Comparison of Planetary Bearing Load-Sharing Characteristics in Wind Turbine Gearboxes

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Abstract. In this paper, the planetary load-sharing behaviour and fatigue life of different wind turbine gearboxes when subjected to rotor moments are examined. Two planetary bearing designs are compared—one design using cylindrical roller bearings with clearance and the other design using preloaded tapered roller bearings to support both the carrier and planet gears. Each design was developed and integrated into a 750-kilowatt gearbox. In field-representative dynamometer tests, the loads on each planet bearing row were measured and compared to finite element models. Bearing loads were not equally shared between the set of cylindrical roller bearings supporting the planets even in pure torque conditions, with one bearing supporting up to 46% more load than expected. A significant improvement in planetary bearing load sharing was demonstrated in the gearbox with preloaded tapered roller bearings with maximum loads 20% lower than the gearbox with cylindrical roller bearings. Bearing life was calculated with a representative duty cycle measured from field tests. The predicted fatigue life of the eight-combined planet and carrier bearings for the gearbox with preloaded tapered roller bearings is 3.5 times greater than for the gearbox with cylindrical roller bearings. The influence of other factors such as carrier and planet bearing clearance, gravity, and tangential pin position error is also investigated. The combined effect of gravity and carrier bearing clearance was primarily responsible for unequal load sharing. Reducing carrier bearing clearance significantly improved load sharing, while reducing planet clearance did not. Normal tangential pin position error did not impact load sharing due to the floating sun design of this three-planet gearbox.

1 Introduction

Although the cost of energy from wind has declined tremendously during the past three decades (DOE, 2018), wind power plant operation and maintenance (O&M) costs are higher than anticipated and remain an appreciable contributor to the overall cost of wind energy. Wind power plant O&M averages $10 per megawatt-hour at recently installed wind plants, accounts for 20% or more of the wind power purchase agreement price, and generally increases as the wind plant ages (Wiser and Bolinger, 2017). Approximately half of the total wind plant O&M costs are related to wind turbine O&M (Lantz, 2013), and a sizeable portion of these costs is related to reliability of the wind turbine drivetrain (Kotzalas and Doll, 2010, Greco et al., 2013, and Keller et al., 2015).

Most of wind turbines installed in the United States utilize a geared drivetrain with a multi-stage gearbox including one or more planetary stages. These gearboxes must operate in a challenging, dynamic environment different from other industrial applications (Struggl et al., 2014). In general, wind turbine gearboxes are not achieving their expected design life (Lantz, 2013), even though they commonly meet or exceed the criteria specified in standards in the gear, bearing, and wind turbine industry as well as third-party certifications. ~~Although planet gear and bearing failures are not predominant (Sheng, 2017), they are extremely costly when they occur because they typically require replacement of the entire gearbox with a large crane.~~ Planet ~~gear and~~ bearing failures, although not the most frequent type of failure (Sheng, 2017), ~~are one of the common drivetrain reliability concerns (Sheng, 2017), and they~~ are extremely costly ~~when they occur~~ because they typically require replacement of the entire gearbox with a large crane and thus merit investigation.

In planetary gearboxes, equal load distribution between the planet gears and load distribution is required to achieve the predicted design life. Unequal load sharing between planetary gears due to manufacturing and assembly errors has been extensively examined in the past ~~two~~ three decades (Winkelmann, 1987; Lamparski, 1995; Predki and Vriesen, 2005; Cooley and Parker, 2014), with analytic models often validated through finite element models or experimental measurements at fixed locations or even on rotating gearing (Mo et al., 2016, Nam et al., 2016). More recently and specifically for wind turbine gearboxes, the ability of a floating sun gear to absorb the consequences of geometrical imperfections has been studied (Nejad et al., 2015, Iglesias et al., 2016). The effects of gravity and the drivetrain tilt angle on planet gear load-sharing and tooth-wedging behaviour were examined (Guo et al., 2014, Qiu et al., 2015). Gravity is an important factor as it introduces fundamental excitations in the rotating carrier frame of planetary gear sets. The effect of carrier bearing clearance on planetary load sharing subject to rotor moments has also been studied (Crowther et al., 2011, LaCava et al., 2013, Guo et al., 2014, Guo et al., 2015). Rotor moments impact planet load sharing, gear and bearing alignment, and bearing contact conditions and stress (Park et al., 2013, Gould and Burris, 2016, Dabrowski and Natarajan, 2017). Steady-state ~~Rotor~~ rotor moments and gravity result in a once-per-revolution ~~effects~~ variation in bearing load in the rotating carrier frame, ~~resulting in~~that both increases~~increased bearing loads and~~ fatigue and ~~reduced bearing loads potentially~~could ~~causing~~ cause wear or skidding ~~conditions~~ (Guo et al., 2014, Gould and Burris, 2016). Although it is generally agreed that a three-planet gear set with a floating central member has equal load sharing regardless of manufacturing errors (Cooley and Parker, 2014), in the wind turbine application, bearing clearance, gravity, and rotor moments can in fact cause unequal load sharing between planet gears.

Although load sharing between planetary gears has been examined extensively, the distribution of loads between the two or more bearing rows supporting each planet has not. In this paper, the load-sharing characteristics between the bearing rows supporting the planetary gears of two different wind turbine gearbox designs are examined and compared. This work extends previous works by the authors (LaCava et al., 2013, Guo et al., 2015, Keller et al., 2017a, Keller et al., 2017b) by examining a wind turbine gearbox planetary section supported by preloaded tapered roller bearings (TRBs) in addition to one supported by full complement and typical caged cylindrical roller bearings (CRBs) that operate in clearance. Loads predicted by design tools are compared to test measurements across a wide range of field-m~~representative~~easured loading conditions, and the resultant planetary section fatigue life for a duty cycle of a typical turbine is also compared. The physical phenomenon responsible for unequal load sharing of the planet bearings is identified.

2 Gearbox Design and Test Program

The National Renewable Energy Laboratory Gearbox Reliability Collaborative (GRC) has been investigating the root causes of premature wind turbine gearbox failures for over a decade. A modular 750-kilowatt wind drivetrain from a NEG Micon 750/48 wind turbine featuring a three-stage gearbox in a three-point mounted configuration, still representative of most utility-scale ~~wind turbine~~ drivetrain architectures, has been used for this effort, as shown in Fig. 1. In the three-point mounted configuration, the rotor and main shaft are primarily supported by a double-row spherical roller main bearing. The main shaft is connected to the planet carrier of the gearbox, which is supported by two torque arms that are mounted to the bedplate with elastomeric bushings. The two torque arms, along with the main bearing, provide a total of three points of support. The three-point mounted configuration transfers torque as well as rotor moments through the gearbox, which is an important design consideration (Guo et al., 2017).

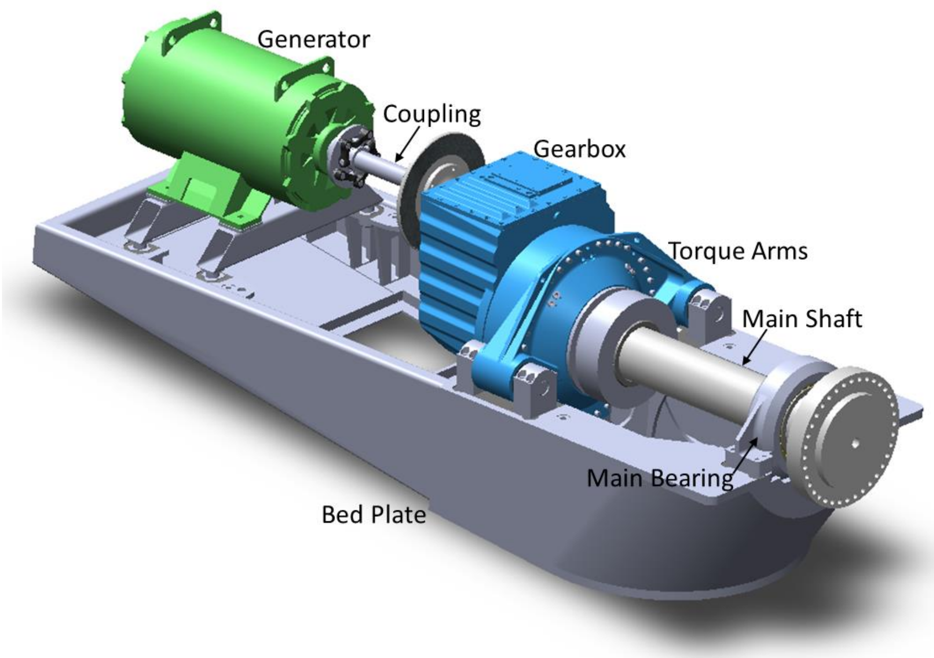


Figure 1: Gearbox Reliability Collaborative drivetrain configuration

The GRC gearbox design has a single input planetary stage followed by two parallel-shaft stages. The output shaft of the gearbox is connected to the generator with a flexible coupling. The rated rotor speed is 22.1 rpm, and with a ratio of 81.491, the gearbox increases the output speed to 1,800 rpm (Oyague, 2011, Link et al., 2011). The planetary stage features a floating sun to help equalize the load distribution among the three equally spaced planets, accomplished with a hollow low-speed shaft that has an internal spline connection to the sun pinion (Guo et al., 2013). Using this drivetrain and gearbox architecture, the GRC has investigated planetary gear and bearing failure modes and load-sharing characteristics through a dedicated research and test campaign. Two different gearbox designs were purposefully developed, manufactured, and tested. As shown in Fig. 2, their primary difference is the bearing types supporting the carrier and planet bearings. One design features planet CRBs with C3 clearance and full complement carrier CRBs with CN clearance, while the other features planet and carrier TRBs under preload.

