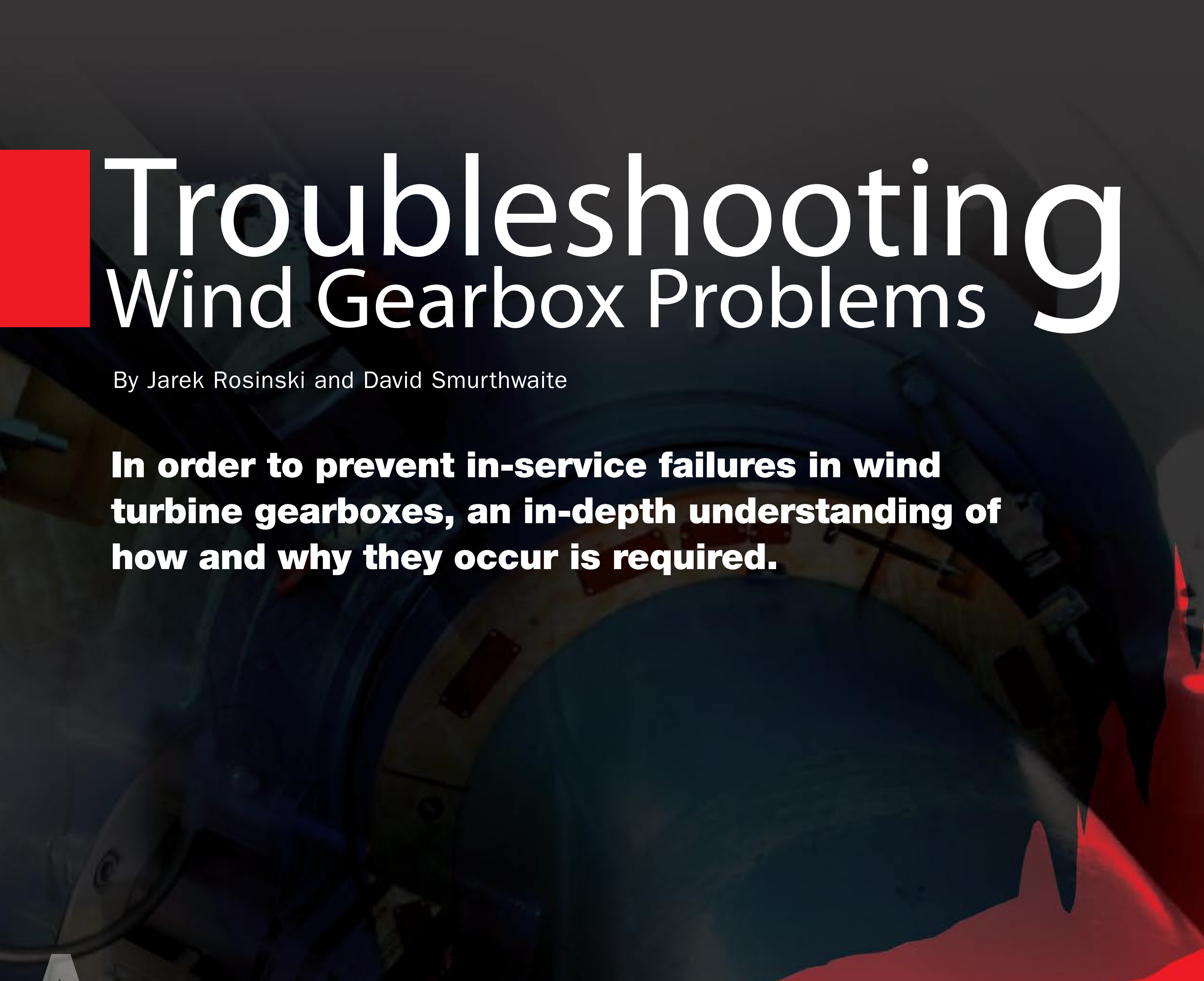
Troubleshooting Wind Gearbox Problems

By Jarek Rosinski and David Smurthwaite

In order to prevent in-service failures in wind turbine gearboxes, an in-depth understanding of how and why they occur is required.



As the growth in wind energy continues, the average size and capacity of wind turbine generators is also increasing, stretching designs and materials to new and unproven limits. As of 2009 the new generation of 4.6MW wind turbines are being deployed, and 10MW machines are prototyped.

With this increase in size comes an increase in the cost of operation, and in particular the cost of repairs, downtime, and unscheduled maintenance. Reliability during operation will become increasingly important, especially as offshore wind farms are developed. The costs of hiring specialized vessels to conduct major work and the downtime waiting for suitably calm seas during winter months are potentially critical.

The effective condition monitoring and early indication that maintenance is required before extensive damage is caused to major components is vital. This requires the development of a fundamental understanding of in-service failures and experimental and analytical techniques to prevent them.

