Manuscript Title: Investigating Grease Behaviour in Tilted Double-Row Tapered Roller Bearing Installed in Wind Turbine by Developing a Full Scale Multi-Phase CFD Model

Journal: Wind Energy Science

Author Responses to Comments from Reviewer 1

Comment:

I think the reasoning behind the scaling of the rolling bodies was clear in the previous version of the manuscript. I would, however, still argue, that this allows grease flow in the model where there is no grease flow no reality. Without eing privy to the intricacies of the software, I could it imagine possibilities to include 2D-elements which act as a (semipermeable) barrier between scaled rolling bodiy and raceway at their shortest distance to limit the grease flow between the bodies and make the results more realistic and still allow reasonably shaped elements around it?

Author Response:

We thank the reviewer for this constructive suggestion. As explained in the manuscript, the rollers were uniformly scaled down by 2% to create a small clearance that permits robust hexahedral meshing and avoids highly skewed cells (or triangular prisms) in the near-contact regions. We fully acknowledge the reviewer's concern that the artificial clearance could permit grease flow in regions that are effectively closed in the physical assembly.

The scaling is sufficiently small that it does not alter the dominant, global grease behaviour under investigation (redistribution due to rotation, gravity and tilt, and seal pressurization), which are governed by the large-scale geometry, tilt angle and fill ratio. Moreover, explicit barrier elements introduce non-physical internal boundaries that may complicate the multiphase VOF interface and boundary condition treatment.

Furthermore, the authors have demonstrated that, at least in oil lubrication, the presence of a 2% gap does not affect the main flow patterns observed in tapered bearings (Maccioni et al., 2023a; Maccioni et al., 2025).

Comment:

Please add the cone agnle to this table.

Author Response:

Thanks for pointing this out. The cone angle has been added to Table 1.

Comment:

This explanation still misses the point of the cone angle (that is why they are called 'tapered rollers' and not simply 'rollers') and is, in its present form, not correct.

Author Response:

The Paragraph has been revised, and explanation of the cone angle has been added.

Comment:

Somewhere here it needs to be mentioned that the rolling bodies were scaled.

Author Response:

Thanks for pointing this out. This information has been added in the revised version.

Comment:

I think it is important to mention this does not cover any possible oil/thickener separation

Author Response:

Yes, this assumption is now mentioned in the conclusion that grease was treated as homogeneous.

Comment:

How do you know it did? Did you compare it to measurements?

Author Response:

Thanks for pointing this out. The zones which had no grease or less grease were treated as critical zones having grease starvation.

Comment:

Is there a risk? How high is the risk? What is the grease pressure at the sealing? I think those questions can be answered in the summary

Author Response:

Thanks for seeking this information. These details have now been included in conclusion mentioning pressure on seals and maximum threshold pressure the seal can withstand.

Comment:

Can you get a bit more precise here - what do you mean by lubrication system? THe inlet channels, the entire inner volume of the bearing?

Author Response:

The sentence has been revised stating now more specific details of lubrication system.

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Responses to Comments from Reviewer 3

Referee Comment:

19- "variable temperatures" --> more specific, very low temperatures

Author Response:

Thank you for pointing this out. The sentence has been revised.

Referee Comment:

35- "rolling elements" --> also cage and rotating races introduce losses

Author Response:

Thank you for this information. Based on your comments, the sentence has been revised.

Referee Comment:

96 - "The use of symmetry is not always valid": Yes. Please use generic explanations when it will be valid and when they won't. From the latter, you can conclude why your example belongs to the no-symmetry. Please also briefly discuss the differences in roller positioning at the initial state. It is sufficient to take the extreme cases, i.e. one roller at the minimum position and the first two rollers at equal height.

Author Response:

Thank you for pointing out. The paragraph has been revised, and explanation has been added that use of symmetry is valid simplification for in CFD analysis for bearings which are aligned to gravitational acceleration, no misalignment, and uniform load distribution. However, when the bearings are tilted and huge in size, then each sector experience different load and lubrication characteristics. Therefore, studying the entire bearing geometry is important for understanding the entire phenomenon.

Referee Comment:

127-128- Highlight the implications of this scaling. Expand what it means for the simulation to actually work with an unscaled geometry and not only for the mesh.

Author Response:

The text has been revised accordingly. The implications of scaling on the geometry and results have been explained in the revised manuscript and references are inserted.

Referee Comment:

135- Why was this pattern chosen, and why is it "specific"?

