Wind turbine main bearing field recordings and performance validation

Alessandro Sansone

Summary

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Postbus 2350
3430 DT NIEUWEGEIN
The Netherlands

On line, in situ recordings have been performed on a multi-megawatt wind turbine. The retaining ring (yellow in fig.1) of the main bearing has been instrumented.

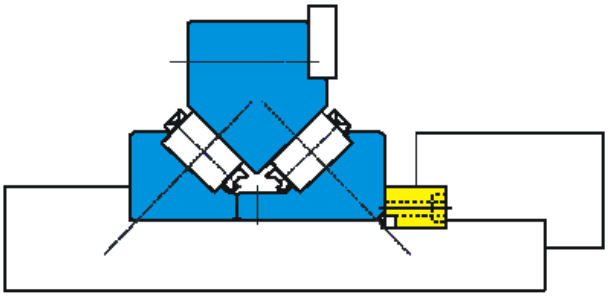

Figure 1. The double row taper roller bearing, approx 2m bore.

Radial and axial displacements of the inner ring, relative to the outer ring were measured at three circumferential positions via eddy current sensors, see figure 2.

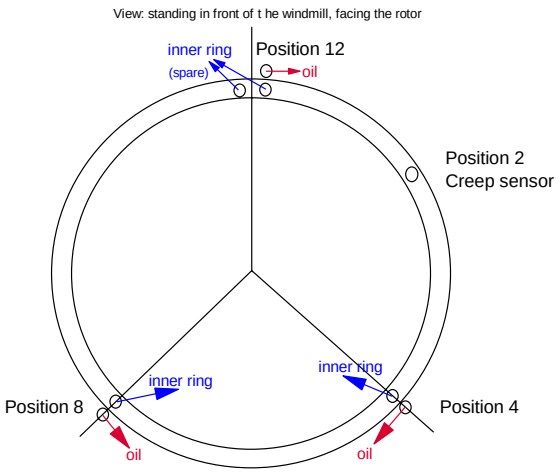

Figure 2. The solution: the retaining ring instrumentation

Temperature measurements were performed on various stationary and rotating components.
For over one month, data have been (remotely) collected from the wind turbine in service. During this period a variety of running conditions were recorded, e.g. start up, normal service, controlled- and emergency stops.
Apart from this, the inner ring creep and bearing temperatures were recorded for three additional months.
The data, consisting of 19 channels, included parameters like temperatures, displacements, rotor position, rotor speed, wind speed (two sensors, left and right), generator power, bearing inner ring creep and, experimentally, bearing acoustic emission.
All collected data were analysed at the SKF Engineering and Research Centre (ERC).
The data were post processed with software created in house in Labview; for each Δt the values given by all the sensors were classified in four levels of magnitude and then combined to calculate "displacement cases", together with the speed signal. These "displacement cases" were used as input for SKF's simulation tool "Orpheus". The applied bearing axial preload and the measured outer ring displacement boundary conditions were entered into the model to calculate the bearing load and other key parameters, including the estimation of the bearing life.

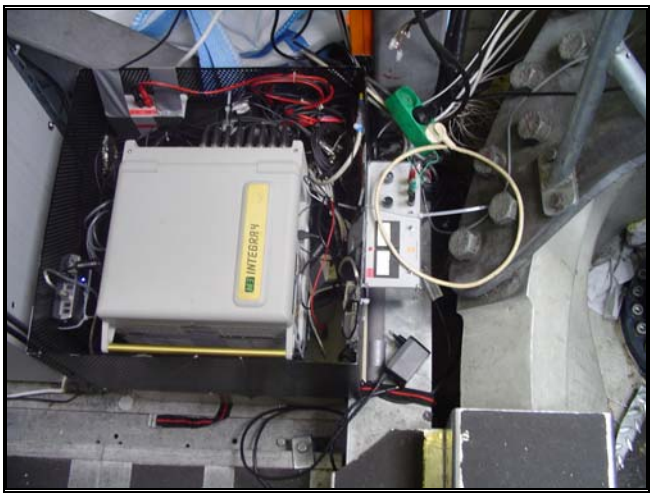

Figure 3. The Integra DAS and its Faraday cage

1. Temperatures

The seven thermocouples have been installed at the factory site. These thermocouples were positioned according to figure 2.