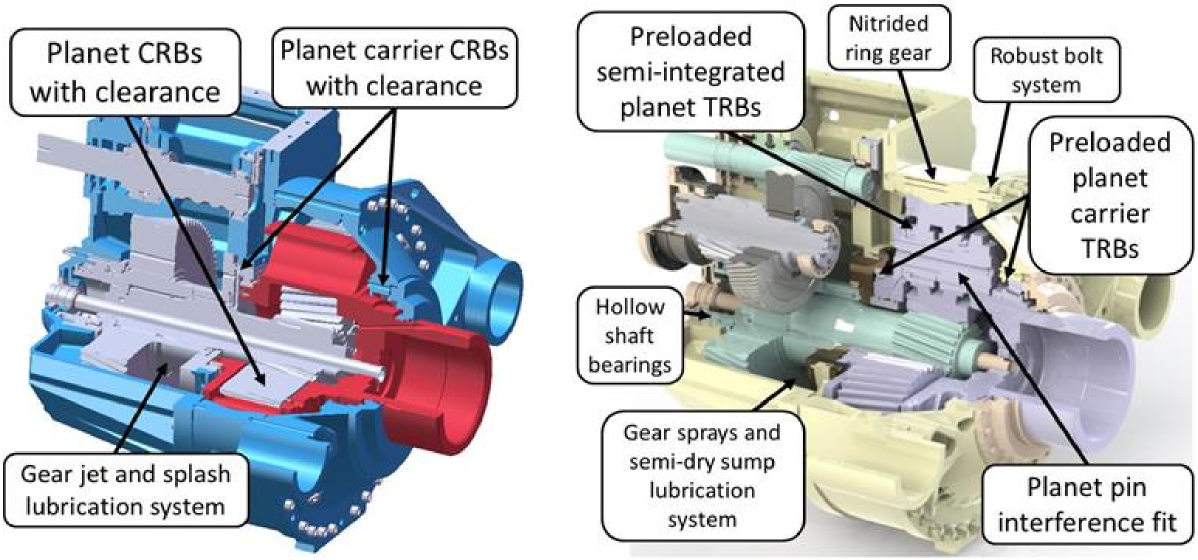


Figure 2. Comparison of the GRC gearbox designs featuring CRBs (left) and TRBs (right). Illustration by Romax Technology (right)

In many applications, a small preload, creating a small negative operating clearance, can optimize roller loads and maximize bearing life (Oswald et al., 2012). These preloaded bearings, along with interference-fitted planet pins, improve planet alignments and load-sharing characteristics. A semi-integrated planet bearing design also increases capacity and eliminates outer race fretting. Other than these planetary system changes, including updating gear tooth microgeometry, the gearbox designs are nearly identical. The front and rear housing components, originally from a commercially available Jahnel-Kestermann PSC 1000-48/60 gearbox, and intermediate and high-speed stage gearing are used in each gearbox. Physical parameters of the planetary bearings are given in Table 1. The models used to design the gearbox with TRBs indicated that it has over three times the planetary stage predicted L10 life compared to the gearbox with CRBs, like the projected increase in fatigue life in other industrial applications (Flamang and Clement, 2003, Lucas, 2005).

Table 1. Parameters of the planetary bearings

	Cylindrical Roller Bearings				Tapered Roller Bearings		
Location	Designation	Bore Diameter	Original Clearance	Modified Clearance	Designation	Bore Diameter	Mounted Preload

Carrier bearing, rotor side	NCF1892	460 mm	275 µm	170 µm	EE244180	457.2 mm	125 ± 25 µm
Carrier bearing, generator side	NCF1880	400 mm	235 µm	130 µm	L865547	381 mm	125 ± 25 µm
Planet bearings, upwind and downwind	NJ2232	160 mm	149 µm	92 µm	HH231649	139.7 mm	150 ± 50 µm

In two separate test campaigns, the gearboxes were mounted in the GRC drivetrain and installed in a dynamometer at the National Wind Technology Center, as shown in Fig. 3. Steady-state, constant-speed drivetrain operations were conducted throughout a range of power levels, from offline to the full 750-kilowatt electrical power and 325 kilonewton-meter (kNm) input torque. ~~Representative rotor p~~Vertical and lateral forces~~itch and yaw moments~~ were applied with hydraulic actuators to an adapter in front of the main bearing, resulting in bending moments up to~~up to ±300 kNm~~ ±300 kNm measured on the main shaft. This range of moments was derived from measurements on the same drivetrain when installed in a NEG Micon NM 750/48 turbine at an operational wind plant ~~were applied with hydraulic actuators based on field testing during power production~~ (Link et al., 2011). Unique to the GRC program is that all engineering drawings, models, and resulting test data are publicly available (Keller and Wallen, 2015, Keller and Wallen, 2017).

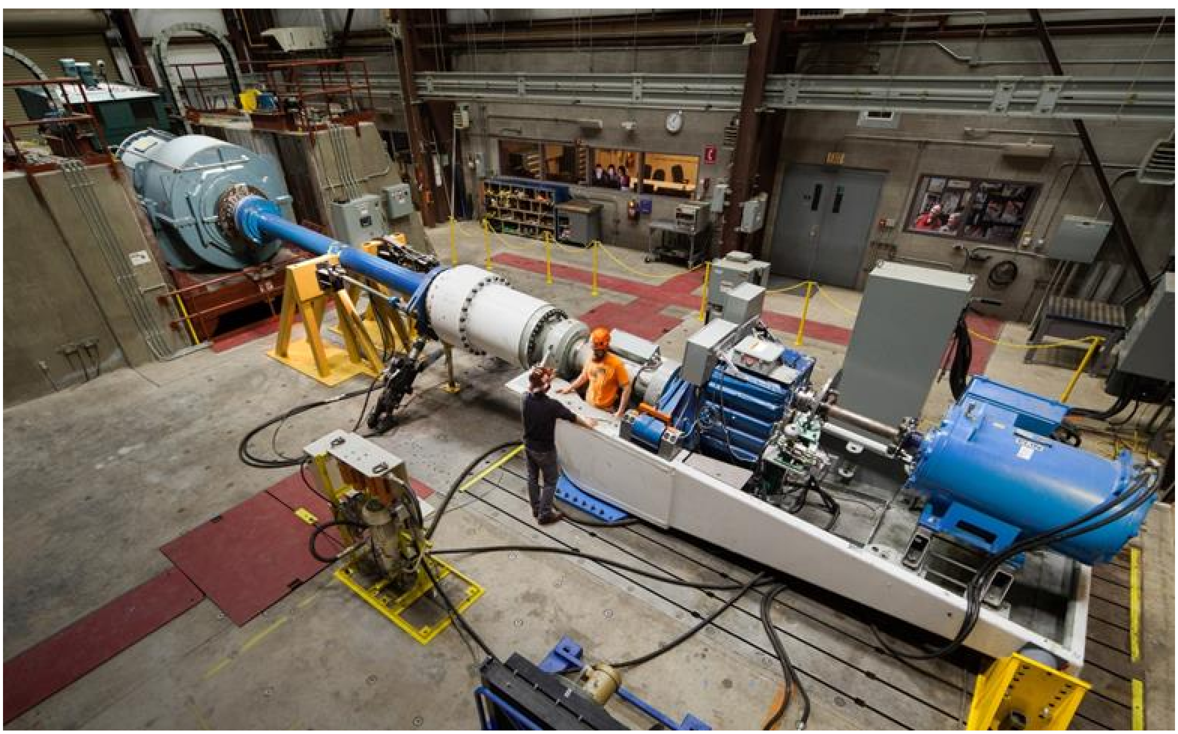
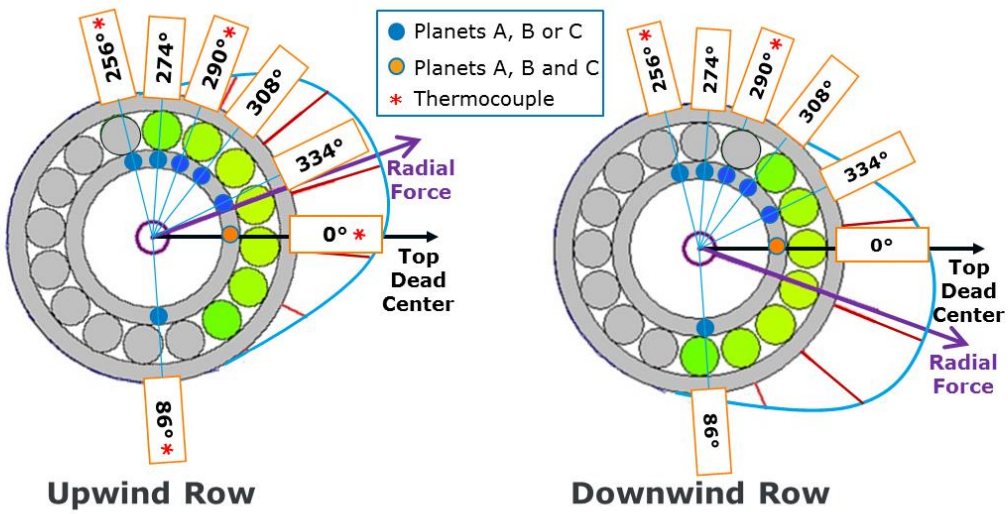


Figure 3. Installation of the GRC drivetrain in the dynamometer. Photo by Mark McDade, NREL 32734

Each gearbox was extensively instrumented, focusing primarily on planetary stage load-sharing characteristics. A total of 36 strain gage pair measurements were evenly placed between the upwind and downwind bearings of the three planets (A, B, and

C) for each gearbox. Most of the measurements were in the expected bearing load zones, as shown in Fig. 4. The helical planetary gearing causes an overturning moment on the planets, resulting in a ±20° offset of the centre of each load zone from the bearing top dead centre (TDC). The measurements were made at identical circumferential locations for the upwind and downwind bearings for the gearbox with CRBs. Two measurements were taken along the bearing inner-race width to investigate the axial load distribution between the upwind and downwind bearing rows (Link et al., 2011). Conversely for the gearbox with preloaded TRBs, the measurements focused on the circumferential load distribution with only one axial measurement on each bearing inner race. Additionally, one planet (B) has measurements at 10 circumferential locations per bearing row, 9 of which span the expected load zone. The other two planets (A and C) have measurements at four circumferential locations per bearing row (Keller and Wallen, 2017). The roller load at each measurement location is determined by converting the average strain range with calibration factors determined from dedicated bench tests (van Dam, 2011, Keller and Lucas, 2017). Several thermocouples were also installed the bearing inner races for each gearbox.