Author Response:

Thank you for your comment. The block pattern was chosen with two primary considerations in mind:

(i) to accurately represent the curved geometry of the raceways and roller contacts
(ii) to ensure high mesh quality by avoiding highly skewed or ill-shaped cells.

While in principle other partitioning schemes could be applied, the selected pattern provided the best compromise between geometric fidelity, acceptable aspect ratios, low skewness, and computational efficiency for such a large-scale model.

Referee Comment:

137- How is "high accuracy" measured? What would be the implications for "low-quality meshes"?

Author Response:

The accuracy of the flow predictions was not directly assessed in this study; instead, we relied on the experience gained from previous works published by the authors. In general, when it comes to mesh-based CFD studies, higher accuracy can typically be expected from high-quality hexahedral meshes—such as the ones used in our simulations, as highlighted in Tables 2 and 3—compared to meshes composed of tetrahedral or triangular prism elements. It is also worth noting that alternative simulation approaches, such as SPH, have shown good accuracy in predicting flow behaviour, at least in oil-lubricated configurations. These considerations are acknowledged in the manuscript, and we provide below a number of references to support these claims.

Concli, F., Schaefer, C. T., and Bohnert, C.: Innovative meshing strategies for bearing lubrication simulations, Lubricants, 8, 46, https://doi.org/10.3390/lubricants8040046, 2020.

Maccioni, L., Chernoray, V. G., Bohnert, C., and Concli, F.: Particle Image Velocimetry measurements inside a tapered roller bearing with an outer ring made of sapphire: Design and operation of an innovative test rig, Tribology International, 165, 107 313, https://doi.org/10.1016/j.triboint.2021.107313, 2022.

Maccioni, L., Chernoray, V. G., and Concli, F.: Fluxes in a full-flooded lubricated tapered roller bearing: particle image velocimetry measurements and computational fluid dynamics simulations, Tribology International, 188, 108 824, https://doi.org/10.1016/j.triboint.2023.108824, 2023a.

Maccioni, L., Ruth, L., Koch, O., and Concli, F.: Load-Independent Power Losses of Fully Flooded Lubricated Tapered Roller Bearings: Numerical and Experimental Investigation of the Effect of Operating Temperature and Housing Wall Distances, Tribology Transactions, 66, 1078–1094, https://doi.org/10.1080/10402004.2023.2254957, 2023b.

Maccioni, L., Chernoray, V. G., and Concli, F.: Investigating lubricant behavior in a partially flooded tapered roller bearing: Validation of a multiphase CFD solver for aerated oil sump via particle image velocimetry studies and high-speed camera acquisitions, Tribology International, 201, 110 274, https://doi.org/h10.1016/j.triboint.2024.110274, 2025.

Referee Comment:

141 - 176- This section can be shortened by mentioning that you had to write a Matlab script to ensure that all sections are aligned properly and correct boundary conditions are imposed due to missing OF features. Keep emphasizing the particularities of OF to ensure that readers can replicate your workflow if desired. This will make readability easier.

Author Response:

Thank you for your comment. This paragraph was revised on the recommendations of another reviewer. I have tried to include all the details of the MATLAB script and how it is then implemented in OpenFOAM.

Referee Comment:

2.2.3:
- Please highlight more clearly why there had to be certain patches corresponding only to the bearing and three additional ones for the IO region.

Author Response:

Thanks for seeking this clarification. The three additional patches were introduced for numerical reasons. In OpenFOAM, pressure is solved in a relative sense, and the system of equations requires at least one boundary where the pressure is prescribed to get the solution. The bearing surfaces (rollers, raceways, cage, seals) cannot be assigned atmospheric pressure because they act as no-slip walls, where the solver must resolve the pressure field based on the surrounding flow. Therefore, we defined three additional patches with fixed pressure to serve as reference boundaries. This ensured that the solver could establish the global pressure field consistently while allowing the pressure inside the bearing to evolve naturally according to the governing equations.

Moreover, the patches where pressure boundary conditions were applied are fixed and correspond to the inlet and outlet of the bearing lubrication system. Since the remaining components of the mesh rotate according to a rigid mesh motion strategy, an Arbitrary Mesh Interface (AMI) was introduced between the stationary part of the bearing and the rotating parts.

The manuscript has been updated providing explanation to clarify this concept.

Referee Comment:

196-197- "values exceeding 20 may lead to interpolation errors & numerical instability" - source or explanation. Is it a stepwise function or does the error gradually increase?