GENERATOR COMPONENTS

Most wind turbine generators are designed to run at a fixed speed so that the AC electricity is generated at the required frequency. Typically, the components of a large wind turbine generator driveline are the rotor, hub, input shaft, gearbox, disc brake, and generator (see fig. 1).

The gearbox transfers the torque from the slowly rotating input shaft driven by the blades (at input speeds between 12 and 20 rev/min) to a high-speed output suitable to drive the generator (at 1,500 rev/min). The layout of a typical gearbox is shown in fig. 2. This example contains one epicyclic stage and two parallel axis stages. The overall ratio of the gearbox is relatively high, typically around 80:1.

It is often the case that gearboxes designed using the accepted standards and operating well within the theoretical design limits still suffer failures in a wind turbine installation. The reasons for such failures are not fully understood. Certainly the load pattern in a wind turbine generator is different to most other industrial applications.

GEAR ALIGNMENT

The correct alignment of gears to within 20-30 microns is important if load is to be distributed evenly across the face width. Uneven loading will overload a small section and lead to premature failure. Flexing of the gearbox casing and shafts under the maximum rated torque can cause changes to alignment that need to be allowed for in the gear micro geometry by providing suitable lead and profile corrections.

Gear alignment can be evaluated by the calculation of the face load factors, KHß and KFß. This can be achieved by installing strain gauges along a root and using the method prescribed in ISO 6336-1.

GEAR ALIGNMENT FOR NEW DESIGNS

The verification of gear geometry using this method for prototype gearboxes of new designs is now commonplace and will become compulsory from 2010. Novel and intricate methods have evolved to enable the extraction of data from complex epicyclic gear sets. Recent tests have been conducted on prototypes of 2MW and 4.6MW gearboxes to verify novel design features and quantify the effects of increased mass associated with additional instrumentation. For information on in-service gear alignment checking, see fig. 3.

Fig. 1: Layout of a typical wind turbine generator (courtesy of Siemens).

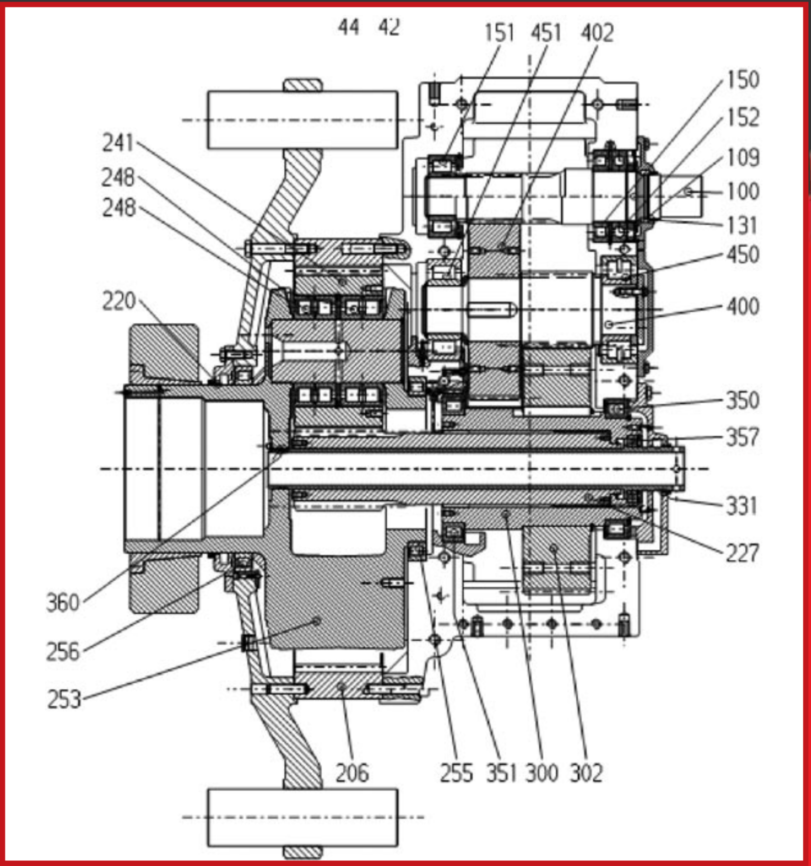


Fig. 2: Cross section of a typical wind turbine generator gearbox.

INPUT TORQUE MONITORING

Long-term, direct measurements of input torque are vitally important to better understand in-service loading. From most practical investigations it becomes apparent that changes in gearbox vibrations due to varying torque can be significantly higher than the changes due to impeding gear or bearing failures. For this reason it is essential to monitor torque for any condition monitoring system to be effective in a wind turbine generator.

A system capable of such measurements has been successfully developed and deployed in service. An example of such application on a 1.3MW input shaft is shown in fig. 4. The system used a strain gauges bonded on the input shaft with signal conditioning, amplification, and FM electronics mounted within an annular housing clamped around the shaft. The signal was wirelessly transferred from the rotating shaft, demodulated and acquired over a two-month period. In this instance batteries were used as a power source, but for longer term monitoring an inductive power supply or a self-inducing power supply is typically used.

