Reviewer 1: D. Strasser

Comment #1: "p. 1, line 16: with a load dependency by the power of 3.3 for line contact rollers one would expect a live increase of about 2.1 at load reduction of 0.8"

Authors' response to comment #1: The authors agree – indeed, "back of the envelope" calculations like this are very helpful. It is similar to previous work quoted in the paper of an increase of 3x in other industrial applications.

No changes were made to the manuscript.

Comment #2: "p. 2, line 5ff: one should also take into consideration the researches applied on planetary gear set load sharing back in 1990ies and 2000ff years at Ruhr University Bochum (e.g. Vriesen, Lamparski, Winkelmann, etc.)?"

Authors' response to comment #2: The following references have been added to the paper. The authors have not yet been able to obtain English versions of 2 of the references by the deadline, so we are placing the references at the most general point in the paper.

Predki, W. and Vriesen, J. W., *Calculating gear tooth corrections for planetary gears. Theoretical basis and practical benefit*, Europe invites the world, International Conference on Gears, VDI reports; 1904.1; 311-326, Düsseldorf; 2005

Lamparski, C., *Einfache Berechnungsgleichungen für Lastüberhöhungen in Leichtbau-Planetengetrieben*, Research reports of the Ruhr University Bochum. Institute of Design Engineering, 95.3; 1-246, 1995

Winkelmann, L., *Lastverteilung an Planetenradgetrieben; Schriftenreihe des Instituts für Konstruktionstechnik*, Heft Nr. 87.3, Ruhr-Universität Bochum, Diss., 1987

Comment #3: "p. 7, line 12ff: area of Downwind and Upwind is nearly the same, area can be seen as total bearing load. Further on, measured values are 20% greater than measured (upwind) and 10% smaller (downwind)"

Authors' response to comment #3: Based on the experimental data (the solid circles) in Fig. 5, the area of downwind bearing is less than the upwind bearing – at +300 kNm by about half. The total load for each bearing is calculated from this area and then displayed in Fig. 7 over the entire carrier rotation. We are slightly confused by the question, as it states "measured values are 20% greater than measured (upwind) and 10% smaller (downwind)" – one of these must be predicted. At any rate, we believe the question might pertain to the fact that in Fig. 5 the area (total bearing load) calculated from the experimental data might appear to be much larger than the area (total bearing load) calculated from the Transmission 3D model. This is a reasonable assumption. However, as stated in the manuscript "For the instrumented CRBs, a direct-calibration factor is used to determine the total bearing load (van Dam, 2011, Harris and Kotzalas, 2006) from only the TDC measurement." That is, due to the calibration test process used by van Dam, just the TDC measurement is used to directly calculate the total load – in this case there is no calculation of the area to get the total bearing load. It is an artifact of this process that the total bearing loads (experimental data and Transmission3D) shown in Fig. 7 are closer than Fig. 5 might otherwise indicate. In contrast, there is a calculation of the area (total bearing load) for the instrumented TRBs.

No changes were made to the manuscript.

Comment #4: "p. 7, line 21: Romax model obvously matches measurements better than Transmission3D model?"

Authors' response to comment #4: For the upwind bearing, the Romax results are closer to measurement. However, for downwind row bearing experimental results lie between Romax and Transmission3D modeling results. The Romax models assume rigid bearing races while Transmission3D consider flexibility of raceways. Additional sentences have been added to highlight this difference as follows:

The RomaxWind model assumes rigid bearing races while the Transmission3D includes the flexibility of the races. This results in a more circular load zone prediction for RomaxWind compared to an elliptical load zone for Transmission 3D.

Comment #5: "p. 8, line 13: probably it is meant: planet carrier bearing clearance leads to misalignment due to gravity force on planet carrier?"

Authors' response to comment #5: The authors agree that carrier bearing clearance has a greater impact on misalignment and loads than the planet bearings. This sentence has been changed slightly to read:

The CRB loads fluctuate over the rotation and are also out of phase because of the combined effect of planet and carrier bearing clearances and gravity and the resulting gear misalignment (LaCava et al, 2013).

Comment #6: "p. 8, line 19: it is not clear how the interference fit influences the bearing loads. Physical effect should be described"

Authors' response to comment #6: Interference-fitted pins also stiffen the connection between planet pins and the carrier, reducing planet pin and thence planet gear misalignment. This is not a large effect; however, so this sentence has also been changed to:

The planet TRB loads are much more consistent over the carrier rotation due to the preload in the bearings and, to some extent, the interference-fitted planet pins that also reduce misalignment.

Comment #7: "p. 9, line 5: the individual bearing load is practical relevant, the relevance of the total measured bearing load is not clear."