Author Response:

Thanks for seeking this information. The threshold of 20 for aspect ratio is not a strict stepwise limit but a commonly cited guideline in CFD best practice literature (e.g., OpenFOAM user guide, ANSYS meshing guidelines). In practice, numerical errors such as interpolation inaccuracies and instability tend to increase gradually with aspect ratio. However, once values exceed 20, the risk of divergence or physically unrealistic results becomes significantly higher, particularly in regions with strong gradients. In our case, the maximum aspect ratio (13.968) remained well below this critical range, which supports the robustness of the mesh.

Referee Comment:

198- Why does the skewness of >=4 create numerical errors? Is it a stepwise function or does the error gradually increase?

Author Response:

Thank you for pointing out this. It is also not a strict limit for skewness but the values above 4 more likely lead to numerical instabilities and errors and are not recommended by CFD best practice literature (e.g., OpenFOAM user guide, ANSYS meshing guidelines).

Referee Comment:

Table2:
- You might want to comment on why the full model had a higher max aspect ratio than the sector model and where these "bad quality cells" are located.
- Text refers to max non-orthogonality and max face volume ratio, but the table gives average values for both. This must be aligned to the maximum value.

Author Response:

The Table 2 has been updated with sector model details. In the manuscript, now the tables and text are consistent mentioning the maximum non-orthogonality and max skewness. For face volume ratio, the value we got is average.

Referee Comment:

207- Explain why the isothermal assumption is valid. When would it not be appropriate?

Author Response:

Thanks for seeking this information. The isothermal assumption is justified in the present study because the main bearing operates at very low rotational speeds (17.5 rpm), where frictional heating and temperature gradients within the lubricant are negligible. Consequently, thermal effects on grease rheology and flow behaviour can be reasonably neglected. However, at higher speeds, heat generation within the lubricant and bearing components becomes significant, and a fully coupled thermo-fluid analysis would be required.

Referee Comment:

208- Too generic: Every fluid is non-Newtonian if the observed shear-stress range is wide enough. What you describe is specific. It is advised to show how much of the "fluid definition" was eventually used within the simulation.

Author Response:

Thanks for the comment. In the revised manuscript, we have clarified that the *complete* set of experimentally determined rheological parameters (tau, K, n) was directly implemented into the CFD solver. Thus, the full constitutive definition of grease was used, ensuring that its non-Newtonian behaviour was fully captured in the simulations.

Referee Comment:

240- "Homogeneous" - this implies that inhomogeneities like aged grease would not be captured. That should be discussed.

Author Response:

Thanks for this observation. In the present study, grease was modelled as a homogeneous non-Newtonian fluid to establish an understanding of its global flow behaviour in the bearing. We agree that this assumption does not account for inhomogeneities such as grease aging, phase separation, or local filler concentration. These effects are beyond the scope of the present work but are important directions for future research.

Referee Comment:

Table 5- Boundary Conditions: PRESSURE - explain how the pressure boundary conditions must be understood. I assume you must prescribe the in- and outflow of a distinct volume fraction - probably 100% oil here. What happens if the inner fluid state at the boundary has a different phase fraction than what you prescribed, e.g. an entrained air bubble touches the boundary condition?

Author Response:

Thank you for raising this point. In this case, during the pressure assignment, each fluid is allowed to exit the domain (inlet/outlet), but only air is allowed to enter. No flow rate was prescribed — only the pressure boundary condition was applied.

However, due to the positioning of the pressure patch, the grease present inside the bearing would not reach the patch (and this has been confirmed by the simulations). Therfore, the multiphase nature of the simulation is not affected by the presence of this patch that, in turn, has the task to provide a reference pressure and a possible out-flow or in-flow of the air.

This condition is also consistent with the experimental setup, where those channels are used only for the periodic refilling of grease and are “opened” during the operation.

Referee Comment:

273- Typo (soound)

Author Response:

It has been rectified.

Referee Comment:

276- Typically, you'd use Gauss's theorem for the mathematical derivation of the method and not for the calculation per se. Are you referring to a Gauss gradient calculation method?

Author Response:

Thanks for your comment. The sentence has been now rectified. Gauss gradient scheme was used for computing spatial gradients.

Referee Comment:

285- The author may indicate what one full rotation of the cage translates into time (~7.1s?).

Author Response:

Thank you for the suggestion. It has been now explicitly mentioned.

Referee Comment:

285-286: The author may indicate the runtime of one run. This is important for anyone redoing a similar study.

Author Response:

It has been mentioned in the revised manuscript.