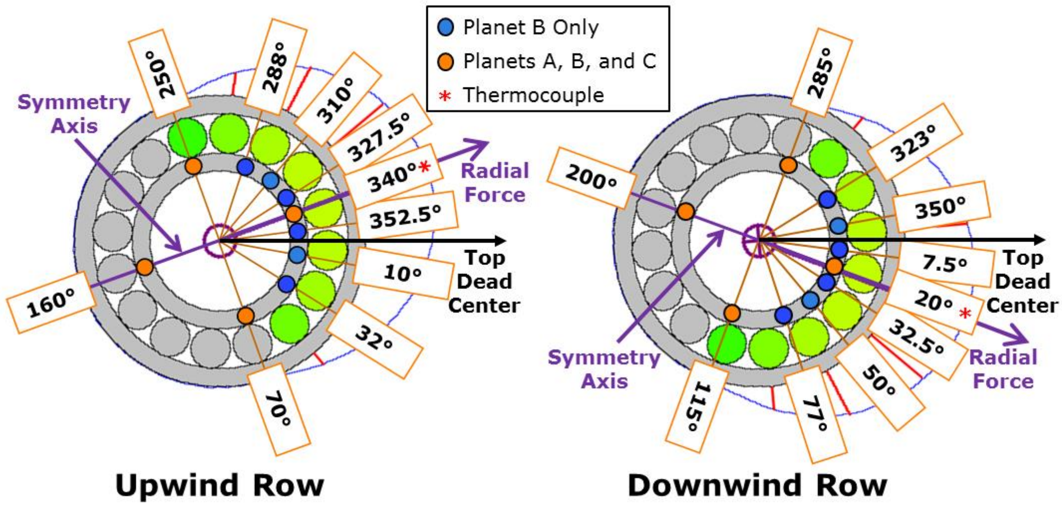


Figure 4. Planet bearing load measurements for the gearbox with CRBs (top) and TRBs (bottom)

3 Gearbox Modelling

~~The g~~Gearbox models were developed in two different finite-element, commercial software applications to predict planetary loads and load zones. The Transmission3D software application implements a three-dimensional, contact-mechanics model (Transmission3D, 2018). The ~~entire drivetrain~~gearbox is represented ~~as~~ with deformable bodies, including the ring gear and gearbox housing as their flexibility can affect gear misalignment and load sharing characteristics. ~~, computing g~~Gear~~,~~ and bearing contacts, including piece-wise clearance nonlinearities, are modelled with a hybrid of finite elements to predict far-field displacements and a Green's function model to predict displacements in the contact region. ~~The flexibility of gearbox housing including ring gear could potentially affect planet misalignment and, thus, affect the load sharing between planet bearing rows. It is essential to model the gearbox housing and ring gear as deformable bodies.~~ Known bearing clearances, preload, and pin position errors were included in the model. The RomaxWind software application implements a beam-finite element representation of shafts and solid-finite element representation of the gearbox housing, gear blanks, carrier, and torque arms (RomaxWind, 2018). The gears and bearings were modelled with semianalytical formulations that account for misalignment, area of contact under load, microgeometry, radial and axial clearances, and material properties. Static nonlinear analysis is performed for prescribed loading conditions, and the global deflections are solved simultaneously. Additionally, bearing modified L10 fatigue life calculations are made for a predetermined drivetrain torque, thrust, and ~~rotor~~ pitch and yaw moment spectrum (Keller et al., 2017a).

4 Results and Discussion

In this section, the planetary bearing loads and load sharing characteristics predicted by the models and measured in dynamometer tests are compared for both gearboxes ~~only for constant speed and torque, full-power cases~~. The fatigue life of planetary bearing designs is also calculated. Finally, several parametric design studies are examined for the gearbox with CRBs to understand the factors contributing to its load sharing characteristics.

4.1 Planet Bearing Load Zones

In this section, the bearing load zones for each gearbox are compared when the planet is at the bottom of the ring gear. The load zones for the pure torque condition are compared to ~~those for extreme positive and negative~~ those for the highest pitch moments. As shown in Fig. 5, the upwind planet CRB load zone increases in size as the applied pitch moment increases. In general, the upwind planet bearing supports up to twice the load of the downwind bearing. The downwind planet CRB load zone is not significantly affected by the applied pitch moment. The theoretical maximum roller load (Harris and Kotzalas, 2006) of approximately 45 kilonewtons (kN) for these bearings generally correlates with the measurements and model predictions.

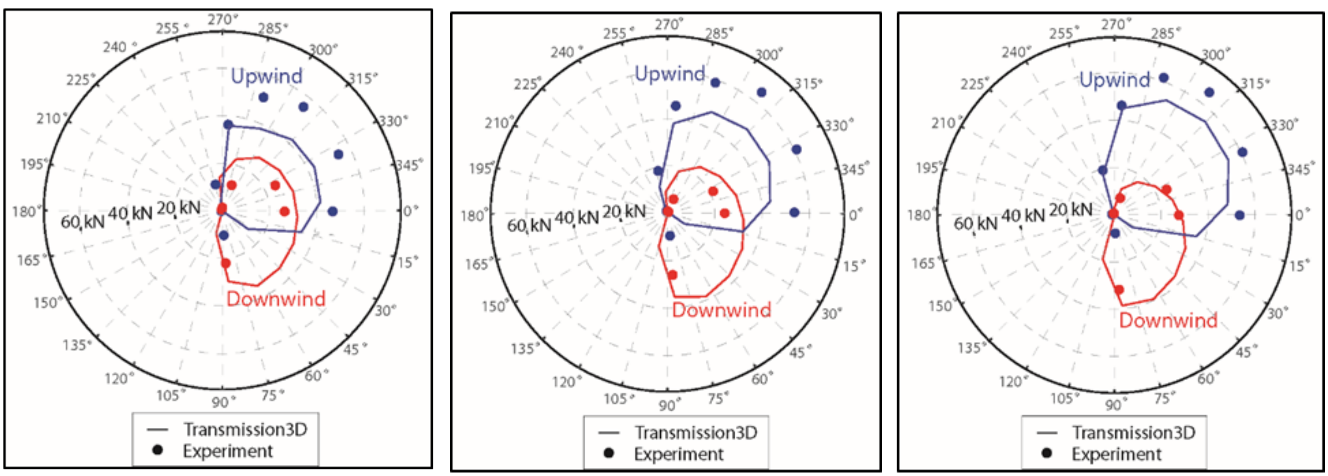


Figure 5. Planet CRB load zones with -300 kNm (left), pure torque (middle), and +300 kNm (right) pitch moments

In contrast, as shown in Fig. 6, the planet TRB load zones maintain their size and orientation regardless of the applied pitch moment. The more circular shape of the load zones reflects the preload in the bearings and the rigidity of the planetary system in general. The measured load zone magnitudes and orientations correlate well with the predictions, including the ±20° offset of the load zone from TDC. The theoretical maximum roller load (Harris and Kotzalas, 2006), also approximately 45 kN, again correlates with the measurements and predictions. The RomaxWind model~~s~~ assumes rigid bearing races while the Transmission3D ~~consider~~includes the flexibility of the race~~ways~~s. This results in a more circular load zone prediction for RomaxWind compared to an elliptical load zone for Transmission 3D ~~model predicts elliptical-like load zone shape compared to circular shape predicted by Romax. The elliptical load zone indicates the deflection of the corresponding raceway~~.

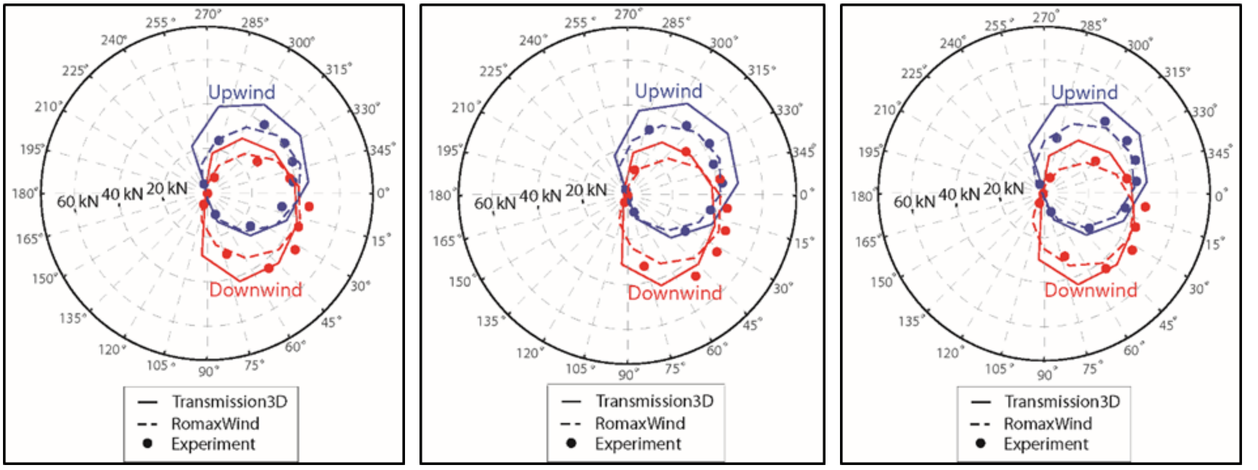


Figure 6. Planet TRB load zones with -300 kNm (left), pure torque (middle), and +300 kNm (right) pitch moments

4.2 Planet Bearing Loads

The upwind and downwind planet bearing loads can be calculated for each gearbox. 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. For the instrumented TRBs, a spline fit is then used to map the entire load zone and determine the total bearing load (Keller and Lucas, 2017, Keller et al., 2017b). The total load supported by both bearings, which is the vector summation of the upwind and downwind bearing loads, can also be calculated.

