Reliability Safeguarding for an 8 MW Wind Energy Gearbox in Serial Production: Prototype and Process Development Ensuring Stable Quality at the Highest Level

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Market Challenges and Motivation

The offshore market segment is forecasted to grow with a 16% cumulative annual growth in 2015 to 2020. The recent tender process leads to low power prices for offshore wind, e.g. — Dutch Borssele III and IV with 54.5 EUR/MWh. Therefore, offshore has the potential of reducing the levelized cost of energy, particularly when leveraging economies of scale and industrial maturity.

At the same time, a gearbox exchange on an offshore turbine causes repair costs of at least a million Eur.; thus, the highest product reliability is required to safeguard business certainty.

Project and Design Characteristics

In 2010, ZF (formerly Bosch Rexroth) started the development of the 8 MW gearbox. In 2013 ZF delivered the first prototype. The gearbox design was validated within a huge validation program on the customer's system test rig, as well as on the prototype turbine.

After passing more than 10 gate reviews and design reviews according to the customer's development process, and having implemented some design improvements, the 0-series production began in 2015. The qualification of the manufacturing process was ensured by applying four audits per year, together with the turbine manufacturer and end-customers. Also a certification body proved and released the processes.

The manufacturing processes' quality has continuously improved. Meanwhile, the 100th gearbox has been supplied successfully in May 2017.

ZF uses the differential gearbox for this application — a proven design applied in more than 2,500 gearboxes in the power range from 2.5 to 3 MW since 2003.

The 8 MW design consists of three planetary stages with a total ratio of about 38 (Fig. 1). The gearbox weight is about 71.5 metric tons, with a diameter and length of about 3,000 mm.

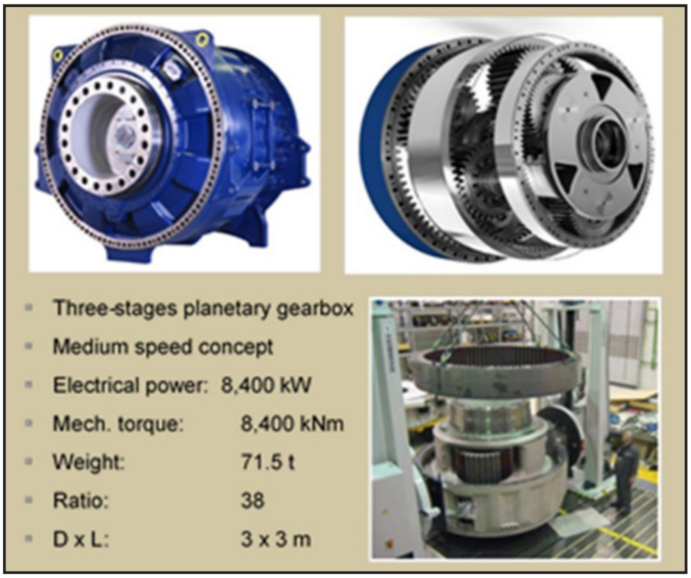

Figure 1 Design and main data of the 8 MW differential gearbox.

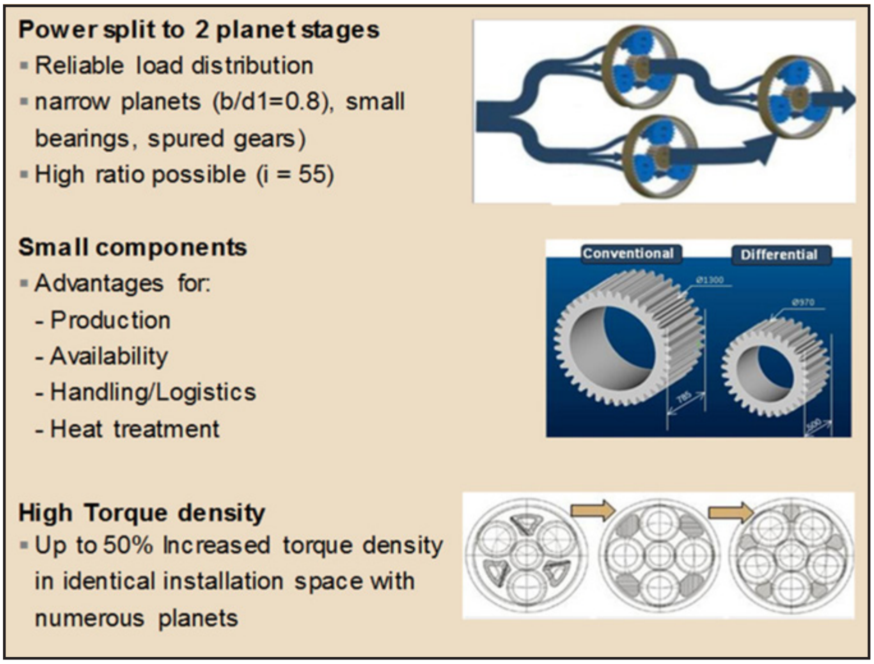

Figure 2 Characteristics of the differential gearbox concept.

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The gearbox is originally rated for 8,400 kNm, with an up-rate margin up to significantly more than 9,000 kNm without design changes.

The concept is characterized by a power split on two planetary stages on the high torque input side. A further planetary stage combines the power flow again (Fig. 2). The concept allows using comparably small gear wheels and small bearings. At the same time, high total ratios up to i = 50 can be achieved, since the stationary gear ratio is only in the range of 3.2-3.5 in each stage. The small stationary gear ratio provides the space for implementing further alignment functionality like, e.g., the double-cardanic sun system. The gear's aspect ratio is approximately 50% smaller than in conventional planetary gearbox architectures. The torque split allows using comparable small modules, which in turn lead to small gear diameters and small teeth. The component sizes and weight lead to advantages for logistics, material handling and heat treatment processes.

The differential gearbox concept provides possibilities for a 50% power density increase in the same volume by using numerous planets.

