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.

General remarks:

Well explained and clear structured paper.

Thank you for the kind words.

The graphics are too small and difficult to read.

Most of the graphics use a similar font size to the manuscript text, for images that are difficult to read, the graphics are all high-quality images and zooming in is possible. As there is a lot of information contained in the images we had to make do with the available space, unfortunately.

Some sources only given with name, no further information (year etc.) à e.g. “see Stammler et. al.”

Corrected where we found this issue (also for Keller, Jonathan and Guo, Yi)

Will the use of a full 3D model of the bearing and the time series allow to predict the circumferential location of the rolling contact fatigue failure?

While this was not the focus of this paper, other works by the authors (see https://doi.org/10.1115/1.4055916) can be used to find the area of highest damage likelihood. Of course, with fatigue being a statistical phenomenon, an exact location can never be foreseen.

Data Quality "Fair" due to the confidentiality requirement.

In detail:

Line 7: In comparison to ball bearings, the calculation of roller bearings [..] leading to significantly more degrees of freedom.

A Ball bearings also have many degrees of freedom, if modelled with balls having realistic kinematic behaviour (instead of springs and fixed contact angle) since the contact angle ball to raceway can differ from ball to ball and is strongly load dependent.

This is true for FE modeling, but this sentence was specifically meant to refer to the DOFs in the fatigue calculation. Changed the “[…] the calculation of roller bearings […]” to “[…] the rolling contact fatigue calculation of roller bearings […]” to highlight this point.

Line 8: For this paper, the bearing is modified slightly: What does modified slightly mean? Please describe more detailed, if possible.

Unfortunately the details of the modification are confidential. The information that the details are confidential was added to the manuscript.

Line 56-58: Does the roller profile used for the calculation differ between the "confidential" turbine pitch bearing and the more "generale" bearing? If so, how is the influence onto the "normalized" results?

This information is unfortunately confidential.

Line 65: But is it possible to get the ISO life calculations **quantitively** reliable?

In the scope of this paper that's a question we won't answer (yet), but generally we believe so. Look out for future publications from us on this topic.

Line 81-83: Can you explain why 21 springs are used in the FE model, whereas ISO/TS16281 requires min. 30 slices in their roller slice model?

At the time of writing the paper the extension only allowed a maximum of 21 slices, which is why this value was chosen. We included this information in the manuscript. To ensure that this does not falsify the results, we undertook analyses (including custom-built FE models with 31 slices) which showed that the amount of slices in the FE model has little influence on the Reusner submodel so long as the Reusner submodel has sufficient slices; this will be published in a separate paper in the future.

Note also that the Reusner submodel, the results of which are used for the life calculation, did include 30 slices.

Line 88: [..] This split of the ring is considered in the model and the surfaces are connected to each other by an internal frictional contact. [..] Axial clamping force? Bolt preload? Influence bolt preload on roller preload?

The axial clamping force is driven by the bolt preload of the bolts. Each bolt along the circumference is preloaded with the same load which corresponds to the target mounting bolt preload. The higher the bolt preload the higher the preload of the axial rollers on both rows of the bearing.

Line 166-167: Influence of pitch angle might only become more important if the blade model differs circumferentially from its mechanical properties. How is the blade root stiffness modelled, uniform or non-uniform, anisotropic?

The orthotropic material behavior of the blade root / blade dummy composite material is considered in the model. The material model contains direction dependent Young's moduli, Poisson's ratios and shear moduli. For the blade root a blade dummy is used as written in a manuscript, however, further details on this subject are confidential.

Line 220: again the question about blade root model and physical properties around circumference - uniform or non-uniform

As explained above, for the blade root a blade dummy was used. Further details on this subject can unfortunately not be shared.

Line 296-298: Is there a difference for small back and forward motion compared to full rotations in calculating dynamic equivalent load and bearing lifetime? If so, further explanation would help. Have time steps with negligible rotation been ignored, if so what is the minimum pitch angle to be considered?

Time steps with negligible rotation have not been ignored, as there is no clear limit for what constitutes a “negligible” rotation known to us.

Yes, generally, there is a difference for small back and forward motion compared to full rotations in calculating dynamic equivalent load and bearing lifetime. When a bearing is fully rotating, the rotating ring is continuously rotating through the highest loaded zone. All spots on the circumference of the rotating ring thus go through the load zone. When it is oscillating back and forth, only one area on the circumference will be in the highest loaded zone. This is discussed in detail in Menck and Stammler (2024) and we added the following sentence to the manuscript to point this out:

These small oscillations cause the oscillating ring to be almost stationary w.r.t. the load, differing from a rotating bearing, where the rotating ring rotates relative to the load.

Line 310: why 21 laminae in FE compared to min. 30 as per ISO 16281

As explained above, this is in part due to limitations in the extension at the time of writing this paper, however, we ensured that it does not affect the results, details on which will be published in the future. The Reusner submodel does fulfill the minimum requirement of 30 slices.

Figure 14: Graphics of convergence analysis in not convincing compared to line 349. The graphics do not show why 30 lamina is a suitable choice.

They show that for 30 laminae, L10r is within a +-3% range of the reference result using 150 slices. This was the intention of the figure.

Line 405: What is the expected difference in rating life between analysis done on time steps compared to load bins?

This is impossible to answer generally as it highly depends on the binning process. In WES - Review of rolling contact fatigue life calculation for oscillating bearings and application-dependent recommendations for use, Figure 8, we are doing a comparison, but we would assume that for other choices of bins the result can be quite different.

Line 516: What impact will the radial row, at 10-15 times the life of the axial raceways, have on the combined rating life?

A very small impact. As you can see in Fig. 20, Column “Radial”, even though the radial row approximations are sometimes wildly off the reference (“FE to Reusner”), the combined life in the “Whole bearing” column deviates very little.

Line 519-520: Will radial load be added to eq. ax. load Pa and compared with axial load capacity Ca? Further explanation required for formula.

Yes, that’s the way Eq. 21 is generally employed. Added the sentence “The radial load is thus included in the dynamic axial load” after Eq. 21 to clarify this point.

Line 529: Further explanation required, also in relation to formula above. Normally ball bearings have higher k factor on moments compared to axial rows of 3 row roller bearings if calculating axial and radial raceways separately on 3 row roller bearings.

We are unsure which k factor you are referring to here, as this calculation is the first we are aware of for a three-row roller bearing in the published literature.

Line 566: What would be a high edge stress?

A significant increase of the stresses at the roller edges as compared to the roller center. We are unaware of any strict definitions (e.g., 50% higher than the roller center… or similar), but the phenomenon of high edge stresses is generally recognized in the literature, see ISO 16281, Harris and Kotzalas (2007), etc.

Line 581: What is the expected failure mode of a three-row roller bearing

We believe this question is beyond the scope of this paper, but rolling contact fatigue is one possible failure mechanism among a number of possible failure mechanisms.