Figure 7 compares the measured and predicted loads—nondimensionalized by the assumed load (i.e., one-sixth or one-third of the load at the planet centre resulting from input torque)—over a complete revolution of the planet carrier for the pure torque condition. The 0° location indicates the planet is at the top of the ring gear in its rotation~~,~~ and the 180° location is at the bottom of the ring gear, which is the same position shown in Fig. 5 and Fig. 6. 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). The maximum measured load carried by the upwind bearing is 1.43, or 43% more than the assumed load. The minimum measured load carried by the downwind bearing is only 0.61, or 39% less than the assumed load. In this condition, the upwind bearing is accumulating more fatigue than expected; conversely, the downwind bearing has an increased risk of skidding for a portion of the carrier rotation. Because the bearing loads are nearly 180° out of phase, there is much less fluctuation in the total load than the individual row loads. The maximum total measured bearing load is only 6% greater than assumed. 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. The maximum and minimum measured row loads are only 12% different than assumed, whereas the maximum measured total bearing load is only 1% more than assumed. There is good agreement between these measured loads and those predicted by Transmission3D for both gearboxes.

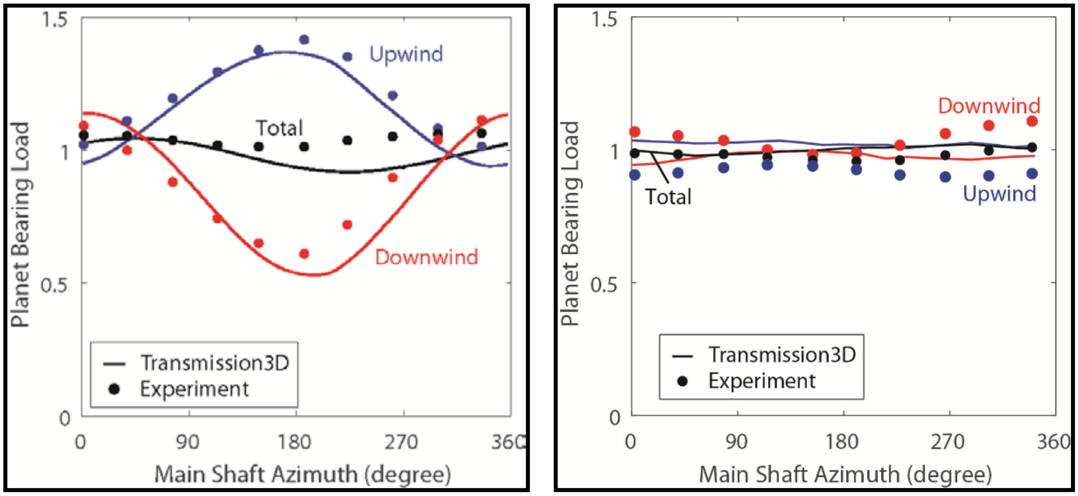


Figure 7. Planet CRB (left) and TRB (right) loads in pure torque

In contrast, Fig. 8 compares the same loads but with a~~n extreme~~ large negative pitch moment. The measured upwind CRB load is relatively constant over the rotation but 25% greater than assumed. The downwind load behaviour is very similar to the pure torque condition. The net effect is that the total measured bearing load fluctuates slightly more than the pure torque condition and 15% more than assumed. The measured TRB loads again fluctuate very little—only 8% for the downwind row and 2% for the total bearing load.

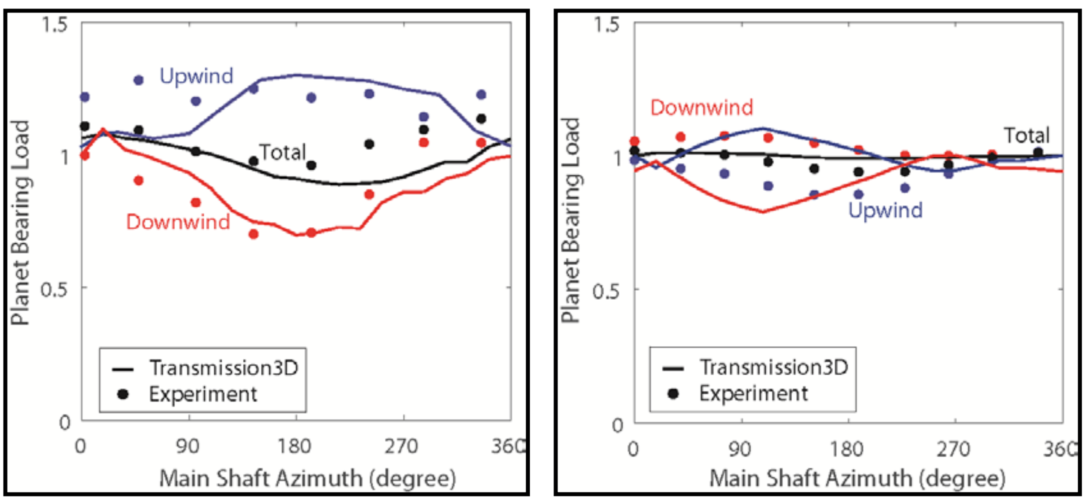


Figure 8. Planet CRB (left) and TRB (right) loads with -300 kNm pitch moment

Figure 9 summarizes the measured upwind and downwind planet CRB loads for all the pitch moment cases. The pitch moment changes the upwind bearing loads significantly; however, it does not affect the downwind bearing loads at all. The behaviour of the upwind bearing loads can be separated into three categories. Pure torque and positive pitch moments all essentially have the same effect, resulting in the largest variation over the carrier rotation and overall magnitude in the upwind bearing load.

Conversely, pitch moments beyond -200 kNm elevate the mean upwind bearing load with much less fluctuation over the carrier rotation. The -100 kNm pitch moment case is a transition between these two categories.

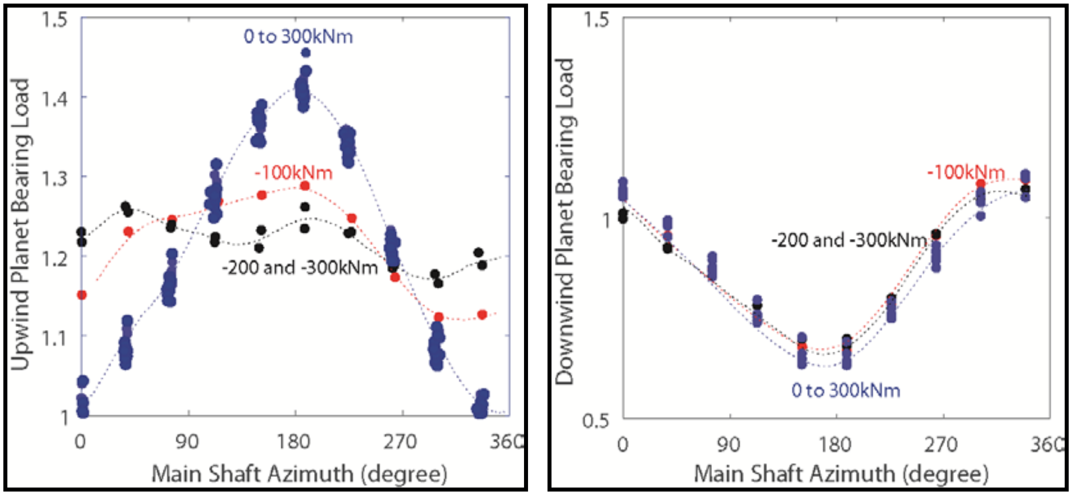


Figure 9. Upwind- (left) and downwind-measured (right) planet CRB loads

The bearing loads shown in these figures contain both a constant difference from the assumed load and a fluctuating component. 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).

4.3 Planet Bearing Load Sharing

The accurate estimation of planet bearing loads is a crucial step in calculating the planetary load-sharing factor, also called the planetary mesh load factor ($K\gamma$). Ideally, all planets share torque equally and the planetary mesh load factor equals 1. However, because of positional-type errors and variations in tooth stiffness, International Electrotechnical Commission standard 61400-4 assumes this factor is 1.1 for three-planet wind turbine gearboxes. 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.~~In this study, the maximum load throughout the main shaft rotation, which accounts for both constant load differences and the fluctuating load from gravity and rotor moments, as shown in Fig. 7–9, is examined for comparison to this assumption~~.