The inner ring temperature thermocouples have been mounted in the retaining ring, against the side of the inner ring. The thermocouple holes have been closed with silicone kit to avoid oil flow around the thermocouples. At position 12 (top of the stationary inner ring) a spare thermocouple is mounted.
The oil temperature is also monitored with three thermocouples, place in the same positions as the

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inner ring thermocouples, but on the outside of the retaining ring, direct in the oil flow around the bearing.
The thermocouples leave the gearbox at the same places (holes) as the displacements sensor wires. The wires have been EMC shielded against external electrical/magnetic influences.
The outer ring temperature was going to be monitored with an Infra Red temperature sensor. This sensor has been calibrated, using the same paint, as used on the outer ring of the bearing, as target. In laboratory situation, this works perfect.

The MadgeTech data logger was installed after the gearbox was mounted in the windmill. During the installation of the other equipment and the trial runs of the windmill, two days of data were collected with 1 Hz sample frequency.
Some statements:

- The air heaters in the windmill (pointing at the gearbox) have little influence on the gearbox main bearing).
- The oil responds to the oil heaters much faster than the bearing inner ring does, which is to be expected.

2. Creep

The creep of the IR has been monitored using a sensor-bearing unit (a 608) mounted in a proper way in the pocket created at 2 o'clock in the retaining ring; for instance, the IR will creep rotating along the Z-axis (the Cartesian axis of rotation); the unit has been mounted with its z-axis perpendicular to the Z-axis of the bearing and putting them in tangent contact (using rubber material for an adequate friction). The sensor was placed in a position of the retaining ring where the circumference of the IR is 6110.44 mm; the circumference of the sensor's wheel is 78.48 mm. This device is capable to give as output a pulse signal (8 pulses per revolution in this case), which was used to evaluate the amount of IR creep. As consequence, each pulse means 9.81 mm displacement.

Figure 4. The creep sensor

3. Displacements

Displacement was measured as this was considered to be the only viable measurement to take with established technology available in the time frame available. By measuring close to the bearing the displacements measured would be dominated by the bearing contact and roller stiffness; hence it was hoped that the behaviour of the bearing could be determined from these data. Eddy current sensors were chosen for their capability to work in a very harsh environment.
The eddy current sensors system consists of:

- Sensor (eddy current type) in 'disc' form or 'pen' form, which is a dimension choice.
- Cable (armoured and EMC screened) permanent fixed to the sensor. The cable length is calibration dependent (not allowed to be changed in length).
- Eddy current driver (electronics to condition and amplify the sensor signal)
- Mounting and protection box, with connectors, power supply and a driver mounting 'rail' to connect the eddy current drivers on.
- Output cables to be connected to the Data Acquisition System (DAS)

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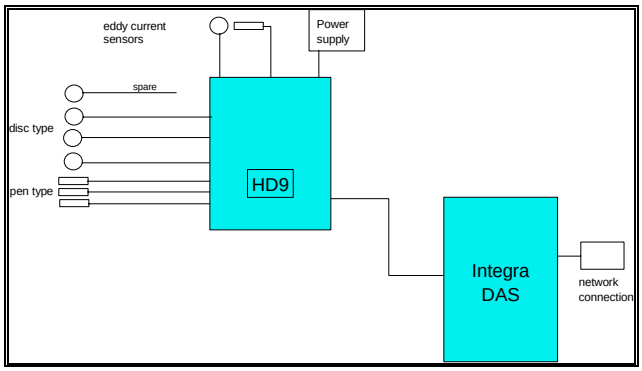

Figure 5. Measuring equipment layout of the eddy current measuring system

The sensors are independently calibrated, all with the standard SKF calibration tool (SKF ID 5128 displacement dial caliper). This caliper has a reference accuracy of 0.01 mm).
The readout is done with a Fluke DVM, calibrated with an accuracy of 0.01 volt (displaying up to 0.001 volt).

The sensors:
1x Disc probe, model PRD12, range 12 mm, 7 meter armoured cable, 1.4 mV/µm. This sensor is used for the radial displacement measurement between the nacelle and the rotating outer ring of the main bearing. This sensor is mounted on a bracket, fixed on the nacelle.

2x (1 spare). Disc probe, model PRD08, range 8 mm, 9 meter armoured cable, 2.0 mV/µm. This sensor is used for the axial displacement measurement between the nacelle and the rotating outer ring of the main bearing. This sensor is mounted on a bracket, fixed on the nacelle.