The results obtained from this work yielded important information about the loading characteristics at the test site during a two-month period. More

Fig. 3: Transmission Dynamics' state of the art Gear Alignment Module. Size: 60x60x10mm, including Bluetooth communication and integrated long-life battery supply.

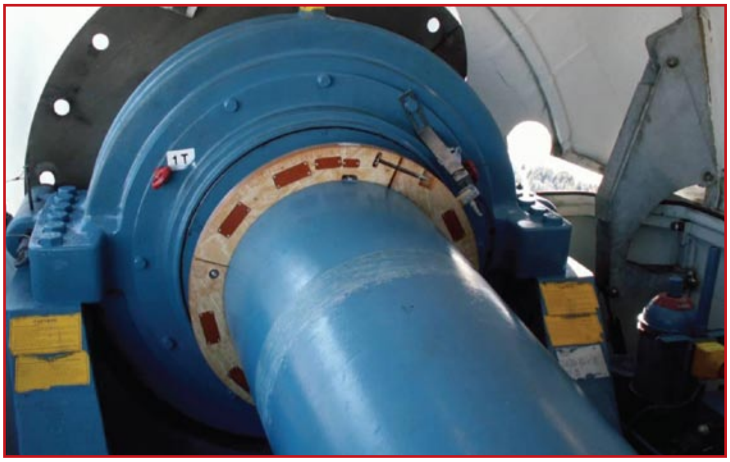


Fig. 4: Input shaft equipped with Transmission Dynamics' torque monitoring equipment.

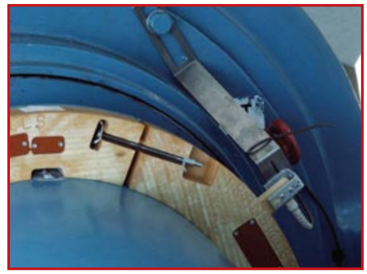


Fig. 5: The strain gauge is visible in the recess on the left of the picture, with the FM telemetry pick-up at the lower right.

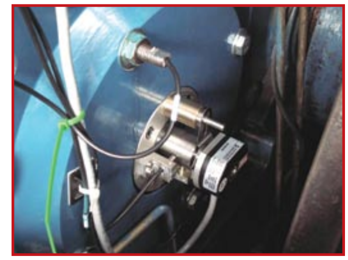


Fig. 6: Instrumentation installed to monitor intermediate shaft radial displacement and bearing element behavior.

importantly, the genuine in-service data revealed regular overload events during the automated stopping and re-starting of the generator. The type involved in this exercise stopped on average three times per day, equating to over 1,000 events per year, as the wind conditions varied.

SHAFT DISPLACEMENT

Having suffered numerous bearing and gear problems on the intermediate shaft of the parallel axis gear sets, inductive probes were installed to monitor radial movement of the shaft. A set of additional inductive probes were installed in axial direction to measure dynamic displacement of rolling elements. The setup is shown in fig. 6.