Figure 10 compares the maximum individual bearing row load and maximum total bearing load for both gearboxes over the complete range of ~~rotor~~ pitch moments. The maximum total measured CRB load ranges from 1.07 in pure torque to just over 1.15 for large negative pitch moments, very close to the assumed planetary mesh load factor of 1.1

. However, as shown previously, the measured CRB load carried by the upwind bearing far exceeds this, reaching 1.43 on average and as high as 1.46 in one test. Counterintuitively, this highest load occurs in the pure torque condition and is not affected by increasing the pitch moment, also as demonstrated in Fig. 9. The upwind measured CRB load does decrease with negative pitch moments; however, it never falls below 1.26. The wide variation in the maximum CRB load can be contrasted to the consistency in the maximum TRB load. In general, the maximum TRB loads are all much closer to the assumed planetary mesh load factor of 1.1. The maximum measured downwind TRB load is 1.13 in pure torque and no more than 1.17 even for a~~n extreme~~ large positive pitch moment—much lower than the maximum CRB load. A significant reduction of the maximum loads and improvement in load sharing was achieved with the design changes in the gearbox with TRBs.

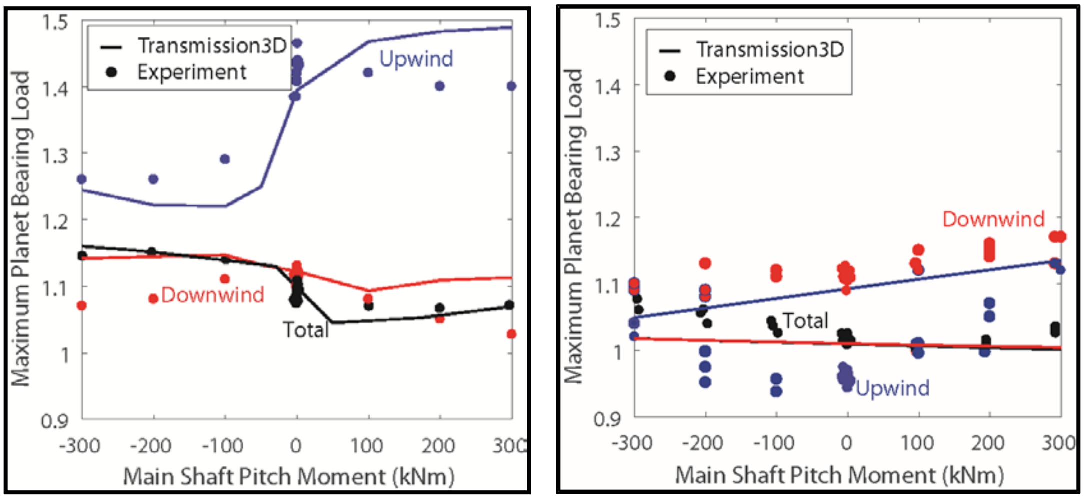


Figure 10. Maximum planet CRB (left) and TRB (right) loads for all pitch moments

To better understand the planet bearing load-sharing behaviour shown in Fig. 10, the effect of ~~rotor~~ pitch moments ~~is explored~~ on carrier bearing loads is explored in Fig. 11. Here only the predicted loads from the model are available; measurements of the carrier bearing loads were not acquired in tests. Carrier bearing loads are also nondimensionalized by the average of the assumed total planet bearing load. 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~~The upwind carrier CRB does not carry any load regardless of the pitch moment and neither does the downwind carrier CRB for pitch moments within ±100 kNm because of their clearances.~~ ~~*When the moments are less than 100kNm, raceway displacement of the carrier bearing is still less than the internal clearance. Thus, carrier bearings are out of contact under this condition.*~~ ~~In between ±100 kNm pitch moment, the planet CRBs are carrying all the loads. The upwind CRB has large clearance than the downwind CRB (listed in Table 1), resulting in only downwind CRB reacted to moment and gravity loads.~~ In contrast, because of their preloaded condition both the upwind and downwind carrier TRBs support loads for any applied pitch moment. It is clear from Fig. 10 and Fig. 11 that for the gearbox with CRBs, ~~rotor ~~pitch moments can relieve the gravity load from the main shaft and planetary system from the downwind carrier CRB and shift it to the planet CRBs. However, for the gearbox with TRBs, ~~rotor ~~pitch moments are essentially entirely reacted by the carrier bearings. In the three-point mount drivetrain configuration, the carrier bearing is expected to be part of the load path from the wind turbine rotor to the bed plate. In this ideal situation, the planetary gear system carries only torque and is not impacted by other loads, resulting in improved load sharing between planets and upwind and downwind rows. From this analysis, the planet carrier TRBs are carrying the moment loads as expected, whereas the planet carrier CRBs are not.

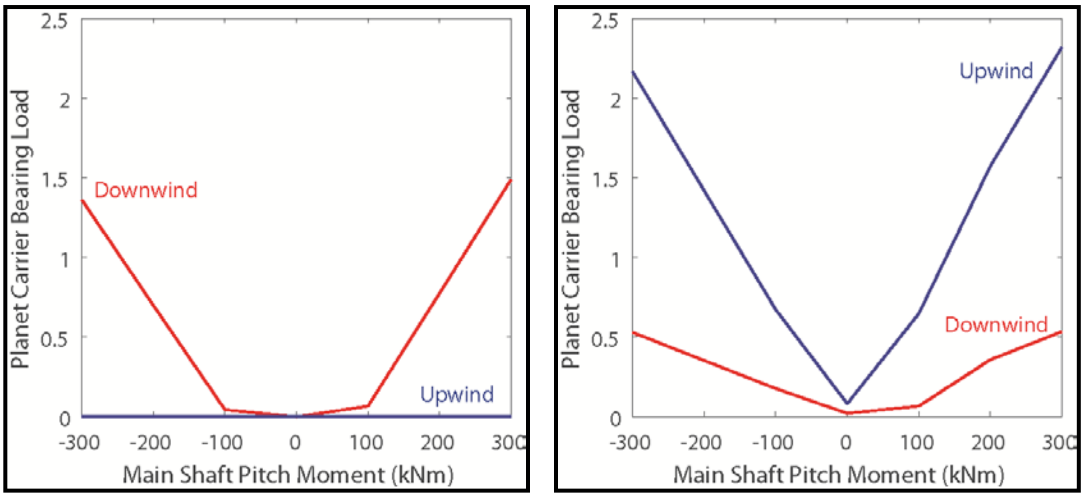


Figure 11. Maximum carrier CRB (left) and TRB (right) loads for all pitch moments

For comparison to ~~rotor ~~pitch moments, Fig. 12 shows the planet loads over the full range of ~~rotor ~~yaw moments for both gearboxes. The maximum upwind CRB loads occur again in pure torque conditions. Both positive and negative yaw moments decrease the maximum upwind CRB load slightly; however, the measured load remains above 1.35. The total measured bearing load follows a more intuitive pattern, in which it is a minimum of 1.08 at pure torque and increases slightly to 1.13 with either positive or negative yaw moments. ~~Rotor y~~Yaw moments have little effect on any of the TRB loads. The maximum measured load of 1.13 occurs for the downwind bearing for a positive yaw moment.

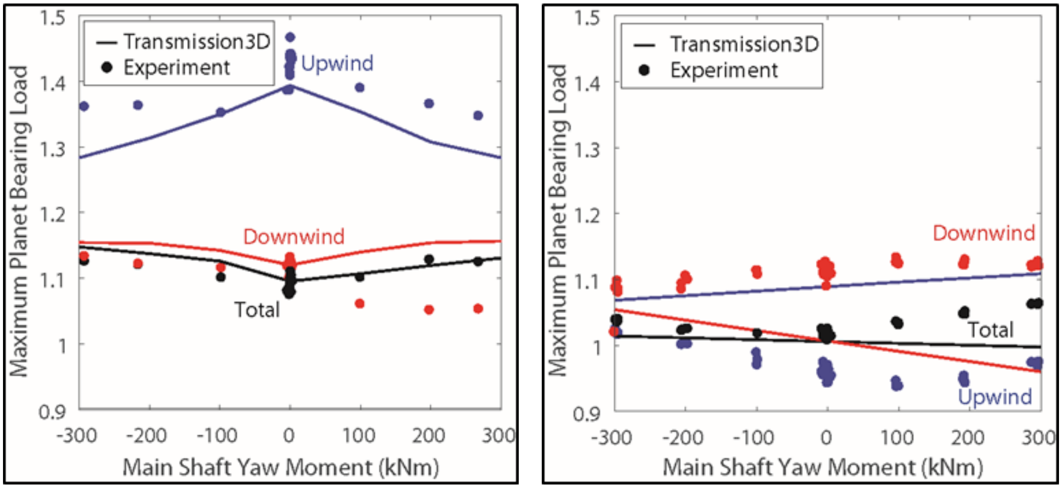


Figure 12. Maximum planet CRB (left) and TRB (right) loads for all yaw moments

Since the largest disparity in load sharing is evident even in pure torque conditions, Fig. 13 examines the maximum individual bearing row load and maximum total bearing load for both gearboxes over the complete range of pure torque conditions tested. Generally, the load increases as torque decreases – although it increases more for the CRBs than the TRBs. The maximum measured total planet bearing load increases from approximately 1.1 at full torque to 1.3 at 25 % torque for both gearboxes. However, the measured CRB load carried by the upwind bearing increases from 1.43 on average in pure torque to as high as 1.85 at 25% torque. In contrast, the maximum measured downwind TRB load increases from 1.13 in pure torque to just 1.40 at 25% torque.