4x (1 spare). Disc probe, model PRD04, range 4 mm, 7 meter armoured cable, 4 mV/µm. These sensors are used for the radial displacement measurements between the inner ring and the rotating outer ring of the main bearing. At the top of the bearing (position 12) a spare sensor was placed, just in case)

5x (2 spare). Straight probe, model PRS 04, range 4 mm, 7 meter armoured cable, thread M8, 4 mV/µm. Three of these sensors were placed (fixed) on the housing of the gearbox to measure the axial displacement of the outer ring, relative to the inner ring.

Eight (8) matching eddy current drivers for the sensors were used. The drivers were pre-mounted in the metal driver housing cabinet (called HD9). This cabinet included the 24 VDC power supply.

The calibration values are used by the Integra (DAS) as the measurement references.

The accuracy of these sensors is up to 2 µm, 10 µm for those mounted in the bracket.

The cables were bundled and therefore the cables have been checked for interference between the sensors themselves (some of them operate on the same excitation frequency). This was done in the factory site. The influence (like increase in noise or signal deviation) between the sensors was not noticeable.
The standardised BNC connectors were used for the outputs of the drivers (the voltage representing the displacement) with BNC coax cables and connected to the DAS.

3.1 The measurements
During mounting, the distance sensors/targets has been constantly monitored in order to evaluate correctly the displacements. Following table summarises these distances in different situations.

*	Original sensor distance		
	Dismounted	Mounted in Gearbox	Mounted in Windmill
Ax 12	2	1.83	1.907
Ax 4	2	2.009	1.917
Ax 8	2	1.969	1.847
Rad 12	2.819	2.667	2.404
Rad 4	1.354	1.496	1.688
Rad 8	1.414	1.76	1.823

Table 1.

3.1.1 The reference
Tests have been performed at the test site to measure the (non-loaded) main bearing displacement during rotation. This was done on a test stand. This resulted in the "reference" of 'ovality' of the bearing (sampling frequency 200

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Hz). Same test has been performed also once the gearbox was mounted in the windmill. These values have been subtracted from the displacements. This measurements match perfectly with those performed in laboratory.

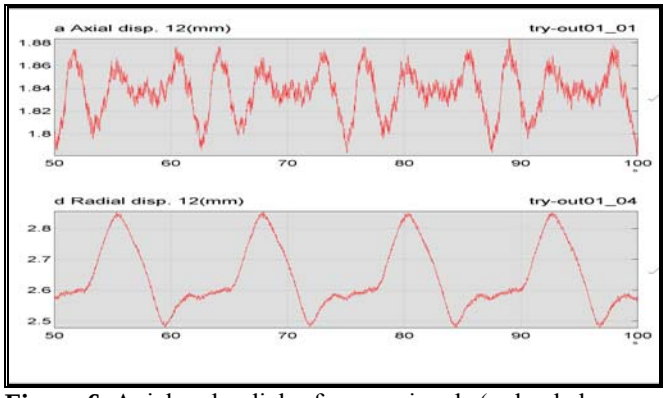

Figure 6. Axial and radial reference signals (unloaded condition).

3.1.2 One-month data

The result of the measurement campaign is here shown; the signals, three axial and three radial are plotted together with the rotor speed signal. Sampling frequency is 10 Hz. Even if these measurements were remotely controlled from ERC, there was no necessity during the campaign to vary this sampling rate.

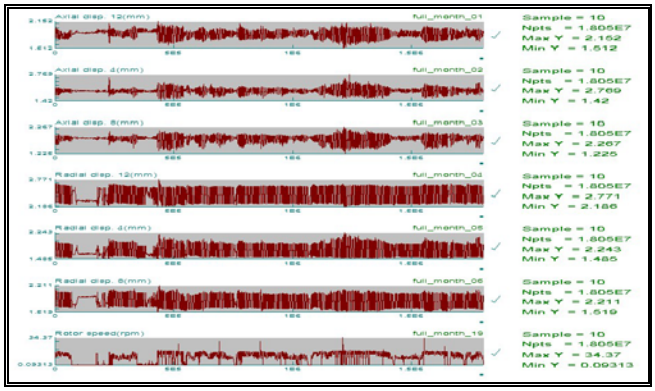

Figure 7. One-month displacement recording signal

3.1.3 Particular events

Measurements have been taken during controlled extreme working conditions such as **start-up**, **braking-down** and **emergency brake**. Reason is that these conditions were supposed to be the most extreme for the bearing. Picture shows the axial behaviour of the bearing, which is the most solicited, during an emergency brake manually obtained when:

- wind speed = ~12 m/s
- power produced = ~2.2 MW
- rotor speed = ~16 rpm

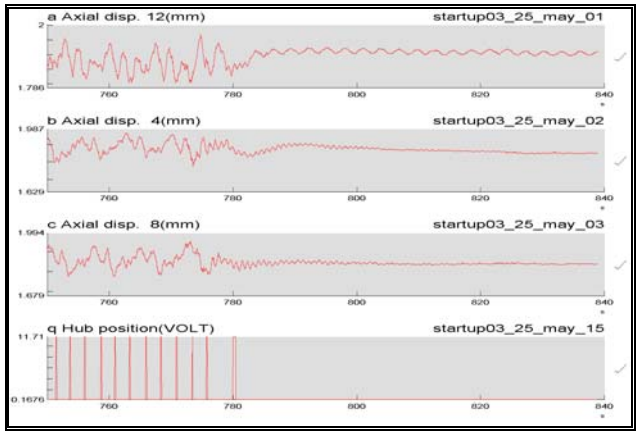

Figure 8. Emergency brake signal response

3.1.4 The blade passing frequency

The blade passing frequency has been recorded in order to evaluate the influence on the bearing behaviour. In the following picture, the purple represents the moment when a blade is perpendicular to the tower. It is possible to find this influence in all the signals registered.

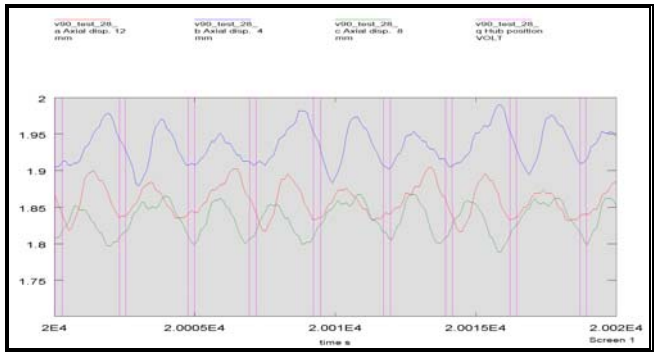

Figure 9. The blade passing effect

3.1.5 Some observations

- The particular events, of which the emergency brake is the most solicitating, do not represent the most extreme conditions for the bearing (mostly this is linked to wind conditions)
- The influence of the blade passing frequency is not relevant for the normal functioning of the bearing

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4. Displacement cases

A deeper analysis of the displacement data has been performed. The purpose is to better characterise the bearing performance and to create specific input for the SKF simulation tool Orpheus.
In total, per channel, at a sampling frequency of 10 Hz, 18050000 points have been recorded. After analysis of the possibilities and after removing of the values at zero speed, a data reduction at 2 Hz has been done, resulting in a total of 2375599 points per channel. Each of these values has been classified per level of magnitude, after defining four classes of amplitude per each channel, including the speed.

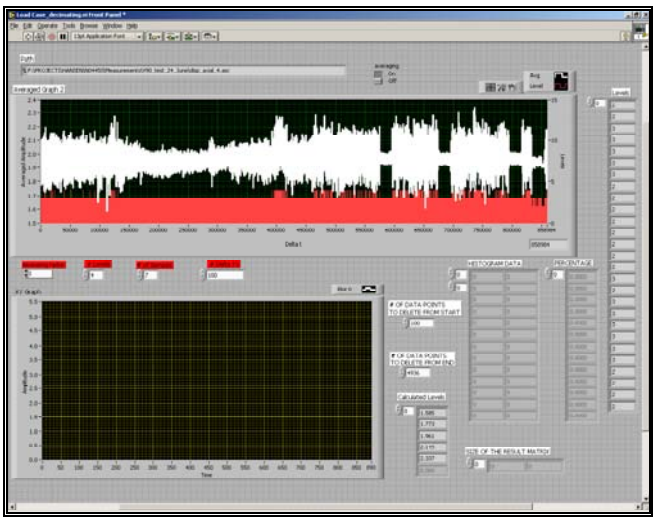

Figure 10. Example of the reduction in magnitude levels

The levels have been defined after removing the "ovality" previously measured and taking in account the sensor position.

