Dear reviewer,

thank you for reading the paper and thank you for your feedback. You can find responses to your comments and questions below in blue.

Comments

Overall remarks

It is a well-written paper with enough material to form a substantial contribution to the pitch bearing literature, particularly as there is a dearth of papers about three row roller bearings. As it is a methodology paper, it should be relatively easy to follow the procedure adopted which it is. The flow of the methodology has been clearly illuminated by the well-separated chapters. The reasoning for the methodology has been illuminated properly with enough validation done for each aspect of the method. There are some minor issues which need to be corrected highlighted in the next section. Since the authors validated this method with its relevant assumptions for a specific wind turbine, it may also be good to describe or postulate if this method will work in general for wind turbines with three row roller bearings as well (as much as possible within the limits of confidentiality).

Thank you for the kind words. We believe the method shown is also applicable for other wind turbines that use three-row roller bearings as pitch bearings. Most of the assumptions we took should also hold true for other wind turbines. This information was added to the conclusions.

Specific comments

1. Line 7: The modification of the bearing is mentioned but never discussed further (except in the conclusion). Please either include a line or so in the main text about this. Even if it is confidential, possibly mention this.

   The modification that was undertaken is confidential, this information was added to the introduction.

2. Line 81: The choice of 21 slices for the FEM model is not explained. Why was this value chosen? Is it standard within the extension?

   It was the maximum possible value of springs in the extension at the time we wrote the paper, this information was added to the manuscript. (Side note: Further analyses undertaken by us later show that the amount of laminae used in FE has little impact on the results of the Reusner submodel; this will be published in a separate paper in the future)

3. Line 83: "contacting roller and raceway" may be a better formulation.

   Changed as suggested.

4. Line 88: Unless confidential, possibly mention if the split happens in the inner or outer ring and approximate location. I see a split in Figure 1; this could be highlighted there, too.

   The location of the split is unfortunately confidential. Figure 1 may or may not be representative of the bearing we calculated. This information was added and the split in Figure 1 is now referred to for readers unfamiliar with the concept.

5. Line 198: The reference "Stammler et al.": Please date these references as there are multiple from the same author.

   The date was added (it is from 2024, specifically this sentence is citing the new NREL DG03)

6. Line 316-318: It is mentioned that the Reusner algorithm gives a more accurate load distribution than the FE simulation. It is mentioned later on about it being due to the Hertzian nature of contact algorithm ("For these 72 simulations, the contact pressures of all rollers were determined using a non-Hertzian contact algorithm based on Reusner (1977). Unlike the Hertzian rolling elements used in FE, the non-Hertzian algorithm allows for detection of edge stresses and is more accurate in general."). To make it clearer, the hertzian nature of the FEM model can be explained before this part of the text.

   The sentences "Pressures of these laminae can be calculated using Hertzian theory. However, Hertzian theory simplifies the roller-race contact and may underestimate the real pressure" were added prior to the cited sentences. Furthermore, the wording "This returns a more accurate load distribution" was changed to "This returns a more accurate pressure distribution" to clarify that pressure is the variable of interest here. (This is because, strictly speaking, the load-deflection relationship used in FEM based on ISO 16281 is based on Palmgren, and only the corresponding pressure calculation is based on so-called Hertzian theory).

7. Line 517: About the bending moment and resulting structural deformation playing a larger role in the loading of the radial row than the radial load itself: How exactly does structural deformation happen in this case? What is a possible reason for this? Can there be numbers put to this (if possible)?

   Structural deformation causes the rings of the bearing to deform in ways that are unfortunately difficult to predict without detailed FE simulations (refer to Fig. 4, 5, and 6). In this case, the two contacting points of the radial rollers (the raceway on the"nose" of one ring and the raceway surface on the other) deflect, too, as a result of the global load, in particular the bending moment as it is the highest load acting on the bearing. This deflection of the "nose" and the other contact point of the radial row (evidently) affects the strange, unpredictable load distribution on the radial row more so than the radial force.

   Putting numbers to this is unfortunately very difficult. In this paper, we are attempting that with these sentences:

   *The radial load Fr is very small compared to the load rating Cr of the radial row. Following the above described procedure, the life of the radial row is over 200 times as large as that of the axial row. This is incorrect: the actual life of the radial row is only about 10-15 times as high as that of the axial rows, see Fig. 18.*

We are currently not aware of a better way to put numbers to this. The above relative difference may very well differ for other bearings on other turbines, too.

8. Line 527: By a simplified approach like in Eq. 21, do you mean to say that only the axial load (ignoring the Fr mentioned in Eq.21) is to be considered here? Also, it could also be clarified then that since it is a roller bearing, the 10/3 exponent is to be used. Also, for the sake of clarity, is Ca of a single axial row to be used for life?

   No, the load Fr mentioned in Eq. 21 should also be included. We call this approach simplified because it is much simpler than the above described procedure that uses FE simulations, Reusner submodeling, etc.

   We clarified that 10/3 should be used, thank you for the note.

   For a three-row roller bearing, Ca of the entire bearing is always the Ca of a single axial row. This is because the bearing is a so-called double-direction bearing, where an *axial* load only loads one of the rows at a time but never both at once. For bending moments, both rows are loaded at once; but as the definition of Ca refers to an axial load, only one row's Ca is used as the Ca for the entire bearing. This definition follows ISO 281: Refer to ISO 281:2007, Sec. 8.1.1: *The basic dynamic axial load rating for single-row, single-direction or double-direction thrust roller bearing is given by Ca = […]*