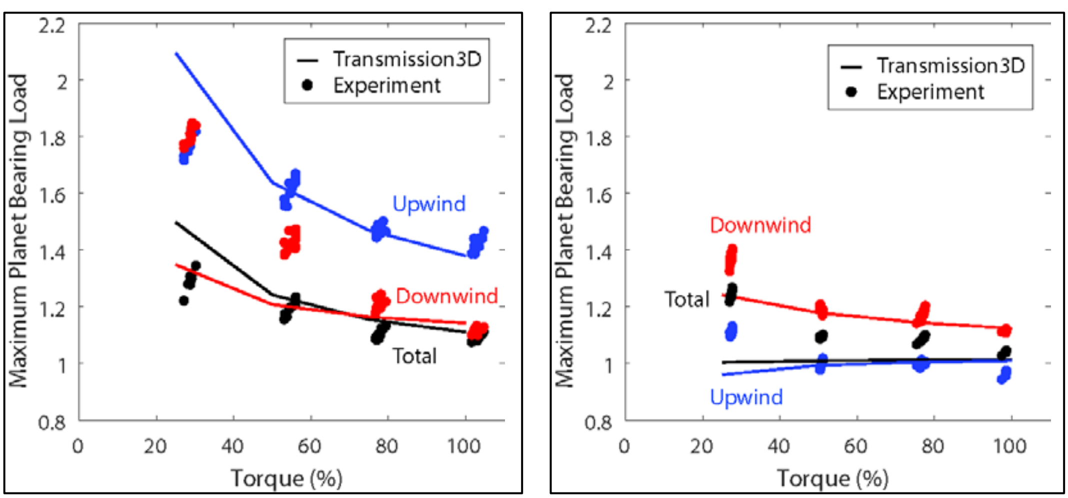


Figure 13. Maximum planet CRB (left) and TRB (right) loads for all torque

As shown in this section, the load sharing characteristics of the planet TRBs were significantly improved compared to the planet CRBs. However, the use of preload in these bearings does raise the question of their temperature characteristics. The measurements from the thermocouples on the bearing inner races for each gearbox, with respect to the gearbox sump temperature, are examined in Fig. 14. In this figure, the average of all thermocouples on each of the planets is examined for the full range of gearbox operating torque and applied ~~rotor~~ moments. The planet bearing temperatures are approximately 5 °C cooler than the sump temperatures for both gearboxes. There is little to no difference in the temperature of the planet bearing inner races between the two gearboxes and thus most likely little to no impact on the gearbox efficiency. This is not necessarily a surprise as the planets are spinning at a relatively low speed compared to the bearings supporting the intermediate stage and 1,800 rpm output shaft of the gearbox. These higher speed bearings generate significantly more heat and cause the gearbox sump temperature to be higher than the planet bearing operating temperature. For reference, the absolute temperatures of the planet bearings for each gearbox were in the range of 50 to 65 °C, while the gearbox sump temperature typically ranged from 55 to 70 °C.

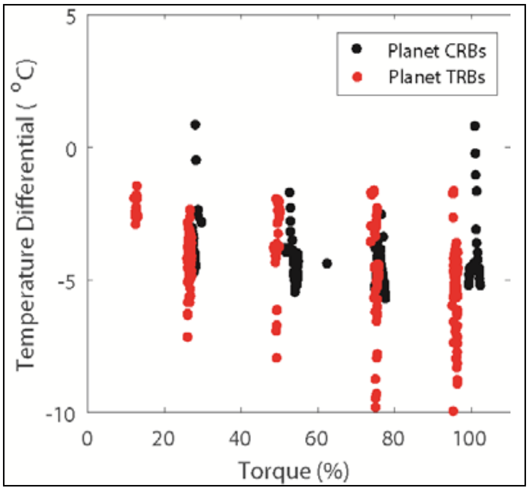


Figure 14. Differential between the gearbox sump temperature and the average of planet bearing inner ring temperatures

4.4 Planet Bearing Fatigue Life

The predicted planetary fatigue life for each gearbox was calculated using a representative drivetrain torque and ~~rotor~~ pitch and yaw moment spectrum derived from field measurements (Keller et al., 2017b). The modified bearing L10 life was calculated per Deutsches Institut für Normung International Organization for Standardization 281 Beiblatt 4 (now superseded by International Organization for Standardization technical specification 16281), including a systems life modification factor. As shown in Fig. 15, the average fatigue life of the upwind planet bearing was increased by a factor of six using the TRBs when compared to the CRBs, in addition to a smaller life extension for both the upwind and downwind planet bearings due to the larger bearing capacity in the semi-integrated design. The modified L10 life for the eight total planetary bearings (all six planet bearings and two carrier bearings) is also shown, combined using a Weibull slope of 1.125 (Zaretsky et al., 2007). The planetary stage bearing predicted fatigue life has been increased by a factor of 3.5 using the TRBs when compared to the CRBs. The overall planetary bearing stage life is driven by the lowest-life components, which in this case are the planetary bearings in both gearboxes. The carrier bearings have much longer fatigue life and thus are not shown individually.

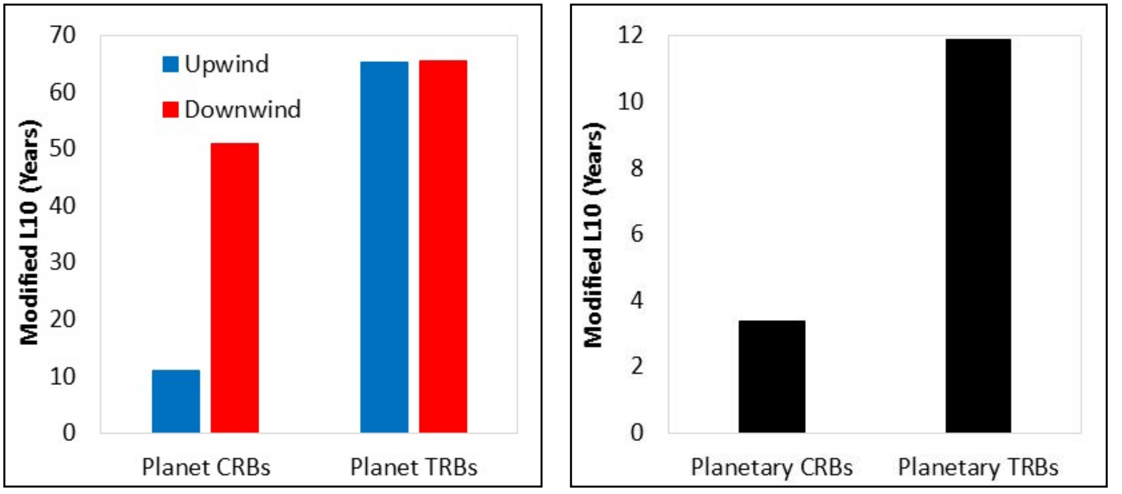


Figure 15. Fatigue life for the planet bearings (left) and planetary bearing stage (right)

4.5 Parametric Studies

The previous section examined planetary load-sharing characteristics in detail. The upwind and downwind planet bearing loads were not shared equally in the gearbox with CRBs, even in pure torque conditions. In this section, the major factors responsible for the disturbed load sharing in this gearbox are examined through parametric studies of bearing clearances, gravity, and pin position error.

4.5.1 Effect of Bearing Clearances

The planet CRB loads were predicted with reduced clearance settings in the carrier and planet CRBs individually, as listed in Table 2, for comparison to the original model clearances. As shown in Fig. 16, reducing the carrier CRB clearance from CN to C2 resulted in a noticeable improvement in load-sharing characteristics for both the upwind and downwind bearings, especially for positive pitch moments. For example, the predicted upwind CRB load decreases from 1.49 to 1.22 at +300 kNm pitch moment, a reduction of 18%. In contrast, reducing the planet CRB clearance from C3 to CN did not significantly reduce the upwind planet CRB loads for positive pitch moments, and it increased the downwind planet CRB loads.

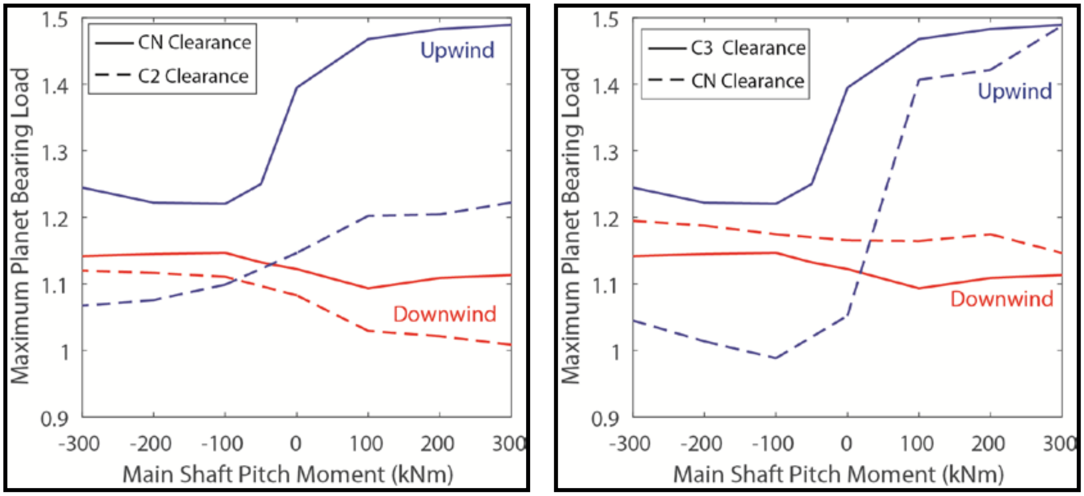
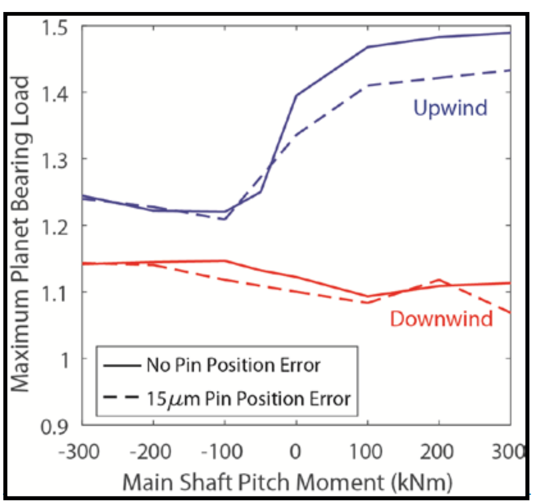