Prototype Development

Reliability engineering. A strong focus was put on reliability during the entire prototype development (Fig 3). Starting with comprehensive field experience with the differential gearbox concept, engineering experience of about 25 years in wind business and knowledge in applying sophisticated simulation tools, three elements of reliability engineering became important:

- The reliability prognosis focuses on the prediction of the component's reliability and the overall gearbox system. By an additional sub-project, applicable calculation methods have been developed (see also ZF's report on International Conference on Gears 2015 (Ref. 3)). The fundamentals of that approach are based on Bertsche (Ref. 1).
- Proper risk management by using structured methods like the failure mode effect analyses (FMEA) for designs — but also for processes. The FMEA has been carried out and updated throughout the entire development process. The method delivers input for the simulation tasks, as well as for the validation program.
- The validation program is adjusted to the verification topics. The results are processed in the FMEA and the risks are mitigated accordingly. The main pillars of the validation program are component, rig and field testing.
- Many of the described actions exceed the requirements of the IEC 61400-4 standard (Ref. 2).

Simulations. The theoretical reliability has to be safeguarded by comprehensive simulations considering all drivetrain loads. The simulation is necessary to predict early failure risks and thus reduce the validation time and costs as good as possible.

Therefore, a complete model of the drivetrain has been set up in a finite element model, as well as in a multi-body model, to adequately examine the static and dynamic influence from the overall system on the individual components (Fig. 4).

Due to the huge size of the model it was important to find

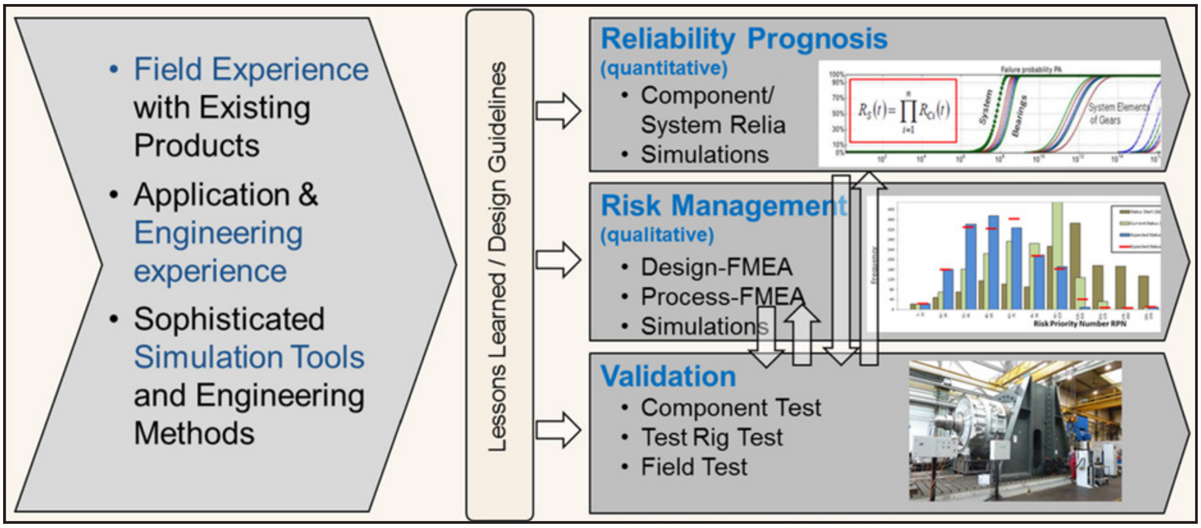

Figure 3 Basic focus points for reliability engineering.

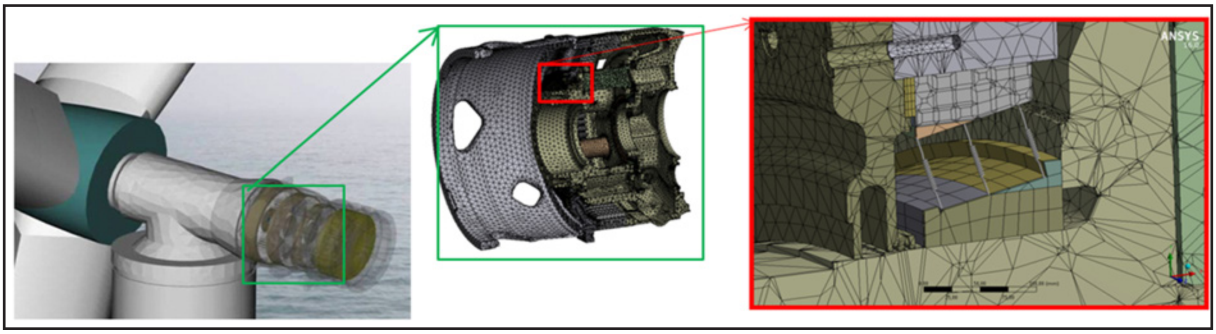

Figure 4 Simulation on system and component level.

the right modelling strategy, safeguarding the needed accuracy on one hand but keeping the computing time under control on the other hand. This means e.g. applying small mesh elements only locally or using so called "super elements" providing required physical characteristics. Particularly a suitable modelling of contact areas plays an important role to simulate accurate deformations. Using sub-models turned out to be suitable for the model exchange with the customer and the component suppliers.

For the loads and dimensions of an 8 MW turbine, the drivetrain deflections are in the order of magnitude of 1-3 mm. These influences have to be taken into account in a variety of ways in the design — especially on the component level; for example, for the bearing load distribution and the tooth modifications, which are in the order of magnitude of 10 to 30 μm.

As an example, Figure 5 shows the load distribution of a pre-loaded taper roller bearing set supporting the planet carrier; each circle represents the force on an individual roller element. Two bearing rows are shown (red: generator side bearing, blue: rotor side bearing). The state-of-the-art rigid calculation shows a proper load distribution over the rolling elements, whereas the advanced calculation reveals an unequal distribution caused by the varying stiffness of the bearing seat. Applying such an advanced method for all bearings provides an accurate assessment of the structure's influence on bearing contact pressure at static and dynamic conditions.

The structure deflection has also been taken into account to optimize tooth modifications. For example, the fixation of the ring gear by 52 pieces of M52 stud bolts leads to an axial curvature. This deflection is superposed to the usually applied crowning to determine a most accurate face load distribution in the planet/ring gear contact (Fig. 6).