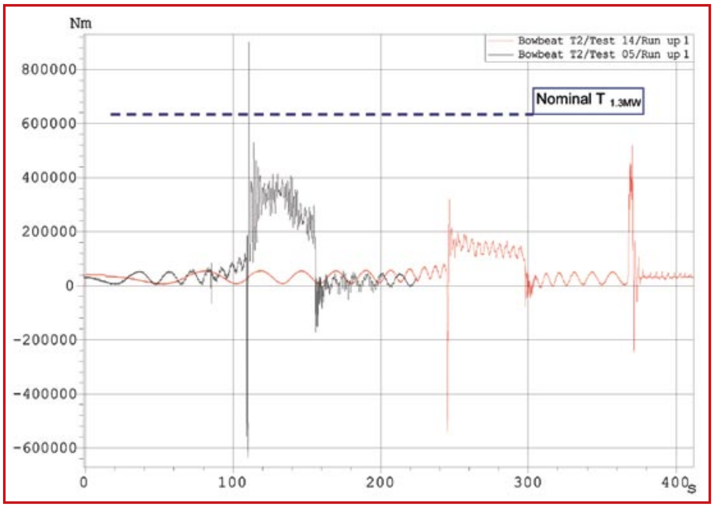


Fig. 7: **Input shaft torque for WTG operation in high- and low-power modes.**

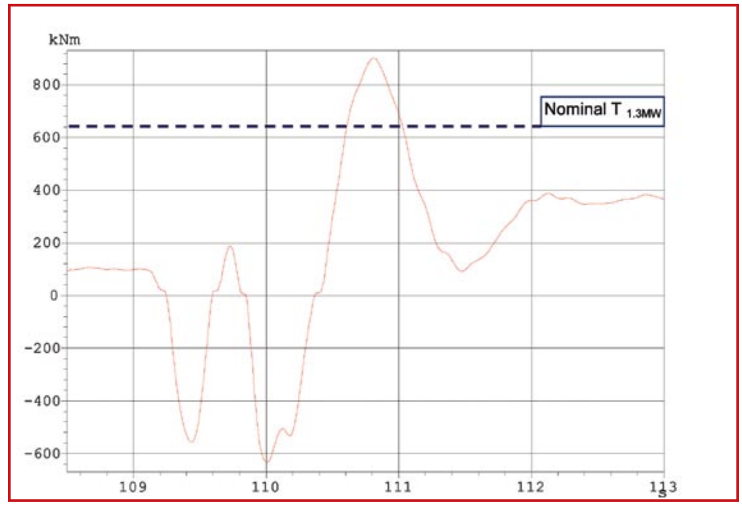


Fig. 8: **Detail of transient torque events during engagement in high-power mode.**

The replacement bearing cover plate was manufactured to accommodate brackets holding two sensors to measure radial displacement of the shaft in two orthogonal directions. The stub shaft was installed on the end face of the intermediate shaft to act as the target and carry a rotary encoder delivering 256 pulses per rev. The encoder enabled accurate measurement of shaft speed using high-speed 50MHz acquisition and analysis of torsional oscillations and shaft speed transient events. A third inductive probe was installed in an axial orientation (visible in the upper part of the bearing cover plate in fig. 6), targeting the end face of the cylindrical rolling elements of the bearing supporting the intermediate shaft.

The interpretation of data from the instrumentation described enabled a clear picture of system behavior to be established, and full understanding of the failure mode in this specific type of turbine, which is described in the following sections.

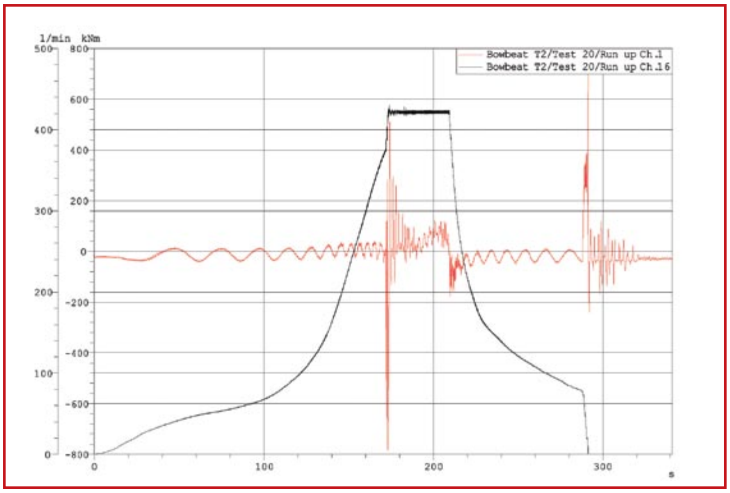


Fig. 9: **Input shaft torque and intermediate shaft rotational speed during test 20.**

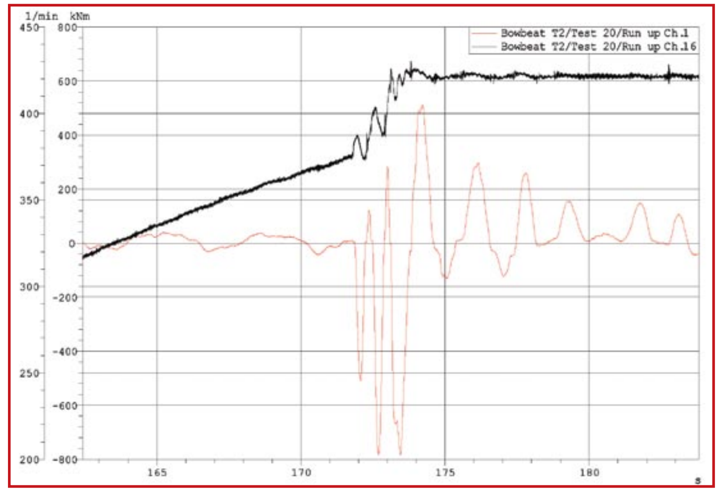


Fig. 10: **Detail of intermediate shaft speed (black) and input shaft torque (red) as the generator is engaged.**

IN-SERVICE TORQUE AND SHAFT DISPLACEMENT/ANALYSIS

Input torque is compared for two tests; test 05 (~12 m/s wind) and test 14 (~5m/s wind), as shown in fig. 7. In both cases the data was acquired from rest, with the wind turbine ramping up to synchronous speed, engaged for 20-30s, disengaging and slowing down. Points to note are:

- The much longer time taken to reach operating speed (of only 1000 rev/min) for test 14 compared to test 05, where the operating speed was 1500 rev/min.
- Very large transient events at the time of contactor engagement, resulting in torque levels in excess of that for maximum rated power, in both positive and negative directions.
- The negative torque peak at the point of contactor switching for the low

wind speed was almost as strong as that for the higher wind speed, when operating in a high-power mode.