Figure 16. Effect of carrier (left) and planet (right) bearing clearance on maximum planet CRB loads

~~4.5.2 Effect of Pin Position Error~~

~~Tangential pin position error is one of the more common manufacturing deviations that is known to affect planetary load sharing (Cooley and Parker, 2014). In this parametric study, the effect of a 15 µm tangential pin position error – a magnitude commonly considered in other applications (Singh, 2009) – on load sharing of the gearbox with CRBs is assessed. As shown in Fig. 17, the pin position error only changes the upwind planet CRB loads by less than 4% for positive pitch moments. This difference is almost negligible compared to the load fluctuations caused by other factors, which agrees with analytical results (Singh, 2009). It does not significantly change the upwind planet CRB loads in pure torque or negative pitch moments or downwind CRB loads. Ideally, pin position error should not disturb load sharing with an adequately floating sun for a three-planet gearbox such as the GRC test article.~~



~~**Figure 17. Effect of tangential pin position error on maximum planet CRB loads**~~

4.5.~~3~~ 2 Effect of Gravity

As shown previously, the interplay between the gravity load from the main shaft and planetary system and the ~~rotor~~ pitch moment has a significant effect on both the planet and carrier CRB loads. Figure 17~~8~~ examines this further by eliminating gravity from the model. Gravity has a significant influence on the planet CRB loads, such as the effect of carrier CRB clearance. Without gravity, the upwind planet CRB load reduces dramatically anywhere from 0.20 to 0.37 over the entire range of pitch moments, including a reduction from 1.40 to 1.07 at pure torque. The effect of gravity on the downwind planet CRB load is small. The effects of gravity, planetary clearances, and nontorque loads on three-point mounted wind turbine gearboxes are unavoidable and should be considered in their design. The effect of gravity on planet bearing loads can be mitigated by using carrier bearings with reduced clearances if possible.

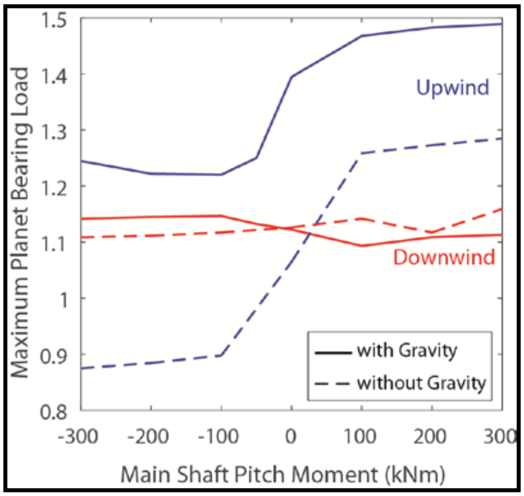


Figure 17. Effect of gravity on maximum planet CRB loads

4.5.3 Effect of Pin Position Error

Tangential pin position error is one of the more common manufacturing deviations that is known to affect planetary load sharing (Cooley and Parker, 2014). In this parametric study, the effect of a 15 µm tangential pin position error—a magnitude commonly considered in other applications (Singh, 2009) —on load sharing of the gearbox with CRBs is assessed. As shown in Fig. 18, the pin position error changes the upwind planet CRB loads for only positive pitch moments by less than 4%. This effect is much smaller than the load fluctuations caused by other factors, which agrees with analytical results (Singh, 2009). It does not significantly change the upwind planet CRB loads in pure torque or negative pitch moments or downwind CRB loads. Ideally, pin position error should not disturb load sharing with an adequately floating sun for a three-planet gearbox such as the GRC test article.

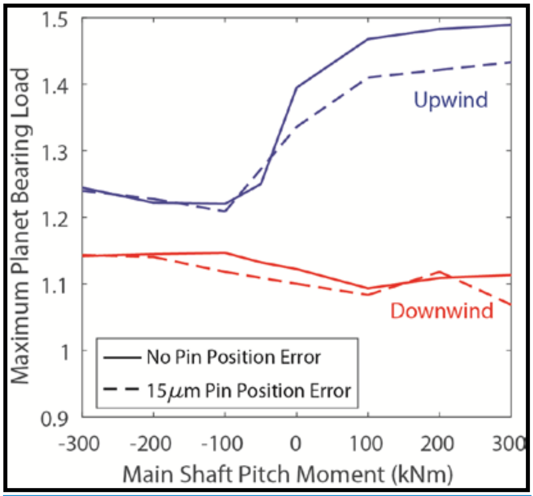


Figure 18. Effect of tangential pin position error on maximum planet CRB loads

5 Conclusions

This study compared two wind turbine gearbox planetary bearing system designs, a conventional design representing most of the gearboxes used in three-point mounted drivetrains and a new design tailored for increased planetary bearing fatigue life. These two designs differ in the choice of carrier and planet bearings. The first design uses planet CRBs with clearance and the latter utilizes preloaded TRBs. Both gearboxes were designed, built, and instrumented and then tested in a dynamometer under the same set of controlled loading conditions, including ~~rotor~~ pitch and yaw moments. The resulting planet bearing load measurements were correlated with predictions from finite-element models of both gearboxes.

The gearbox design using CRBs with clearance did not demonstrate equal load sharing between the planet CRBs, with one bearing supporting up to 46% more load than expected. This unequal load sharing occurred, counterintuitively, in pure torque conditions as well as for positive pitch moments. The gearbox design with preloaded TRBs demonstrated improved planetary load-sharing characteristics compared to the gearbox with CRBs. The preloaded TRBs significantly reduced the planet bearing loads from a maximum of 1.46 to 1.17, a 20% reduction, in pure torque conditions. Furthermore, ~~rotor~~ pitch and yaw moments did not significantly affect the upwind and downwind TRB row loads. Parametric studies indicate that the unequal load sharing in the gearbox with CRBs is primarily a result of the combined effects of gravity, ~~rotor~~ pitch and yaw ~~moments, and~~moments and bearing clearances and can be substantially improved by reducing clearance in the carrier bearings. This reduction and equalization in planet bearing loads, along with slightly larger capacity bearings through a semi-integrated design, resulted in a modified L10 life 3.5 times greater for the gearbox with preloaded TRBs than for the gearbox with CRBs.

Acknowledgements

This work was authored in part by the National Renewable Energy Laboratory, operated by Alliance for Sustainable Energy, LLC, for the U.S. Department of Energy (DOE) under Contract No. DE-AC36-08GO28308. Funding provided by the U.S. Department of Energy Office of Energy Efficiency and Renewable Energy Wind Energy Technologies Office. The views expressed in the article do not necessarily represent the views of the DOE or the U.S. Government. The U.S. Government retains and the publisher, by accepting the article for publication, acknowledges that the U.S. Government retains a nonexclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this work, or allow others to do so, for U.S. Government purposes~~This work was supported by the U.S. Department of Energy (DOE) under Contract No. DE-AC36-08GO28308 with the National Renewable Energy Laboratory. Funding for the work was provided by the DOE Office of Energy Efficiency and Renewable Energy, Wind Energy Technologies Office~~.

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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

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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)

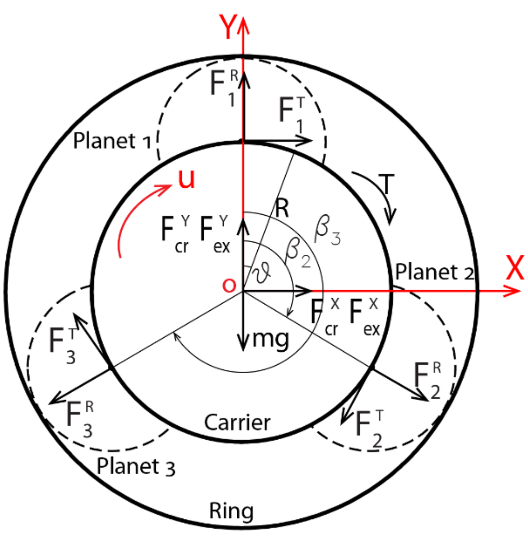
$\sum_{i=1}^{n} F_i^T - \frac{T}{R} = 0$ (3)

where n is the total number of planets and θ denotes the rotation angle of the carrier. $\beta_i . i = 1, \dots n$ 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.

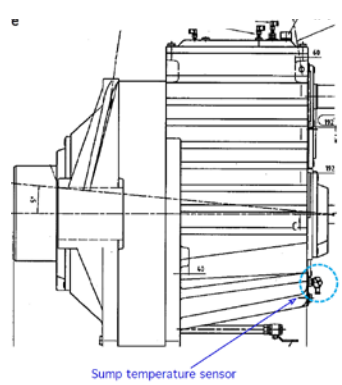
Comment #11: "figure 11: unstetic behaviour of CRB at +/-100 kNm Needs Explanation, dito for upwind load at Zero pitch Moment. Model achitecture should be explained via sketch and text.."

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.

Reviewer 2: A. Natarajan

Comment #1: “Page 2, lines 1-4; One sentence states that "wind turbine gearboxes are not achieving their expected design life" and the next sentence seems to negate this by stating "Although planet gear and bearing failures are not predominant...". Re-phrase by stating specifically what is the problem that is being addressed here.”

Authors’ response to comment #1: For clarity, the latter sentence has been changed to:

Planet bearing failures, although not the most frequent type of failure (Sheng, 2017), are extremely costly because they typically require replacement of the entire gearbox with a large crane and thus merit investigation.