The dynamic behavior of the entire drivetrain was investigated by means of multi-body simulations (MBS); thus, the interaction of drivetrain structure and gearbox was optimized. On one hand, the Eigen frequencies of the components were adjusted in order to avoid harming resonance conditions; on the other hand, the gear mesh excitation was reduced by investigating the total pitch error of each planetary stage as well as the stiffness variation in the gear meshes (Fig. 7).

Applying a proper teeth number so that a synchronous or asynchronous mesh of sun/planet and planet/ring gear occurs, has been well proven; the right choice depends on the interaction with the gearbox structure. Also, the stiffness variation over the path of contact for both — sun/plant and planet/ring gear mesh — plays an important role for the total pitch error. Finally, a proper microgeometry is useful to influence the mesh frequency. In this case it turned out to be important to focus on the right excitation mode, i.e. — the basic mesh frequency or one of their harmonics.

The structure-borne noise could thus be reduced by approximately 95% — comparing the origin to the optimized design. This means excitation amplitudes in the range of 10 mm/s were simulated with a non-optimized design respective in the order of magnitude of 1.5 mm/s for the optimized design.

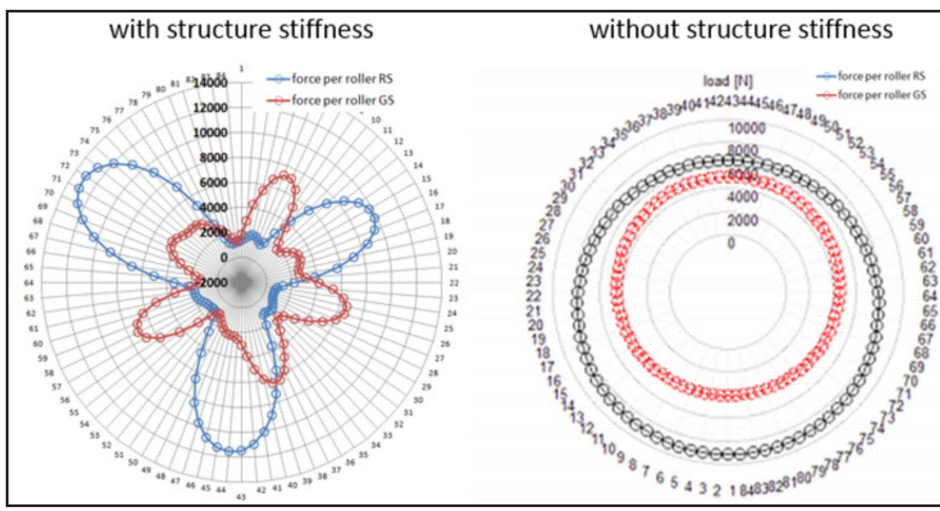

Figure 5 Bearing load distribution with state-of-the-art and advanced simulations.

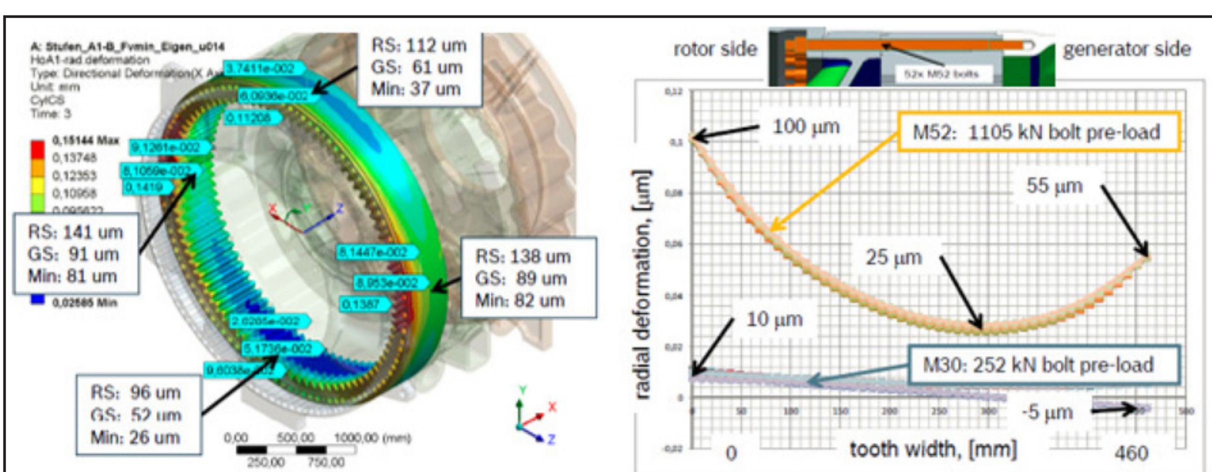

Figure 6 Gear modifications by considering structure deflections.

Validation. According to the risks detected during FMEA and simulation a validation procedure was set up. Different tests were conducted like component tests, robustness and endurance tests as well as a field test (Fig. 8).

With various component tests, several sub-assemblies and process steps were validated; special attention was paid to the roller bearings. Several ZF-owned bearing test rigs were used for that purpose. Also the heat treatment process for the present gears, comprising a module of 29.5 mm, was investigated safely. Destructive component tests supported by heat treatment simulations were conducted. The target was to find a proper combination of surface and core hardness for case hardening depths of around 3-6 mm. Choosing the right quenching technology is a success factor.