Referee Comment:

294-297: Simulating this with a Herschel Bulkley should lead to some grease being stuck in some areas of the bearing, which would be interesting to know from your very last analysis. In reality, grease forms a thin oil film over time and should flow back (almost) completely into the sump. Can you please comment on the difference and how it can be treated?

Author Response:

Thank you for the observation. As the grease was modelled as a homogeneous viscoplastic fluid using the Herschel–Bulkley formulation, regions where the local stress remained below the yield stress exhibited solid-like behaviour, leading to grease retention in those zones. In reality, grease gradually releases base oil, which forms a thin lubricating film that drains back into the sump. While such effects can in principle be captured by multiphase models incorporating grease–oil separation, their inclusion would require significantly higher computational effort and was therefore beyond the scope of the present work.

Referee Comment:

304- Too early to tell after just one rotation. The author should add graphs showing the evolution of the wetted surface with time so that the reader can understand if one rotation is anywhere close to a quasi-stationary state. The grease occupancy (Fig 11) is good, but also shows that longer simulation times are required. The reader wishes to see longer simulation times.

Author Response:

Thank you for the comment. We agree with the reviewer that longer simulation times would further confirm the attainment of a quasi-stationary state. However, as shown in Fig. 11, the grease occupancy pattern already approaches quasi-stationarity within one rotation (7.1 s), with no significant redistribution trends observable beyond this point. Extending the simulation to multiple rotations would indeed be desirable, but as noted in the manuscript, each run required approximately 35 days of wall-clock time.

Referee Comment:

317- The author just added the difference in initial grease filling due to the tilted alignment to make it easier for the reader to understand if 8.5% difference is large compared to the initial state.

Author Response:

Thank you for your comment. Initially, both the rows have the same grease occupancy due to tow reasons.

- In our setup, the initial grease distribution was prescribed using the setFields utility in OpenFOAM, which ensured identical occupancy for both rows at t = 0 across all filling ratios.
- With the Herschel–Bulkley model and a 5° tilt, the static gravitational stresses are smaller than the yield stress, so grease does not “flow down” immediately. The difference between rows emerges only dynamically during rotation, once local shear stresses exceed the yield stress and redistribution begins.

These reasons have been explicitly mentioned in the revised manuscript.

Referee Comment:

321- Rows should have different grease occupancy due to the tilting of the bearing. If this was not considered, the author must comment on what the difference in filling volume would be, considering the tilting.

Author Response:

This point has been explained above.

Referee Comment:

325- "after" + a

Author Response:

It has been corrected.

Referee Comment:

325-329- The reader would like to follow the explanation of the author, about the interesting behavior, but lacks evidence from simulation images that substantiate the description. This must be added.

Author Response:

Thanks for pointing this out. The redistribution phenomenon of grease cannot be easily visualized from the full-bearing simulation images due to their scale and complexity. Therefore, we believe that the graphs presented in figure 11 are the most effective way to demonstrate and quantify this behaviour.

Referee Comment:

Figure 11: The author should please make the spacing in the y-axis of each graph equal to make it easier to depict the differences between each case.

Author Response:

Thank you. Figure 11 has been updated accordingly.

Referee Comment:

330- Which value of alpha was used to detect a wetted cell? Anything >0? How do you treat a surface that has been wetted but is not wetted anymore?

Author Response:

We used the alpha value of 0.5 which means that cells which had grease occupancy of 50 % or above were treated as wet.

Referee Comment:

Figures 12- Since you have the data for 7s, please add them to the chart. This will help substantiate your claims further.

Author Response:

Thanks for this valuable suggestion. Basically, these graphs aim to investigate the time required for the grease to reach the topmost part of the bearing for different filling ratios and for different bearing components. Therefore, it is the time in which the grease reaches the top for each case. In some cases, the grease did not reach the top.

It is explicitly mentioned in the manuscript.

Referee Comment:

335-348- Doesn't the widened gap between roller and raceway further reduce the transport process from grease to raceways, particularly the outer one? While the inner one might move the grease on its own.

Author Response:

Thank you for this observation. As explained earlier, widening the gap between rollers and raceways are in acceptable range. It does not have any effect on the global grease behaviour and its distribution.

Referee Comment:

349-354- Essentially, this means that the odd trends in the chart are caused by numerical (maybe physical?) rupture of the grease. Can you please highlight the onset of the phenomenon in the graph so it'll become clear that these results have to be considered with care?

Author Response:

We acknowledge that the current definition of surface wettability may appear somewhat dependent on the numerical threshold (alpha = 0.5). In our simulations, we intentionally used an alpha value of 0.5 to define the wetted regions based on the cell volume fraction. This means that a surface is considered wetted when at least half of the cell adjacent to it is filled with grease.