Authors' response to comment #7: We agree. The individual bearing load, especially for the CRBs, is most relevant. However, the calculation and discussion of the total bearing load does still have relevancy, we think, in this paper. The main point being that the total bearing load is within the range assumed and desired in planetary gear design standards (Kgamma < 1.1) across almost all of the moments, even though the individual CRB loads are not. It is the contrast and disparity between the two that we think is interesting, especially in pure torque conditions. This discussion and comparison is included in the next section of the paper related to the planet bearing load sharing factor (Kgamma).

No changes were made to the manuscript.

Comment #8: "p. 10, line 18f: for the sake of clarity the formulas with which the curves have been computed should be shown. It would for instance be logic to put the individual bearing loads in relation only."

Authors' response to comment #8: In this section of the paper, there was no real "formula" used to translate results like Fig. 7 over a carrier revolution into the summation in Fig. 10. The paper states "*In this study, the maximum load throughout the main shaft rotation shown in Fig. 7–9, which accounts for both constant load differences and the fluctuating load from gravity and rotor moments, is examined for comparison to this assumption*." That is, the highest point seen in Fig. 7 is one of the points shown in Fig. 10 for pure torque. That process is repeated across all of the pitch moment cases. If this was not the point that the reviewer was commenting upon, we would ask for further clarification.

No changes were made to the manuscript.

Comment #9: "p. 10, line 20f: practical experience shows significant lower values, values far above 1.1 are implausible. The physical effect should be described. At a three-planetary system the self-aligning functionality leads to the assumption that Kgamma should be nearby one."

Authors' response to comment #9: The authors agree that this is the traditional assumption and is true in many, if not most, gearbox applications. It is even a good assumption for this gearbox when examining the total bearing load. However, we find that it may not be a good assumption at all in the wind turbine gearbox application where the gearbox is mounted horizontally - especially when examining the maximum of the individual bearing load over the carrier rotation. What are thought of as implausible values, on the order of 1.4, have been demonstrated conclusively in this work by both full-scale tests and the highest fidelity finite-element models available. These values are primarily a result of the effect of gravity on the planetary system with bearing clearance; and to a certain extent also because of the effect of moments. This is exactly the main point of this paper – to demonstrate that this assumption is not true, even in what is thought of as a benign (or even best case) pure torque condition. Regarding a description of the physical effect, it is true that we have offered relatively short explanations and references to prior work at various points in the paper such as "*The loads are not equally shared in practice. The constant difference is a result of deformations, displacements, and manufacturing deviations causing consistently higher loads on one planet than the others (Cooley and Parker, 2014). The fluctuating component is a result of the rotor moments and gravity, exacerbated by planet and carrier bearing clearances and resulting in misalignment in the gearbox with the CRBs, causing a once-per-revolution load variation over the carrier rotation (Guo et al., 2015)*".

Having said that, these results were surprising enough to us that we have undertaken the formulation of a purely analytical description of this phenomenon as an extension to the load-sharing work of Singh (2009). For a given gearbox design (including bearing clearance), operating torque, and applied moment the load-sharing factor will be larger than 1 even for a self-aligning 3-planet design due to the effect of gravity. The effect becomes more pronounced with 4-planet and higher systems. We are preparing it in a separate manuscript, as it is generally applicable to any planetary gearbox mounted horizontally. We felt that it was beyond the scope of the present paper, as the formulation is quite lengthy. But a short explanation follows here.

The forces acting on the planetary system are illustrated in the below figure. A force-balance in the x, y, and u directions yields

$\sum_{i=1}^{n} F_i^R \cos(\beta_i + \theta) + F_i^T \sin(\beta_i + \theta) + F_{ex}^X + F_{cr}^X = 0$ (1)

$\sum_{i=1}^{n} -F_i^R \sin(\beta_i + \theta) + F_i^T \cos(\beta_i + \theta) - mg + F_{ex}^Y + F_{cr}^Y = 0$ (2)

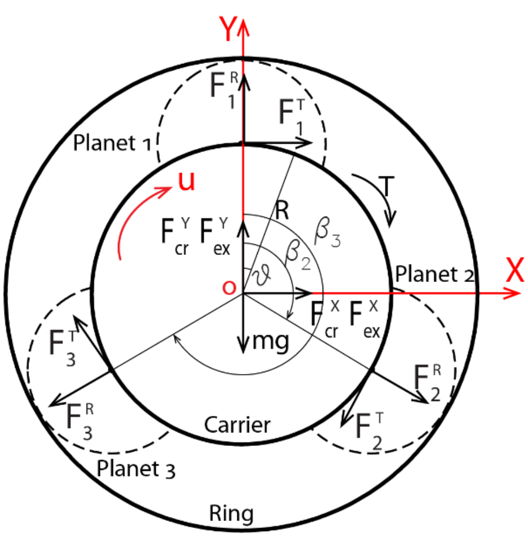
$\sum_{i=1}^{n} F_i^T - \frac{T}{R} = 0$ (3)