The overall system behavior was investigated on the system test rigs, applying dynamic overloads up to 190% of the rated

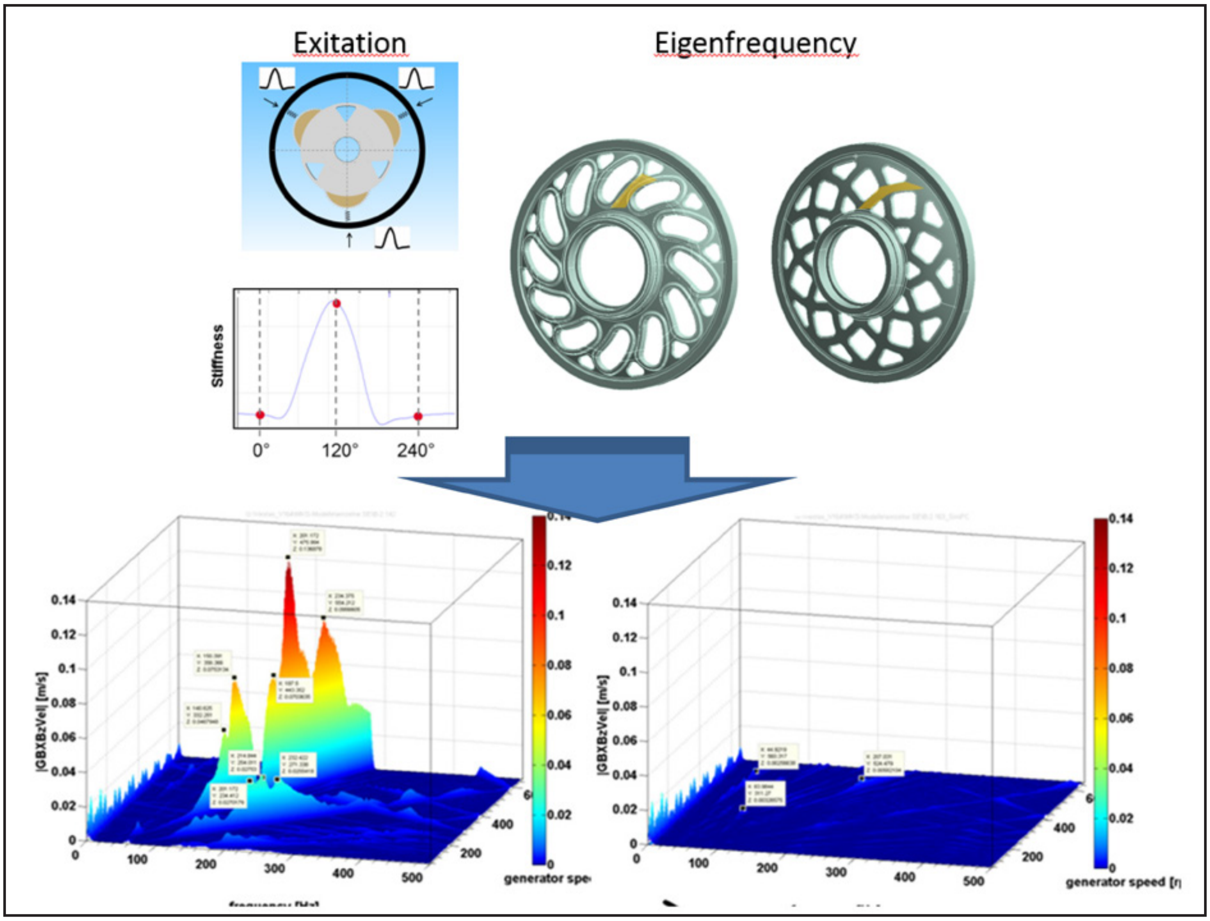

Figure 7 Prediction of structure-borne noise by optimization of structure shape, gear macro- and micro-geometry.

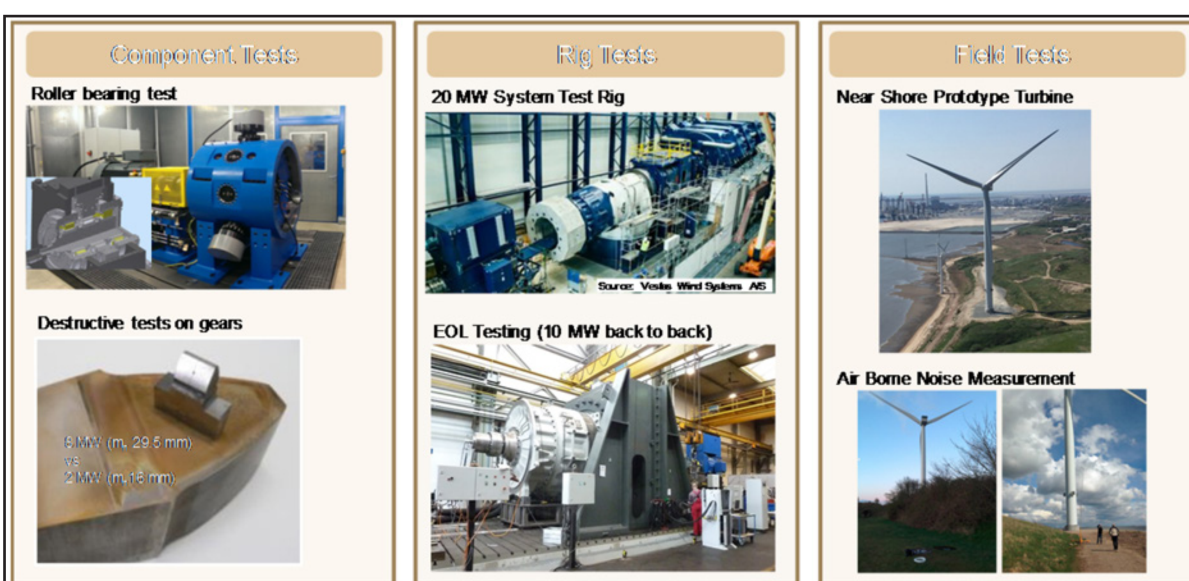

Figure 8 Validation procedure with component, system and field tests.

torque with a duration of more than 1,500 hours. The focus here was on the load distribution in contact elements and on the deformation of the structure. After the system tests the gearboxes were disassembled. All contact surfaces — meaning rolling contact elements as well as flange connections — have been inspected and compared to the simulation.

The inspection revealed that the contact pattern of functional surfaces, as well as the observed micro-movement in flange surfaces, appeared as simulated.

In the upper part (of Fig. 9 as an example), a connection element of the first and second planetary stage is shown. The inspection (left-hand side) demonstrates that the contact location to the mating part is of the same size and at the same location as previously simulated (right-hand side). A reverse calculation of the occurred deformation and pressure was possible.