- The transient torque peak when the brake stops the system (around 370s) is the largest positive torque event for test 14 in the low-power mode. (Note that the actual stopping event was not recorded for test 05).
- The ripple in the torque signal is due to shear load and corresponds to once per input shaft revolution. The shear load may be due to hub and rotor mass, or misalignment during assembly. This characteristic is seen throughout the test for the red trace (at low wind speed, in low-power mode), and also for the black trace in test 05 during the run-up and slow-down. Torque fluctuation during the period of test 05 when the generator is connected in high-power mode, however, is dominated by three times input shaft speed, i.e. blade passing frequency.

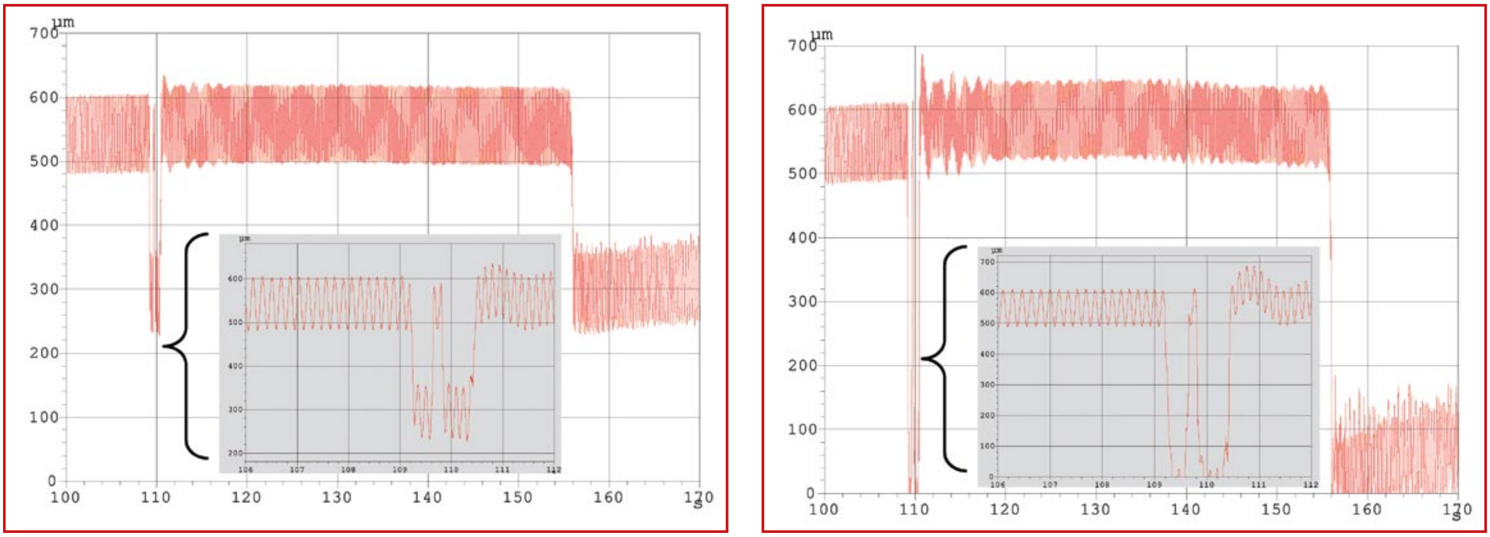


Fig. 11: **Test 05—intermediate shaft movement (lateral direction-left graph, vertical direction-right) with detail of the engagement events inset.**

Analysis of the transient torque events in more detail is shown in fig. 8, showing two negative peaks followed by one even larger positive peak. Data from the rotational encoder indicating the intermediate shaft speed was analyzed using a dedicated tacho channel counting at a 50MHz rate. This enabled small fluctuations in rotational speed to be accurately measured.

Figure 9 shows the input shaft torque (red) and intermediate shaft speed (black) during a start up test in 8-10m/s wind. The system accelerates towards synchronous speed, the generator engages at 173s and runs until 210s then the rotor slows with air brakes (blade tips) deployed until the disc brake is applied at 288s and brings the system to rest. Torque fluctuations, including reversals, continue for around 30s after the system has stopped and is being held by the disc brake applied on the high speed shaft.

INTERMEDIATE SHAFT SPEED

Closer examination of these events shows that, due to premature engagement of the generator, the system is forced to rapidly accelerate from 375 rev/min to 422 rev/min in a 2 second period. It can be seen that during that time the generator forces the gearbox-rotor system to accelerate at a rate of 126 (rev/min)/sec, which is approximately 18 times faster than the prevailing average acceleration (see fig. 10).