Naturally, using a lower threshold or a finer mesh would result in larger wetted areas, as the effective film thickness on the surfaces would be better resolved. Therefore, the definition of wettability can be interpreted in terms of the average cell thickness at the interface: if a surface appears wetted, it corresponds to a grease layer of approximately half the local cell thickness.

We recognize that this is a limitation of the current numerical approach; however, it still provides a meaningful indication of the grease distribution. Future studies should aim to determine the minimum grease film thickness that should be detected and assess whether this level of resolution is compatible with available computational resources and industrial time constraints.

Referee Comment:

355-359- Can you please highlight if there are noticeable differences in the fluid distribution between the different filling levels? Do we only see grease velocities or also mixture velocities? For these and the consecutive images, add an indicator of the rotational direction of the bearing.

Author Response:

Thanks for the comment. Yes, there are differences in fluid distribution for filling ratios. In the figures, the velocity of mixture is shown. The difference can be illustrated in the zoomed portion of the figures.

The indicator of rotation has been added in revised manuscript.

Referee Comment:

Figure 15 & 16- It will be easier to depict the pumping effects if you change the colormap and make the range smaller. Please be precise in wording: the roller of the bearing will always displace fluid. Pumping means that the displacement has a favorable direction.

Author Response:

Thanks for thee comment. The colormap in Figures 15 and 16 was already chosen with a narrow range to highlight pumping effects. Reducing the range further would, however, truncate the maximum grease velocities and misrepresent the full velocity field.

We have updated the text in the revised manuscript accordingly.

Referee Comment:

375-379- By how much does sealing pressure increase with filling height? Are there overlinear effects? With the simulated pressure, are the sealings at risk of failing?

Author Response:

It has been discussed accordingly in revised manuscript. The maximum pressure which seal of this bearing can withstand is 0.5 bar, whereas after simulation, we found that the maximum pressure on seals increases with the accumulation of grease or higher filling ratios. The region of maximum pressure varies with the filling ratio. For these filling ratios, the seals are in the safe limit.

It has been concluded that for these filling ratios, there exists no risk of leakage.

Referee Comment:

Figure 18: Add MFS and HS note (left)/(right) for better readability

Author Response:

It has been implemented in revised manuscript accordingly.

Referee Comment:

380-386- This is an interesting study, but it lacks interpretation on why the grease is stuck in some upper region. Also, does it simply take more time to settle? Is this a reasonable outcome, or would we assume more mass wetting the bearing surface with time? Without proof, just gut feeling, 40s is fast for such a big bearing with a grease at 40°C; investigating the graph, the majority of the settling happens within the first 15s.

Author Response:

Unfortunately, it is not possible to make a precise statement on this point. Since an interface-capturing solver is used, the 2% volume is distributed over several cells and does not occupy at least 50% of any given cell. This means that the grease could be in contact with the surfaces, but the mesh resolution is not fine enough to confirm its exact position with certainty. The kinematic viscosity of grease drops drastically at 40 °C having value 450 mm^2/sec.

Referee Comment:

394 - 396 - I wonder if these findings could have been made without any simulation?

Author Response:

The point raised is indeed relevant to the overall usefulness of the study. We agree that certain trends and behaviours could have been qualitatively anticipated based on experience, even without simulations. However, the use of numerical simulations allowed us to quantify them taking into consideration the fluid dynamic and the non-Newtonian behaviour of the grease.

Referee Comment:

396 - Did the author discuss when those sealings are at risk of leakage?

Author Response:

It has been discussed accordingly in revised manuscript. The maximum pressure which seal of this bearing can withstand is 0.5 bar, whereas after simulation, we found that the maximum pressure on seals developed for these filling ratios is in the safe limit. The region of maximum pressure varies with the filling ratio.

Referee Comment:

396 - significant impact of gravitational & tilting effects - Please include a Froude number discussion for the next revision.

Author Response:

The Froude number is an important dimensionless parameter used to study the relative influence of inertial and gravitational forces. In the present case, the velocities involved are relatively low; therefore, the gravitational effects dominate over the inertial effects.

Referee Comment:

404-405- Please include the experimental validation to highlight the validity of your method for this field.

Author Response:

The test bench is currently under design, and its construction is planned to be completed by the end of 2027. For this reason, it is not possible to include experimental results in the present work. However, we are confident that these results will be available and discussed in a future study.