	Levels related to sensor position				Sensor position*
	1	2	3	4	
Ax 12	-0.299	-0.152	-0.003	0.145	1.907
Ax 4	-0.445	-0.068	0.309	0.687	1.917
Ax 8	-0.457	-0.213	0.032	0.277	1.847
Rad 12	-0.016	0.047	0.111	0.175	2.404
Rad 4	-0.005	0.105	0.215	0.325	1.688
Rad 8	-0.116	-0.019	0.079	0.177	1.823
Speed	2.595	7.601	12.607	17.613	

Table 2.

The table above gives an idea of the amount of displacement per sensor. The purple column indicates at which distance is the sensor from its target, according to table 1. Anyhow, to characterise the performances this make sense if the percentage of staying in these levels are taken in account. Next table better explains this concept.

	Percentage of level per sensor			
	1	2	3	4
Ax 12	0.06%	45.02%	54.88%	0.04%
Ax 4	0.09%	95.96%	3.94%	0.00%
Ax 8	0.04%	8.71%	91.14%	0.10%
Rad 12	31.86%	42.69%	23.73%	1.72%
Rad 4	51.15%	31.25%	17.57%	0.03%
Rad 8	18.85%	53.73%	26.30%	1.12%
Speed	0.20%	24.23%	44.41%	31.16%

Table 3.

Following this procedure, per every Δt's the combinations of the seven levels (six displacements plus the speed) have been counted and their percentage calculated. 2375400 combinations on 2375599 possibilities repeat themselves, representing the 99.99% of the total. A threshold of 10000 events has been defined resulting in 1896694 combinations, the 79.8% of all the possibilities. To cover the remaining 20.2% of the possibilities, after a convergence study, some of the combinations have been counted in the worse of the cases; for instance, if there is the same combination of the displacement levels at two different speeds, it will be taken in account as if it was happening at the highest speed.
Finally, 18 displacement cases have been defined.

5. Orpheus

Orpheus is an in-home-developed software to simulate the behaviour of rotating machinery with an emphasis on system integration. It is designed to handle in efficient and accurate manner moving contacts over flexible bodies (rolling bearings, gears, etc.)
Orpheus uses FEM models in combination with a special developed reduction method (an extended Component Mode Synthesis) to minimize the model size to almost the absolute minimum hence reducing the calculation time tremendously. It offers the possibility to perform static and dynamic analyses of complete machines (gearboxes, electro motors, windmill, pumps, suspensions, etc.)

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5.1 Orpheus analysis

An Orpheus analysis was done using as input the defined 18 displacements cases. These cases, representing OR-IR relative displacements, were applied to the simulation model to calculate bearing forces and moments, load distributions, max contact pressure, friction torque and estimated bearing life.

In benefit of the reader, the most important items of this analysis are here summarised together with the most interesting results.

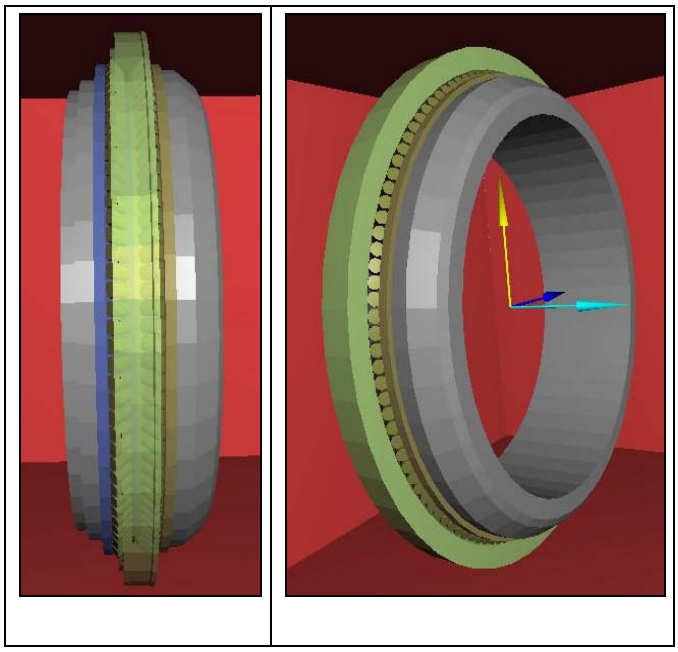

Figure 11. The Orpheus model

5.1 Model description

The double row taper roller bearing modeled in Orpheus features the following components:
• A flexible model of the IR inboard (in color blue)
• A flexible model of the IR outboard (in color gold)
• A flexible model of the OR (in color green, transparent)
• A flexible model of the stator (in color metal). This is the stationary gear for the first planetary stage. This component includes also the retainer ring.