As a result the system winds up due to torsional compliance and the released potential energy results in rebounding of the rotor system, which is seen as three significant torque overloads in the negative direction. During that time the application of torque in the reverse direction, combined with inadequate helix correction of the mesh at the non-driven flank may result in a significant increase of load intensity across all mesh stages. The combination of excessive load intensity at the localized face width and the application of torque in the reverse direction may explain the tooth fracture witnessed previously in this type of gearbox. It is possible that the above events combined with systematic errors in helix correction applied on the non-driven flank for all such gearboxes may result in a large number of similarly catastrophic tooth failures in the future. Due to the potential implications of these findings it was strongly recommended that the micro-geometry of the mesh at each stage of this gearbox at the non-driven flanks, including the epicyclic stage, should be carefully verified.

The influence of high dynamic overload transients, occurring in reverse direction on bearing elements, needs special consideration. Most of the rolling element bearings are designed to operate with a relatively smooth transition between the loaded and unloaded zones. From the acquired information a sudden torque reversals recorded during start up operation resulted in almost instantaneous change in shaft loading direction. It is noted (fig. 10) that torque change from negative 800KNm to positive 430 kNm, occurs in less than 100ms. Such rapid changes in shaft load direction, combined with high load amplitude, acting on misaligned rollers, will have significant impact of roller element surface durability. In the subsequent sections it is shown that unloaded and significantly misaligned rollers in the unloaded zone may be subjected to sudden load application which will result excessive contact pressure between the roller and the inner and outer bearing races.

SHAFT DISPLACEMENT: PART II

Data from the two channels measuring intermediate shaft radial displacement is shown in fig. 11. Generator load was switched on at 110s and off at 156s. There was sudden shaft displacement when the generator was engaged. The lateral movement (200 µm away from the high speed shaft) and vertical movement (550 µm downwards) gave a total displacement greater than 600µm. The maximum radial clearance specified by INA for bearing type LSL 192332 in this class is only 120 µm (to DIN 620-4 class CN).

The number of transient torque reversals occurring during each engagement cycle varied from test to test. In fig. 11 (test 05) two transient events can be seen as the generator engaged in the high-power mode with a wind speed of around 12ms-1. For test 14 only one single event occurred as the low-power mode was engaged in low wind conditions (see fig. 12). In test 20, however, six events are recorded as the generator engaged with the wind being only just strong enough to operate in the G1 mode (see fig. 13).

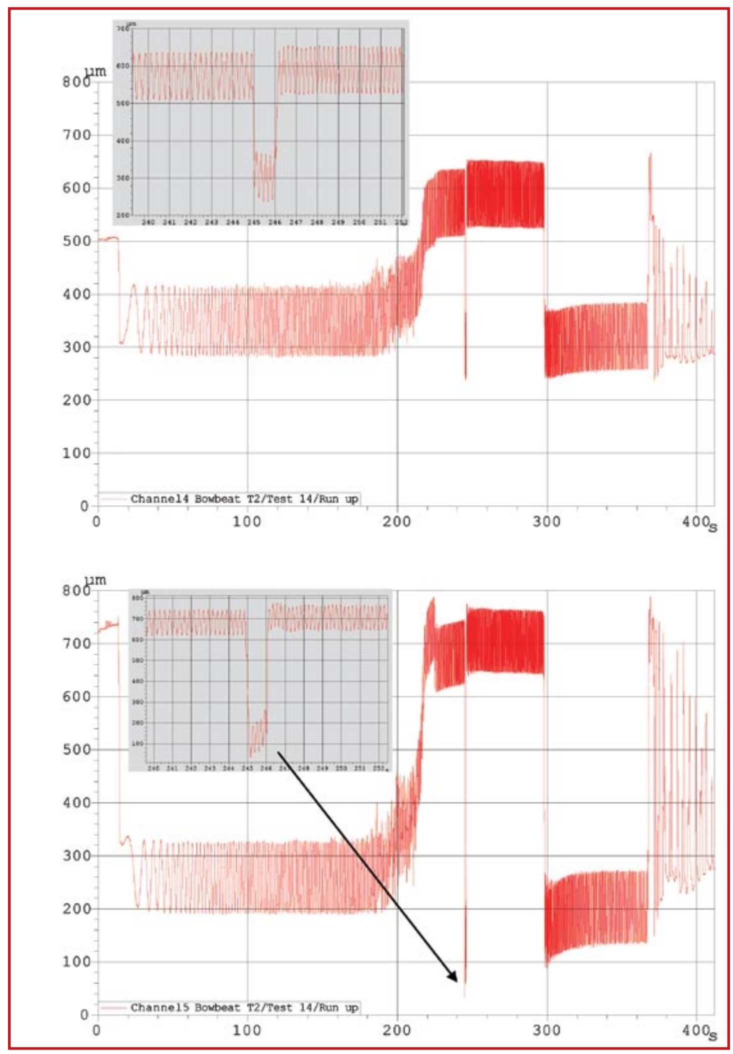


Fig. 12: Test 14—intermediate shaft movement; low-power mode (lateral direction-upper graph, vertical direction-lower).