The lower left part shows the flange connection of the ring gear to the housing. The dark grey zones reveal micro-movement in the flange connection. The lower right picture shows the corresponding simulation. One can see that the micro-movement area (red color) correspond to the inspected

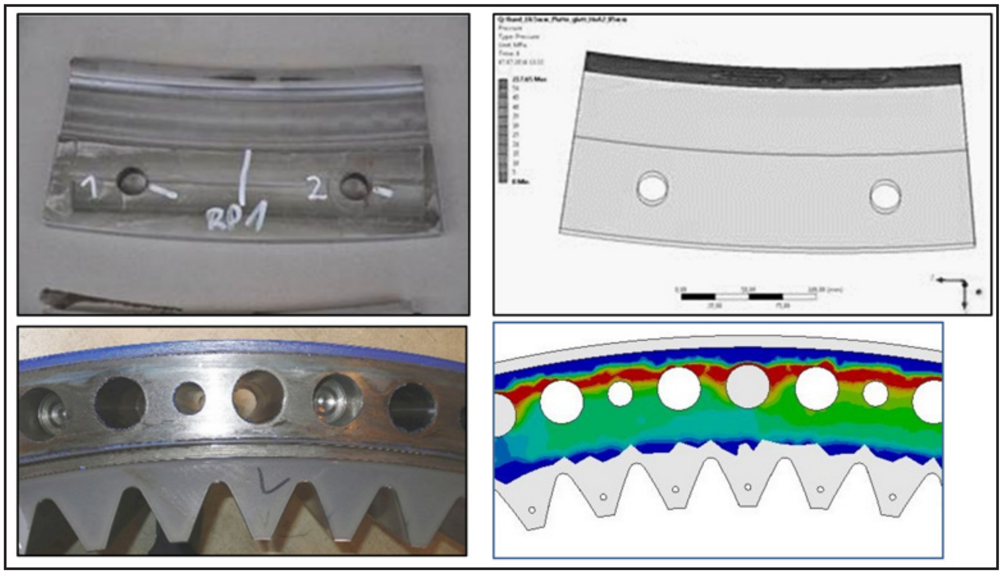

Figure 9 Comparison of system deformation and micro-movement to predicted values.

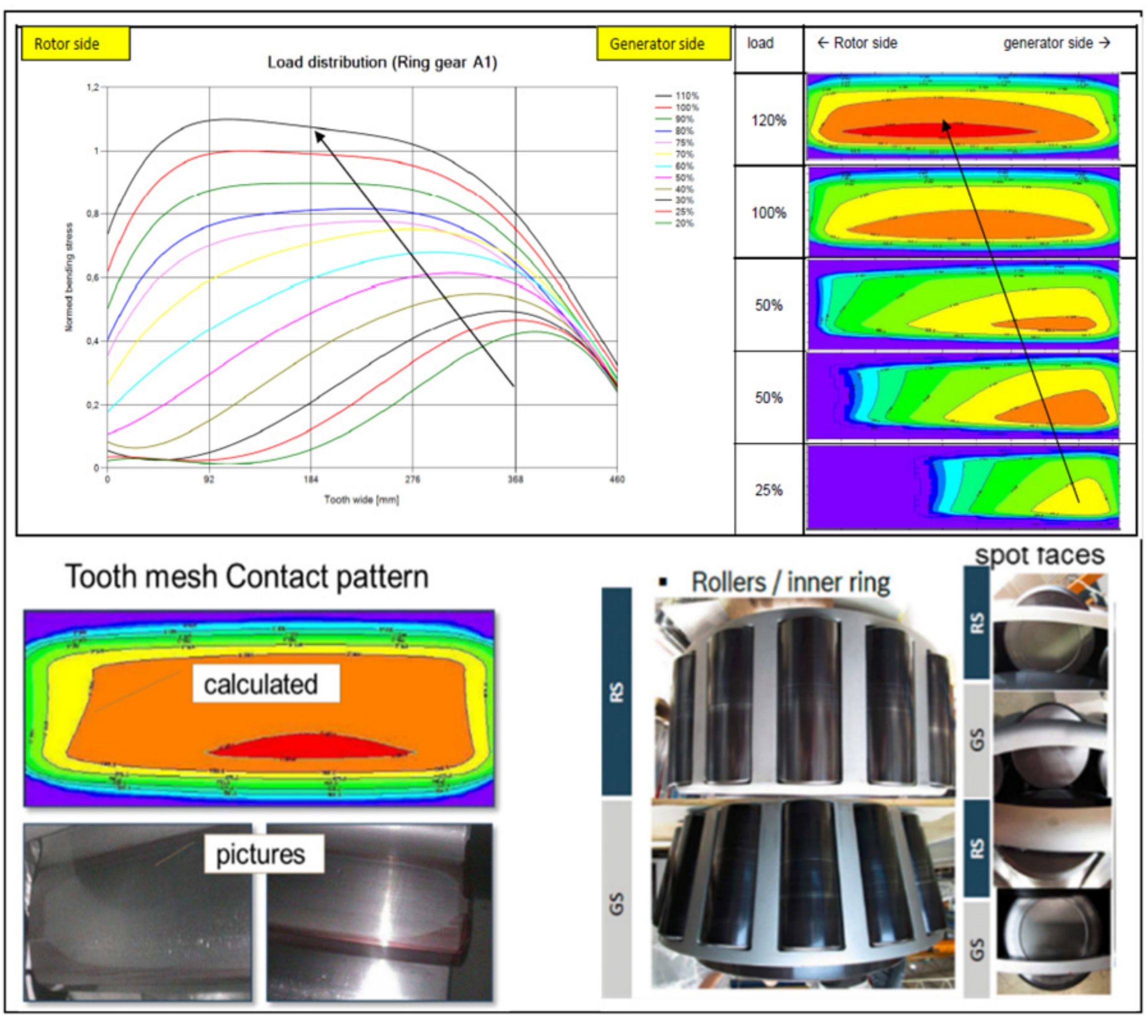

Figure 10 Contact pattern of gears and bearings and comparison to measurement values.

micro-movement. By means of an accurate simulation the flange and pin/screw design can be optimized such that the amount of micromovent is kept in a controlled and acceptable value. The acceptable values could be derived from field experience.

