Building Reliable Wind Energy Gears without Micropitting

1 Introduction

Founded in 2008, SET is an innovative company developing the DSgen-set®, an advanced variable-speed medium-voltage drive system with a nominal power rating of ≥2.0MW. The system's planetary gear stage is controlled by a servo drive which compensates for the varying speed of the wind energy converter's (WEC) rotor and guarantees constant speed in the synchronous generator.

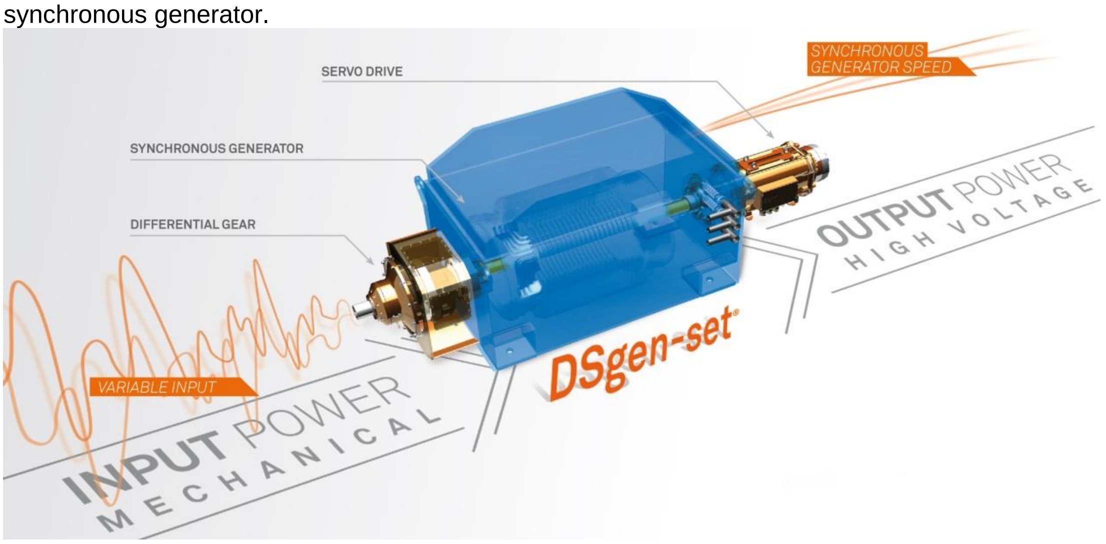


Figure 1: DSgen-set®

SET optimized the new system to achieve a total system efficiency of 97%, while keeping system costs below benchmark level. To achieve such high efficiency and minimize costs, every component needs to be analyzed and optimized to the very limit (in the development process).
The same is true for the gearbox, which is very compact and highly efficient at the same time. Designing a very compact gearbox requires careful examination and analysis of the gear stage along with the other components. SET uses various state-of-the-art analysis tools (AVL Excite, KISSsoft/sys, Ansys, etc.) in this process. This drastically reduces the risk of any kind of damage, especially the risk of micropitting.

2 MBS Simulation in Analysis of Axis Misalignment

Consideration of the manufacturing process, manufacturing tolerances, and even axis misalignment (axis deviation and inclination errors) is currently state of the art in every gear stage design process, particularly in the contact analysis of gears in mesh. Dealing with compact designs usually means a small pinion is paired with a considerably larger gear where a high ratio of pinion width to pinion diameter is often the case. As a result, there is a high risk of micropitting. Furthermore, the risk of micropitting at the tooth ends is even higher when the micro geometry corrections are not designed properly. In order to reduce this hazard, the axis misalignment needs to be determined as accurately as possible.
With innovative approaches using multi-body simulation (MBS) and simulating the exact gear movement (depending on the level of modeling detail), the information missing to date are obtained.

Introducing a new design methodology is a major advantage in determining axis misalignment for both inclination and deviation error, thus greatly reducing the risk of failure, especially the risk ofmicropitting.MBS Model

We at SET do perform MBS simulation using AVL Excite software by importing condensed matrices with exactly the same properties as the real parts. These parts are then linked using various joining elements representing the desired connection (gear joint, bearing joint, spline, etc.). The model accuracy and proper joint settings in the model affects the stability as well as the simulation accuracy. SET therefore makes a great effort to determine these properties correctly.

The properties of spline connections are determined by means of FEM analysis. The input geometry of hub and shaft are created in KISSsoft, where manufacturing tolerances are included. Later on, the calculated backlash of the spline connections is removed in order to improve the stability of the MBS. Each spline connection is split into two nodes per spline width (two nodes per connected part).

The properties of the bearings (radial and axial stiffness) are taken as provided by the bearing manufacturer. The bearing connection is made between the connecting node on the inner and outer bearing seat. The inertia properties of bearing rings and rolls are applied to the particular node as a parameter.

The properties of gear connections are calculated in KISSsoft. It uses the theoretical stiffness, free of any micro-geometry corrections, for the specific gear mesh (in the second optimization loop, the meshing stiffness of corrected gear could be used). No backlash is included in the connection because it would decrease the stability of the MBS. The gear is linked with only one node (per part) for each connection used. Nevertheless, two additional axially split nodes are created per gear. We use these nodes for determining the axis misalignment.

The synchronous generator's rotor is connected to the "electrical grid" by a special connection which represents the magnetic force (magnetic torque).

The servo motor is modeled with a similar connection to the synchronous generator.