5.2 Boundary conditions

The stator is constrained in all the DOF's at the bolt location where it is connected to the nacelle, which is considered infinite stiff (grounded).
IR – OR relative displacements applied on the OR in the three measuring positions (at 120°) in radial and axial direction.
Being an OR rotating application, speed is applied on the outer ring.

5.3 Bearing operating conditions

• The IR's are mounted on the stator with a radial interference fit
• IR inboard retaining ring is mounted so that is preloaded.
• Fixed operating temperature.
• Lubricant with same characteristics than the one in the application
The load cases were solved in quasi-static analysis (effect of the rotation taken in to account).

5.4 Observations

- Bearing load distribution (IR- roller contact forces) was successfully calculated; as example the load case 11 is here shown.

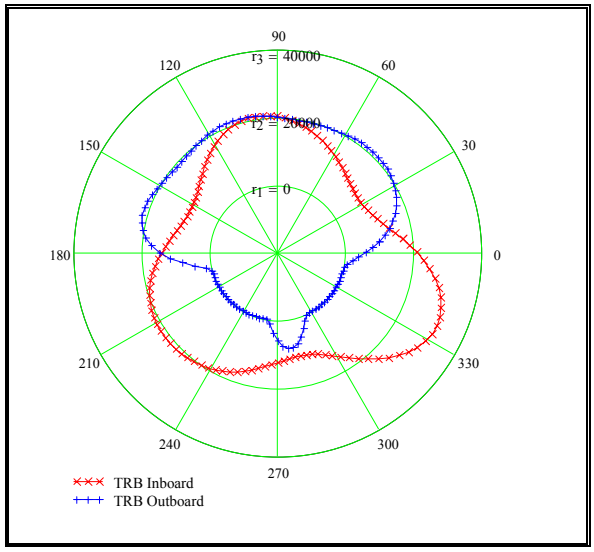

Figure 12. An example of the bearing load distribution

- Forces resulted in:

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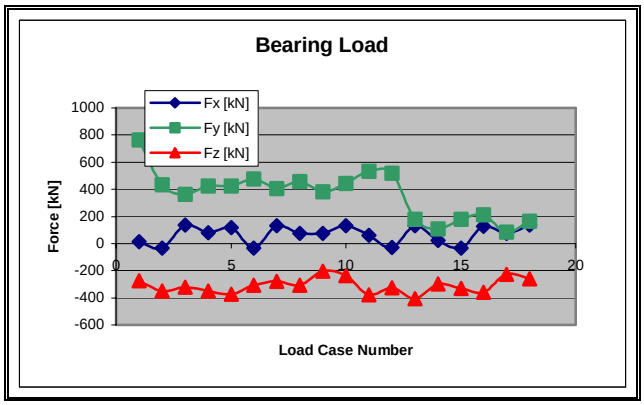
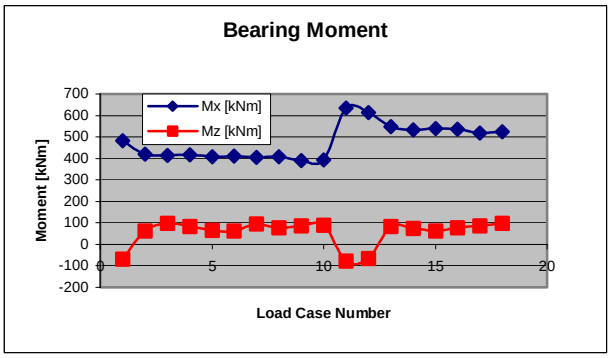


Figure 13. Forces calculated by Orpheus

- Bearing moments have been also estimated



- Bearing friction torque has been successfully calculated for all the 18 cases.

- Life of the bearing was estimated:
  - Fatigue life reference: 4.22E+05 h
  - Fatigue life FC: 8.20E+05 h

6. Conclusions

In conclusion, the turbine was undergoing an intensive test period for several months.
The work clearly demonstrates that the double row taper roller bearing operates according to requirements, in terms of clearance, loads, friction torque, working temperature, contact pressures and estimated service life.

Moreover:
• The project resulted in a powerful approach combining testing with theory

7. Recommendations

There is the possibility for such applications to calculate the load on the bearing per each position of the blades and per blade; this could be extremely useful to know, especially for wind turbine manufactures and operators.

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