Comment #2: “Page 2, lines 15-16, "Rotor moments and gravity result in once-per-revolution effects...". Are rotor moments only 1P in frequency? Don’t you have any higher frequency components from the rotor passed to the main shaft?”

Authors’ response to comment #2: It is certainly true that rotor moments vary in time and in frequency. This sentence was originally intended to explain the simplest situation where even steady-state rotor moments (in the fixed frame, like those applied in dynamometer testing) and gravity (naturally constant in the fixed frame) result in 1P effects (in the carrier frame). Steady moments are the only ones that have been examined in this literature review section. For clarity, the sentence has been changed to:

Steady-state rotor moments and gravity result in a once-per-revolution variation in bearing load in the rotating carrier frame, that both increases fatigue and could cause wear or skidding (Guo et al., 2014, Gould and Burris, 2016).

Comment #3: “Page 5, Line 2, "Representative rotor pitch and yaw moments up to _300 kNm....". Under what situations is this a representative load? Are these for normal operating conditions under turbulence?”

Authors’ response to comment #3: The moments are based on a 3-month field test of this drivetrain when it was installed in a NEG Micon 750/48 turbine at a wind plant in Wyoming, USA. Moments this high were measured, but not frequently – the majority were under ±300 kNm. This is accounted for later in the L10 life calculations. Stating that these highest 300 kNm moments are "representative" might imply that they are common, which is not the case. This sentence has been revised to include some additional clarification and detail as follows:

Vertical and lateral forces were applied with hydraulic actuators to an adapter in front of the main bearing, resulting in bending moments up to ±300 kNm measured on the main shaft. This range of moments was derived from measurements on the same drivetrain when installed in a NEG Micon NM 750/48 turbine at an operational wind plant (Link et al., 2011).

Comment #4: “Page 6, Line 7, "The entire drivetrain is represented as deformable bodies...". Is this really required? Provide some explanation on what role do the housing stiffness, ring gear stiffness etc. play on the loads on the bearing?”

Authors’ response to comment #4: The "entire drivetrain" being modeled with deformable bodies is a mistake that has been fixed. Other clarifications were also made, changing these sentences to:

The Transmission3D software application implements a three-dimensional, contact-mechanics model (Transmission3D, 2018). The gearbox is represented with deformable bodies, including the ring gear and gearbox housing as their flexibility can affect gear misalignment and load sharing characteristics. Gear and bearing contacts, including piece-wise clearance nonlinearities, are modelled with a hybrid of finite elements to predict far-field displacements and a Green’s function model to predict displacements in the contact region.

Comment #5: “Page 7, Line 10, "load zones for the pure torque condition are compared to those for extreme positive and negative pitch moments". Where do you get the extreme pitch moments from? I don’t think _300 kNm are extreme moments, if that is what is referred to here.”

Authors’ response to comment #5: As stated earlier, the range of main shaft moments were derived from field tests. For clarity, the sentence has been changed to:

The load zones for the pure torque condition are compared to those for the highest pitch moments.

Comment #6: “Page 9, Figure 7: Why should the loads on the upwind and downwind bearing be out of phase for a pure torque condition that is shown here? The explanation given is due to clearances, but possibly there is additional loading than pure torque?”

Authors’ response to comment #6: The gravity load is the additional load. The difference in phase between the upwind and the downwind loads is a direct result of the difference in size of the load zones in Fig. 5. At the 180 degree location, the larger upwind load zone results in a higher load while the smaller downwind load zone results in a lower load. The authors are actually developing a separate journal manuscript which examines this behavior analytically, but for now just refer to previous modeling studies. For additional clarification, the sentence has been changed to:

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 #7: “Page 10, Figure 9, What about the main bearing load? Does the main bearing hold a part of this pitch moment load?”

Authors’ response to comment #7: Rotor moments can certainly be carried by both the main bearing and low-speed stage of the gearbox, depending on their relative clearances and stiffnesses. As mentioned in the response to comment #3, the moments discussed herein were actually measured on the main shaft and so are related but not equal to the rotor moments. The authors have made minor changes in the manuscript text to distinguish between the rotor moment itself, which then results in the main shaft pitch and yaw moments that were used during the tests. Note that the figures in question use this easily measurable quantity, “Main Shaft Pitch Moment” and “Main Shaft Yaw Moment”, on the x-axis. The main spherical bearing in this case most likely supports a minor and unknown moment, but this uncertainty is not relevant as all moments were measured on the main shaft – including the field tests.

No changes were made to the manuscript.

Comment #8: “Page 11, line 11: "The upwind carrier CRB does not carry any load regardless of the pitch moment and neither does the downwind carrier CRB for pitch moments within _100 kNm because of their clearances". This is not clear.”

Authors’ response to comment #8: The authors agree that this sentence was confusing. The section has been modified and also better linked to the previous result:

To better understand the planet bearing load-sharing behaviour shown in Fig. 10, the effect of pitch moments on carrier bearing loads is explored in Fig. 11. Here only the predicted loads from the model are available; measurements of the carrier bearing loads were not acquired in tests. Carrier bearing loads are also nondimensionalized by the average of the assumed total planet bearing load. *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 #9: “Paqe15. Effect of bearing clearances: Overall all load effects shown are explained through the effect of clearances. If this is indeed the case, then the initial sections of this manuscript should explain the clearances over the different parts of the gearbox and discuss their modeling in the software.”

Authors’ response to comment #9: The planetary bearing types and clearances were described earlier in the paper in Table 1. The description of the “*piece-wise*” CRB clearance modeling has been added to the modeling section as mentioned in the response to comment #4.

Comment #10: “Page 16, line 5: 15 μm tangential pin position error is investigated. How is this done in practice? Is this simulated or measured? Why are the load results in Fig. 17 said to negligible? They seem significant for such a small error.”

Authors’ response to comment #10: Pin position error was simulated in the model. There is no direct measurement of the error itself. Pin position error commonly exists during the manufacturing process, as stated in the introduction. The authors agree that describing the effect of pin position error as “negligible” is an overstatement, so this sentence has been changed to:

This effect is much smaller than the load fluctuations caused by other factors,….

Comment #11: “Page 18, Line 3: The conclusions state that "resulted in a modified L10 life 3.5 times greater for the gearbox" . I don’t think the cases simulated here are representative enough to compute the L10 life directly. If they are claimed to be so, then that should be substantiated with some load simulations of normal operation. Otherwise the conclusions should just focus on the effects of clearances and gravity on loads and not extrapolate to the L10 life."

Authors' response to comment #11:

As mentioned earlier, the loading conditions were measured in field tests over a 3-month field test. Although brief, this test period was used to calculate a representative duty cycle for the turbine. It was this duty cycle that was then used to make the L10 life calculations. The life calculations are based on this whole duty cycle, not just the extreme load cases presented here. The duty cycle is actually comprised of mostly low or near pure-torque conditions, and indeed the gearbox with CRBs shows disturbed load-sharing even in what is thought of as this "benign" (or even best case) condition. Regardless of how accurate the duty cycle may be, what is important to quantify is the relative difference in L10 life between the two gearboxes (which is significant) rather than just stopping at an examination of the difference in planet bearing loads.

No changes were made to the manuscript.

Reviewer 3: H. Polinder

Comment #1: "The gearbox used in this paper was originally designed for a 750 kW wind turbine that was probably introduced about 20 years ago. Current wind turbines have torques that are more than an order of magnitude larger. What does this mean for the relevance of the paper? How do these effects scale? How has gearbox technology (gears and bearings) developed since then?"

Authors' response to comment #1: This historical perspective and future outlook is certainly true. However, for many modern turbines the drivetrain architecture (3-point) and gearbox planetary design (3-planets supported by CRBs and a floating sun) have not changed and are still commonly used. For larger turbines, the torque has increased but so has the rotor moments and size (mass) of the gearbox. It is anticipated that the planetary load sharing problem persists for modern, large wind turbine gearboxes. The authors are actually developing a separate journal manuscript which examines this behavior analytically – the formulation itself being worthy of a journal manuscript in our estimation. For a given torque, rotor moment and mass for a larger drivetrain this question could be examined in more detail. But, we felt that it was beyond the scope of the present paper.

No changes were made to the manuscript.

Comment #2: "To me, it looks like the correlation between measurements and models is reasonable, but not excellent, while the authors characterize the correlation as good. Does it make sense to comment on the reasons for these differences? Does this mean that models should be further improved? Or does this mainly caused by manufacturing inaccuracies so that it does not make sense to try to improve models?"

Authors' response to comment #2: This is a valid point and question, and was part of the reason the authors undertook the parametric studies – especially for pin position error. In retrospect, for the gearbox with CRBs we wish we had acquired more data points (instead of just 1) for the pitch and yaw moment cases like we did for pure torque conditions (this can be seen, for example, in Fig. 10). This would allow a better assessment of how well the models and experiments match. We did acquire more of these points for the gearbox with TRBs and we think it improves the quality of the work. The Transmission3D software is the highest-fidelity software that the authors are aware of for this type of gearbox modeling.

No changes were made to the manuscript.

Comment #3: "Cylindrical Roller Bearings with clearance and Tapered Roller Bearings with preload are compared, and it is concluded that TRB with preload result in a significantly longer lifetime. Does preload lead to a reduction in efficiency? Can I conclude from the temperature measurements that there is no increase in losses?"

Authors' response to comment #3: Also a valid concern, and it was the motivation for examining the temperature measurements as you have surmised. To better address this fact, the authors have added a phrase stating:

There is little to no difference in the temperature of the planet bearing inner races between the two gearboxes and thus most likely little to no impact on the gearbox efficiency.