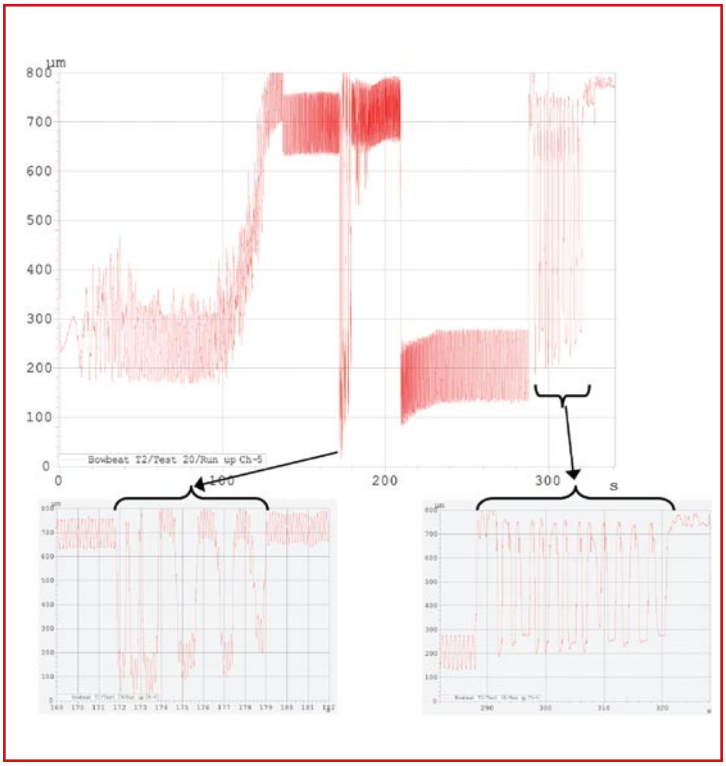


Fig. 13: **Test 20—intermediate shaft vertical displacement (G1 high power mode); inlaid graphs show details of six events upon generator engagement and multiple oscillations after braking to standstill lasting around 30 seconds.**

It should be noted that after the rotor was brought to rest by the disc brake, many transient events can be seen as the system continued to oscillate with the disc brake locking the high-speed shaft. These can be seen after 370s in fig. 12 and after 290s in fig. 13 for tests 14 and 20, respectively. Detail has been shown in an inlaid graph for test 20 only.

BEHAVIOR OF ROLLING ELEMENTS

Intermediate shaft bearing element axial behavior was investigated using an eddy current probe targeted centrally at the end face of the elements. Typical output from this sensor is shown in fig. 14, where five elements can be seen to pass the sensor. The gaps between the elements are indicated by discontinuity in signal level.

Four startup tests were conducted with the axial sensor in different locations around the bearing element pitch circle; nominally 11:00, 2:00, 5:00, and 8:00 o'clock when viewed in the cover plate. The results shown in fig. 14 indicated the greatest skew of the elements (250 µm) to have been measured in the 2:00 o'clock position.

Considering the direction of rotation of shafts within this gearbox, the loaded zone of the IMS bearing will be the upper portion of the race, so the large skewness shown in the upper right graph of fig. 14 was thought to be most significant. The sensor was replaced in that location (2:00 o'clock), to be left for the duration of the campaign. Bearing element axial behavior in the one second when the generator was engaged is shown in fig. 16. Elements moved axially back and forth by around 300µm in 0.5 s, and the direction of skew is reversed.

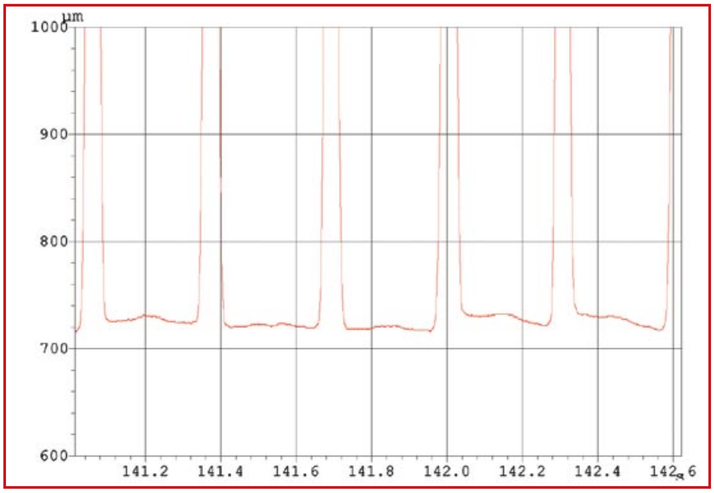


Fig. 14: Typical data from the axial eddy current probe, showing five bearing elements. Gaps between the elements are indicated by high displacement values.