As usual for wind gearboxes, all meshes have been instrumented by strain gauges to measure the face load distribution, as well as the load sharing between the planets. Also the contact pattern development over several load stages has been visually inspected.

A measured face load distribution of $K_{H\beta}$ 1.15…1.17 and a load sharing of $K_\gamma < 1.05$ has confirmed the gear mesh simulations (Fig. 10). Also, the pre-loaded taper roller bearings revealed an expected contact pattern.

Also the structure-borne noise measurement confirmed the simulated values, i.e.—a deviation of below 5% occurred between simulation and measurement (Fig. 11). The excitation levels are in the range of approximately 1.6 mm/s (e.g. first order of gear mesh frequency), and the Eigen frequencies modes are as predicted. The finally applied airborne noise measurement demonstrated that no tonal audibility occurred and that the turbine fulfills onshore noise requirements.

Optimization of Serial Production

Already during the prototype development phase, manufacturing design was a significant factor. Further process optimization had been reached in a subsequent project, which started with entering the zero-series phase.

Complex correlations between machine and process influences on one hand, and measurement results, on the other hand, have been identified by production-oriented tolerance evaluation with statistical methods (Fig. 12).

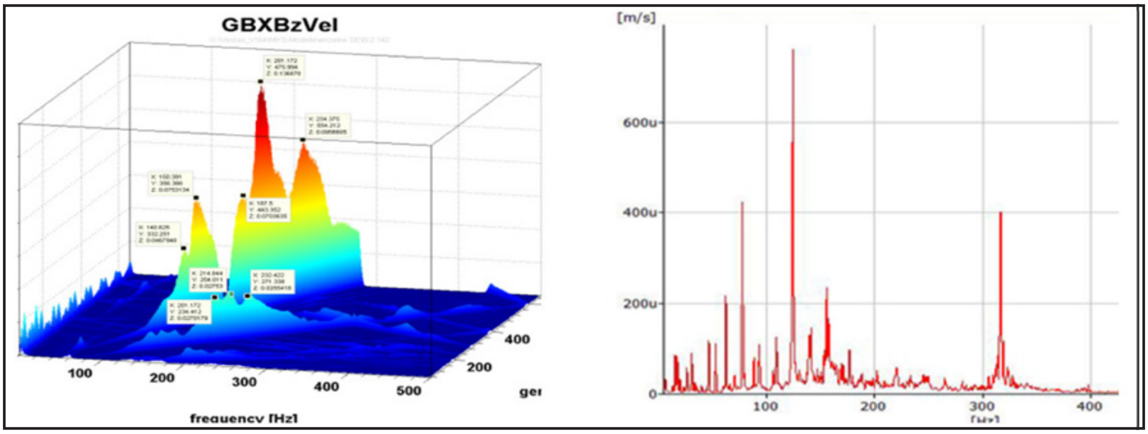

Figure 11 Structure-borne noise prediction (left) and measurement (right).

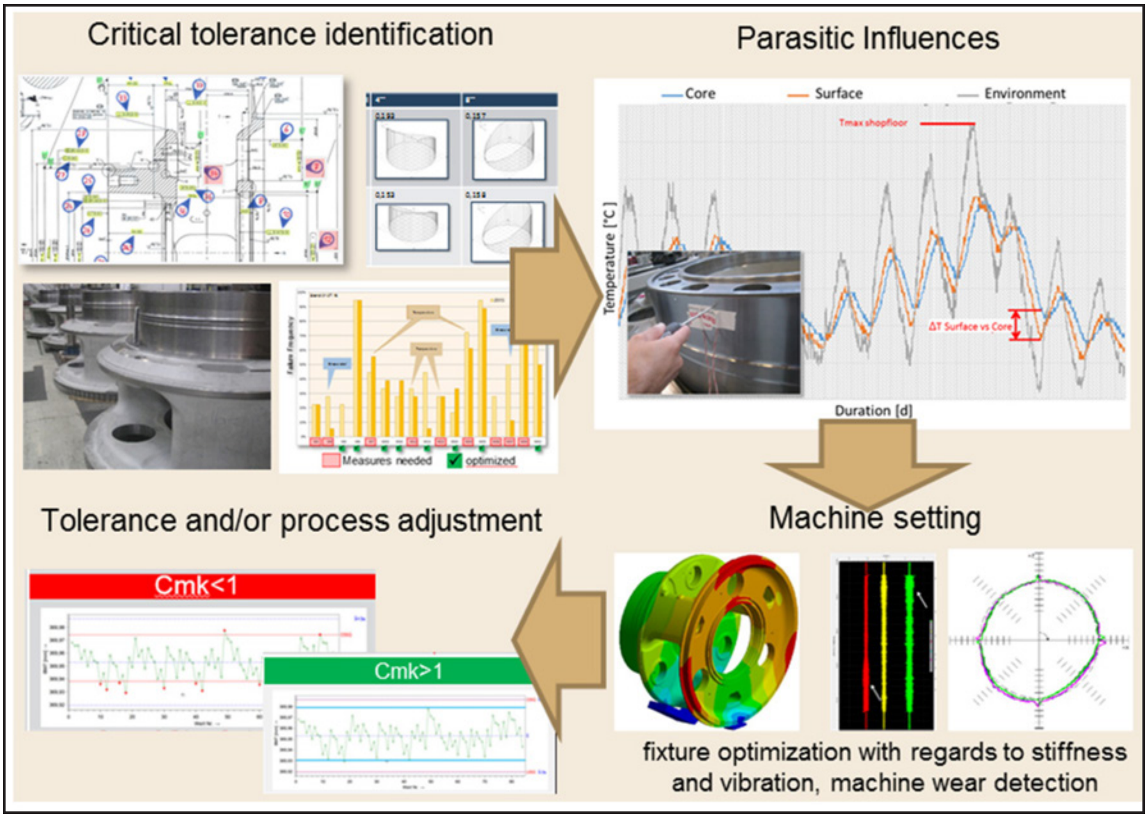

Figure 12 Process and tolerance optimization with focus on cost and quality.