where n is the total number of planets and θ denotes the rotation angle of the carrier. $\beta_i . i = 1, \ldots n$ is the position angle between planet *i* and planet *1*. $F_i^{R,T}$ are the forces acting on planet *i* in its radial and tangential directions. *T* is the transmitted torque and *R* is the center distance between the sun and planets. $F_{ex}^{X,Y}$ are the reaction forces as a result of rotor moments. $F_{cr}^{X,Y}$ are the reaction forces at the carrier bearing. The planet loads can be estimated analytically based on above equations as follows:

$F_i^R = -\frac{2}{3}(F_{ex}^X + F_{cr}^X) \sin(\beta_i + \theta)$ (4)

$F_i^T = \frac{T}{3R} + \frac{2}{3}(mg - F_{ex}^Y - F_{cr}^Y) \sin(\beta_i + \theta)$ (5)

The planet bearing total loads for three-planet system depends on torque, gravity, and rotor moments. Even under pure torque conditions where there are no external moments ($F_{ex}^{X,Y}$ = 0), the bearing loads are still affected gravity, resulting in fluctuating loads during carrier rotation.



Force diagram of a planetary gear system

Comment #10: "p. 10, line 22ff: extreme values of greater than 1.2 are implausible, also values lower than 1.0."

Authors' response to comment #10: We offer a similar response to this comment as to the comment prior. The key point is that the value that we discuss in this paper is the maximum value of the fluctuating bearing load component over the entire carrier rotation, not the average. This fluctuating effect is due to gravity and any applied moment. Although these values are not constant over the full rotation, they still have a significant impact on the predicted bearing L10 life.

No changes were made to the manuscript.

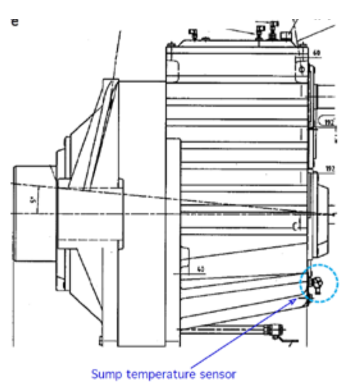
Comment #11: "figure 11: unstetic behaviour of CRB at +/-100 kNm Needs Explanation, dito for upwind load at Zero pitch Moment. Model achitecture should be explained via sketch and text.."

Authors' response to comment #11: Regarding the gearbox architecture, we were hoping that Fig. 2 could suffice to show those not directly familiar with a typical 3-stage wind turbine gearbox. The carrier bearings were modeled as springs with a constant stiffness, but with a piece-wise nonlinearity due to their individual clearances. The upwind (rotor-side) carrier CRB has larger clearance than the downwind CRB (as listed in Table 1 on Page 4). Because of this, the downwind CRB comes into contact first and reacts the applied moment and gravity loads. For this set of bearings, the upwind CRB does not carry any loads. This is certainly not desirable; the load distribution between the upwind and downwind carrier TRBs is much better. The discussion of this figure has been changed and slightly expanded to:

Beyond ±100 kNm pitch moment, the downwind carrier CRB load increases while the planet CRB load does not. The downwind carrier CRB supports essentially all the additional load. Within ±100 kNm pitch moment, the planet CRBs carry any load while the carrier CRBs are both unloaded. For this gearbox, the upwind carrier CRB does not carry any load regardless of the pitch moment. This behaviour is a direct result of the relative clearances of all the carrier and planet CRBs.

Comment #12: "p. 13, line 7ff: fits to practical experience. Should be shown where t_sump is measured.."

Authors' response to comment #12: Gearbox sump temperature is measured at the bottom rear of the gearbox, near the oil return line from the sump to the oil cooler. We offer this figure below to better show the location (also provided as a reference in the manuscript), but are not sure if such a figure is worthwhile to add to the paper or what additional explanation of the location of the sump temperature measurement is worthwhile.



Comment #13: "p. 16, figure 17: increased bearing load for upwind bearings at pure torque condition not plausible, physical phenomenon should be explained."

Authors' response to comment #13: We offer the same response as comments #9 and #10. The main and most valuable conclusion from this work, we believe, is that the planet CRB loads are not equal in the wind turbine application **even for a self-aligning 3-planet system in pure torque conditions**. The disturbed load sharing is a direct result of bearing clearance, gravity, and gear misalignment.

No changes were made to the manuscript.