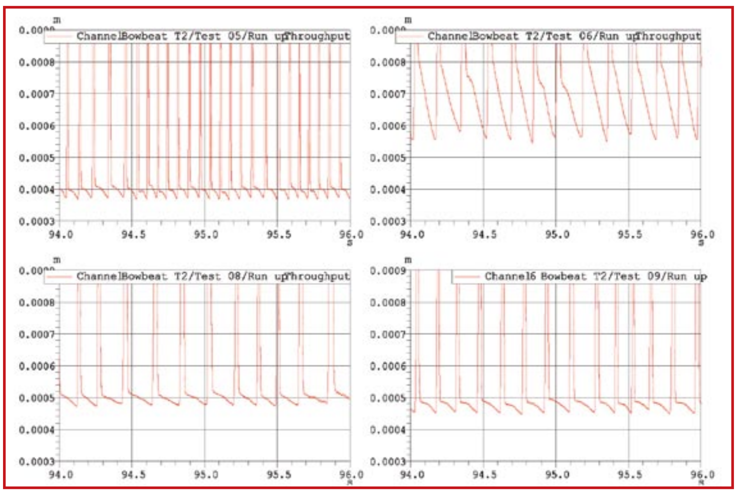


Fig. 15: Bearing element end face axial displacement (Z). Clockwise from top left, the sensor was positioned at 11:00, 2:00, 5:00, and 8:00 o'clock in the cover plate. Up to 250µm skew was measured with the sensor in the 2:00 o'clock position (top right graph, test 06).

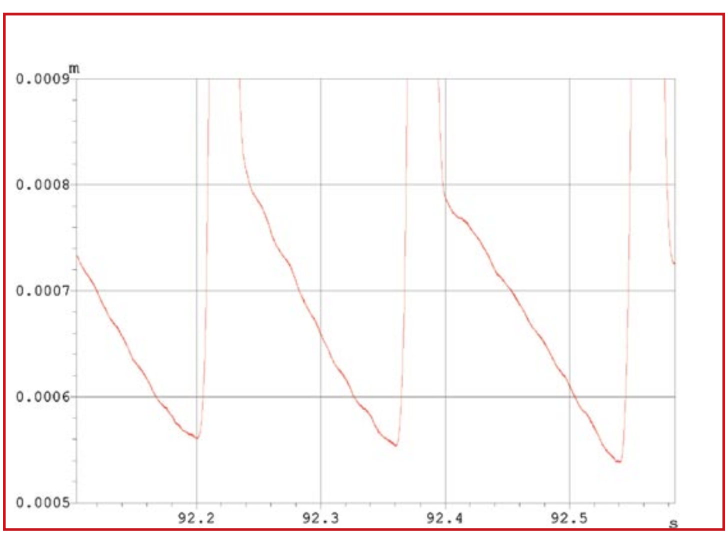


Fig. 16: Detail of bearing axial displacement from test 6, showing over 200µm skew.

Fig. 17: Rolling element with embedded face load intensity distribution measurement system. High-speed telemetry system operates at 80 ksps, 12-bit SHH and is powered from an integrated miniature battery.

An important deduction from the combination of this data is that significant damage arises due to the repeated impacts of the heavily overloaded rolling elements in the unloaded zone, when they are instantaneously loaded while in heavily misaligned conditions. In such conditions, they are not loaded along an edge as they would be when running parallel in the loaded zone, but instead at one or two points. It is suggested that the increased bearing clearance caused by surface durability failure is sufficient to result in an unacceptable misalignment across the gear face width, which results in gear root fatigue failure due to increased misalignment.

A new test tool is now being deployed which allows the measurement of roller face load intensity distribution during normal operation of industrial bearings. This new instrumentation involves a high-speed digital telemetry system (80 ksps SSH data transfer at 12-bit resolution) built inside the rolling element (see fig. 17). Modern power management techniques allow operation of the smart roller for up to three months on one coin battery.

CONCLUSION

Troubleshooting gearbox problems in wind turbine applications requires the development of comprehensive experimental and analytical tools and specialized instrumentation for long-term in service measurements. From the routine measuring campaign carried out during the end of warranty inspection of an average size wind turbine farm it was found that bearing and gear element failures were attributed to the fundamental shortcomings in WT control strategy. Premature engagement of the wind turbine to the grid resulted in strong overloads and rapid torque oscillations in the negative direction.

The high amplitude of rapid torque reversals present during every start-up operation results in not insignificant amounts of overloads on highly misaligned rolling elements. With typically more than 3,000 start cycles per year and on average five torque oscillations during each start (15,000 overload cycles per year) high contact surface loading results in wear and increased bearing clearance. This in turn leads to an unacceptable level of mesh misalignment resulting in tooth root fatigue failures.

The growing number of wind turbine failures occurring in various locations around the globe requires close collaboration of gearbox designers, electrical control, and software and system integrators.

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