Amongst others, the temperature behavior of big-size structure components (e.g. the temperature development from surfaces to core at different wall thicknesses and shapes), has been investigated and implemented in the measurement strategy.

For a better understanding of the component's deformation in clamping devices, their behavior has been investigated by using FEM analysis; the findings could be used for an enhanced manufacturing strategy with optimized fixations. Additionally, a regular monitoring of the machine conditions has been introduced. Besides the geometrical measurement, frequency analysis of selected units leads to a permanent view on the process conditions and allows measures at an early stage.

A calculation tool has been developed and implemented with the start of series production. Based on the analysis and display of the grinding stock data of gears before grinding, this tool permits quality and cost improvements (Fig. 13).

This tool enables a systematic regulation of the grinding allowance for each type of work piece, and a systematic reduction of the allowance variation spread by identifying and eliminating its root causes.

By using all these strategies on fields of environment, machine, process management and measurement, it is possible to obtain process capability and to face the triangle of tension: time, cost and reliability.

Not only for manufacturing but also for assembly of the 8MW gearbox, special adaptions to the processes had to be done (Fig. 14).

One major aspect concerns the health and safety of the operators. Exemplary for several special tools, the picture on the left side (Fig. 14) shows the adjustable gantry in the main assembly. With respect to the dimensions of components and sub-assemblies, one critical process is the mounting of gears. To minimize the risk of damage, as well as assembly time and efforts for mounting devices, it is necessary to optimize the shape of the facing edge. The shape has to fulfill the requirements of the two different mounting phases, i.e.—"finding" and "sliding." The angle of the front edge needs to be big enough for parts finding and the angle for transition into the flank needs to be small enough to ensure a smooth sliding.

For a permanent, stable operation of the gearbox, the quality and the process capability of screw connections are of vital importance. All internal screws and all torque-transmitting

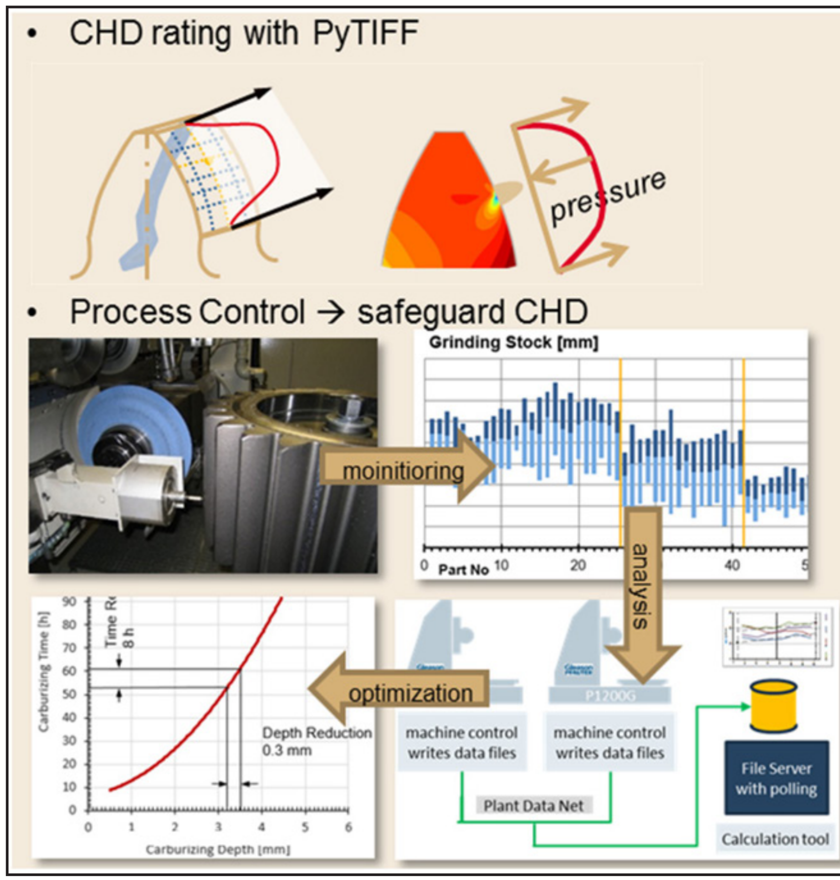

Figure 13 Process optimization with focus on reliability.

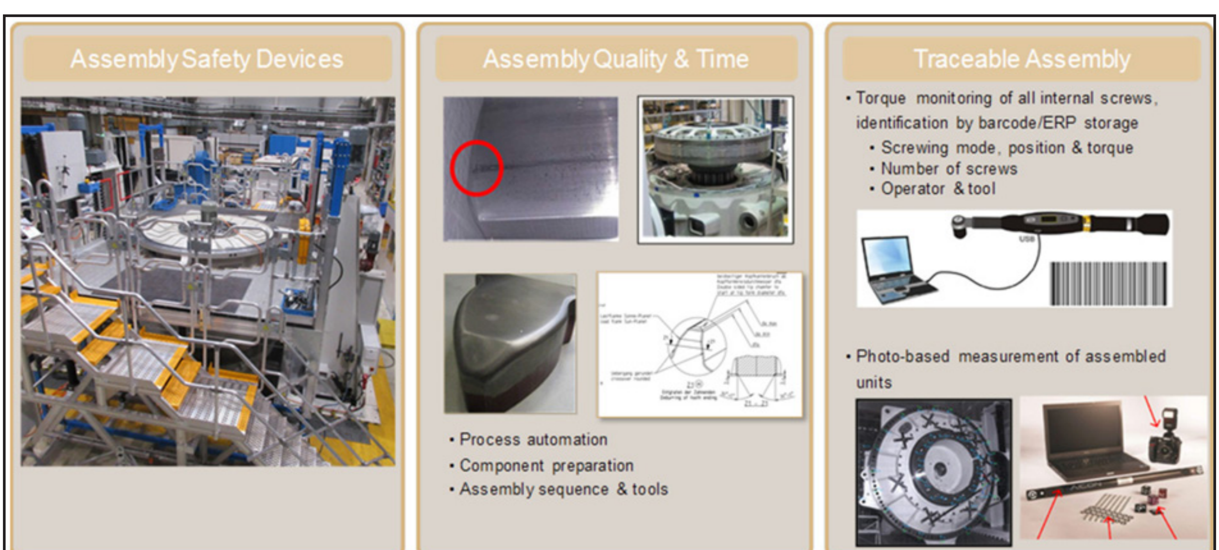

Figure 14 Stabilization of serial production with focus on quality, health and safety.

