P1. Also for every other load case the truncation factor and with that, the risk of contact ellipse truncation has been significantly reduced and occurring truncation effects are weakened.
4. Discussion

In this study, it is shown that a shape-optimized stiffening plate leads to significantly better contact conditions of pitch bearings. Loads are better distributed between the two bearing rows, which results in lower maximum contact forces. However, despite shape optimization, local peaks of the load distribution cannot be prevented. This suggests that the equiangular parameterization of the inner contour in particular does not provide an adequate definition of the stiffening plate’s shape. Much more the parameterization should be oriented on existing discontinuities of the hub geometry by adding
additional radial support point for the derivation of the inner contour. Nevertheless, the advantages of shape optimization become obvious regarding contact ellipse truncation. As shown, by taking truncation effects into account within the optimization procedure, occurring effects can be significantly mitigated. The fact that the effect is not completely eliminated indicates a limit given by the bearing geometry. Reducing the bearing’s osculation also reduces the risk of contact ellipse truncation and may lead to a more robust design regarding this effect.

5. Limitations

Even though the underlying FEM model already has been optimized by using surrogate models for the ball contacts, because of the still high computational cost, only a simplified load collective could be used for optimization. Indeed, rotor blade bearings are subject to a complex load spectrum. This being the case, shape-optimization of a stiffening plate actually may requires a more complex load collective that has to be considered within the optimization procedure.

Furthermore, the chosen definition of a load case does not involve the consideration of pitch adjustment. Since the stiffening plate under consideration is mounted on the rotor blade side of the bearing, a corresponding rotation of the stiffening plate around the bearing axis should actually be considered. Therefore the pitch angle has to be seen as an additional dimension of the load collective.

6. Conclusion

It is shown that on basis of a shape-optimized stiffening plate, a significantly improved load distribution of pitch bearings can be achieved. Maximum contact forces and contact angles can be reduced, which lowers the risk of damaging the raceways’ edges by contact ellipse truncation. It could be determined that the resulting stiffening plate’s shape strongly depends on the load cases under consideration. Basically, it can be assumed that by considering available time series data from design load calculations, probably results in a design that is a good compromise across all loads and optimized for the statistically most common load situations. If, however, the aim is to mitigate critical load cases, especially with regard to contact ellipse truncation, it is much more sensible to develop a special load collective for this purpose. In any case, for considering more complex load collectives, it will also be necessary to further reduce computational costs of the underlying FEM model as such by using advanced modelling techniques as substructuring. This has already been carried out in previous studies [13] and allowed the calculation of bearing load distributions for several thousand load cases, as required to consider a more complex load collective for optimization.

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