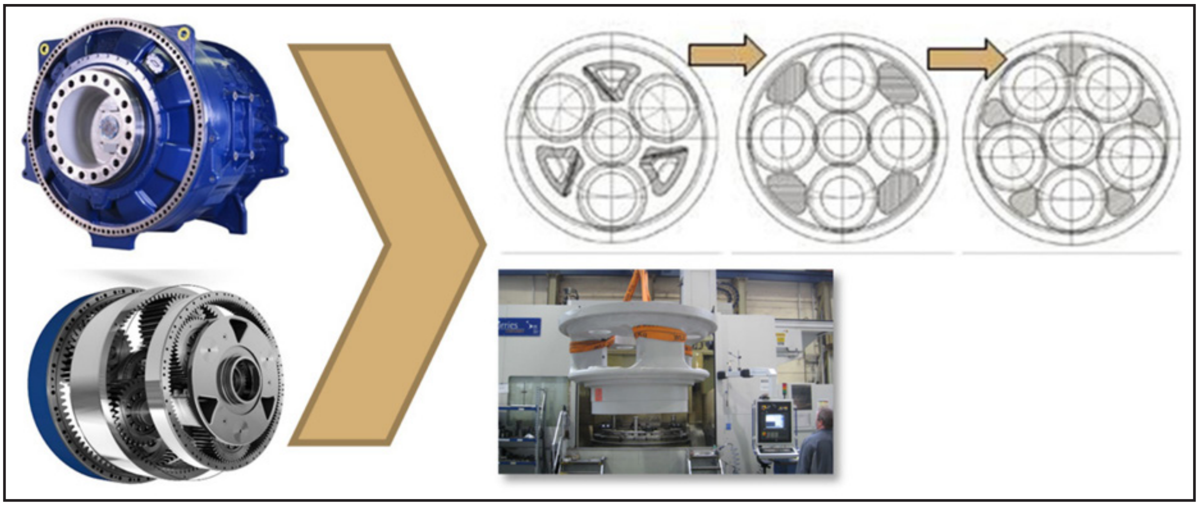

Figure 15 Power increase with differential gearbox concept.

screws are tightened with electronic torque control.

The interface dimensions of the assembled product are measured by means of advanced measurement methods — here, with a photo-based system. For this reason the gearbox is equipped with reference points and then photographed. With a special software tool, the interfaces dimensions are evaluated with accuracy in the range of about 0.05 to 0.15 mm, depending on the kind of tolerance. This method allows a quick measurement without any need for accessing a measurement machine.

Summary and Outlook

The offshore application requires the highest reliability to safeguard business certainty. At the same time, increasing component sizes challenge the entire supply chain. Thus, ZF focused consequently on reliability engineering for the 8 MW differential gearbox, as well as on reliable serial processes.

In future the focus will be on operational and field data exchange in order to close the information loop and verify further the reliability models. The proven differential gearbox concept yields a power increase up to at least 12 MW by use of same gearbox size and utilization of the existing supply chain and facilities (Fig. 15). **PTE**

For more information.

Questions or comments regarding this paper? Contact Dirk Strasser at *dirk.strasser@zf.com*.

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Dr. Dirk Strasser in 2000 graduated from FH Iserlohn and Ruhr University Bochum with a degree in mechanical engineering, focusing on construction. His professional work includes (2000-2005) research associate, Institute of Machine Elements, Transmissions and Motor Vehicles, with Prof. Predki, Ruhr University Bochum; (2005-2010) in various positions in the field of industrial and wind gear design for medium-sized transmission companies in the Ruhr area; and (2010-2015) as head of wind gearbox development, Bosch-Rexroth, Witten. Strasser currently (since 2015) has global responsibility for the development of wind gearboxes for ZF Wind Power Antwerp (after sale of Bosch Windbusiness to ZF in 2015).

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Dipl.-Ing. Ralf Sperlich began his schooling in 1988 -1991 with an apprenticeship as tool mechanic at VDO Adolf Schindling AG, Frankfurt/M, Germany. From 1994-1997 he pursued mechanical engineering studies at the University of Applied Sciences FH, Frankfurt/M, Germany. From 2005 thru 2007 he took up French Language Studies at the Centre d'Enseignement du Français, Albertville, France and theological studies at Emmaüs, St-Légier, Switzerland. He was later (1997-1998) a design engineer for the mechanical components of robots and hydraulic presses (up to 200kN) at Reis Robotics, Obernburg, Bavaria; (1999-2005) calculation engineer for cranes and components, focus area calculation of machine elements for helical, hypoid and bevel gearboxes for industrial use DEMAG Cranes & Components GmbH, Wetter/Ruhr, Germany; (2006-2012) social welfare development work, in cooperation with international partner organizations and the Evangelical Church in Chad and director of an orphanage there for SAHEL LIFE, Kirchheim/Teck, Germany. Since 2012 Sperlich is design engineer for multi-megawatt offshore wind turbine gearboxes, design lead engineer, for ZF Industrieantriebe Witten GmbH, Witten, Germany.

Jörg Münch in 1983 received vocational training as a mechanic. In 1988 received his degree in mechanical engineering from the University of GHS Wuppertal. Upon graduation, his professional work includes: (1998) gear factory at Köllmann GmbH, in the design and development of extruder and special gearboxes; (2001) at Jahnel-Kestermann GmbH & Co. KG designing and developing special gearboxes (rolling mill, drives chemical plants dredger pumps, water turbine drive); and, since 2007, at Bosch Rexroth AG Witten, in the design and development of transmissions for wind power transmissions. Most recently, Münch is in charge of the 8 MW GPC 840 D gearbox.