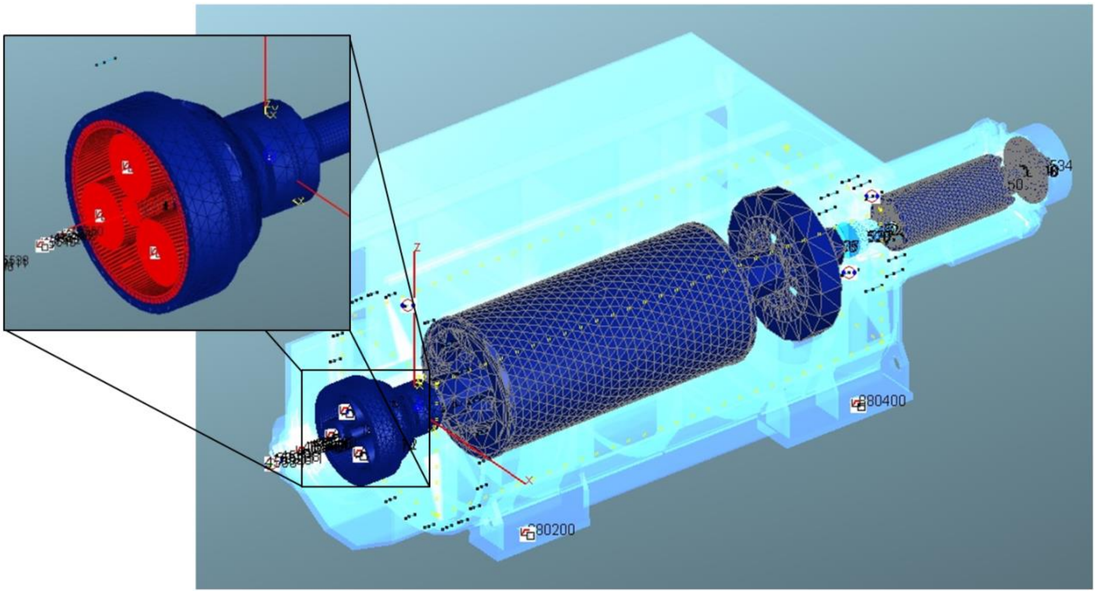


Figure 2: MBS model

Axis Misalignment

The MBS results need additional post processing before they are in the appropriate form for contact analysis of the gear mesh. The displacements of nodes of interest are evaluated in the global coordinate system.

Evaluation of axis misalignment for the internal gear and sun gear is straightforward. First, the difference between both nodes for the x and z components (the y component is neglected – axial direction) of the displacement vector needs to be calculated, equation (1).

$$\begin{aligned} \Delta \boldsymbol{x} &= \boldsymbol{x}_{II} - \boldsymbol{x}_{I} \\ \Delta \boldsymbol{z} &= \boldsymbol{z}_{II} - \boldsymbol{z}_{I} \end{aligned} \tag{1}$$

While a value for axis misalignment relative to the internal gear is relevant in KISSsoft, few recalculation steps are required to prepare the misalignment values in the correct form, equation (2).

$$\begin{aligned} f_{DEV,INC}^{sun-int\,(1)} &= \max_{sun}\left(\sqrt{\Delta x^2 + \Delta z^2}\right) - \min_{int}\left(\sqrt{\Delta x^2 + \Delta z^2}\right) \\ f_{DEV,INC}^{sun-int\,(2)} &= \max_{int}\left(\sqrt{\Delta x^2 + \Delta z^2}\right) - \min_{sun}\left(\sqrt{\Delta x^2 + \Delta z^2}\right) \\ \left|f_{DEV,INC}^{sun-int}\right| &= \max\left(f_{DEV,INC}^{sun-int\,(1)}, f_{DEV,INC}^{sun-int\,(2)}\right) \end{aligned} \tag{2}$$

Although equation (2) calculates the relative axis misalignment between the internal and sun gear, the sun gear misalignment affects the planet gear-sun gear meshing. Therefore, the value should be taken as a magnitude which alternates between negative and positive and between axis deviation and axis inclination error.

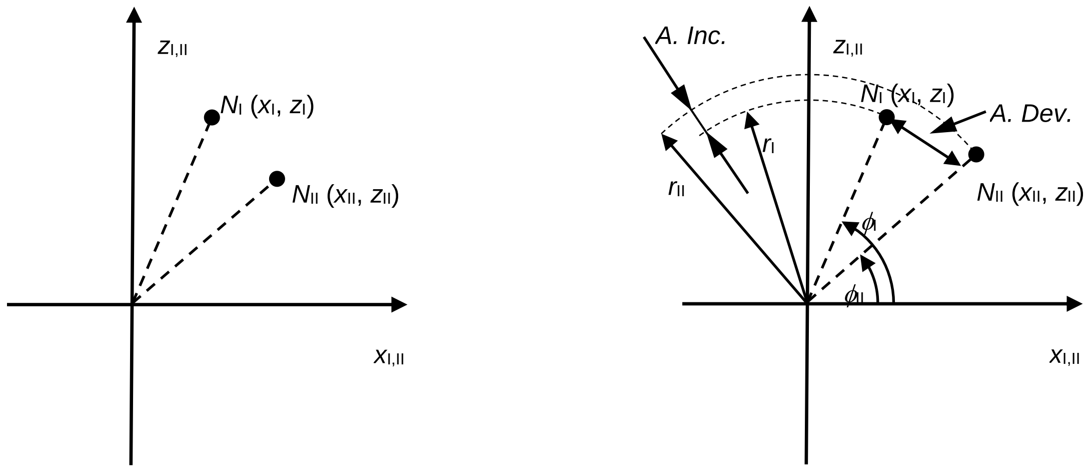


Figure 3: Coordinate system (left) and planet gear axis misalignment (right) description

Evaluating the influence of the internal and sun gear on the planet gear axis (relative) misalignment is not so straightforward. Applying equation (1) and (2) would produce inaccurate results. The gear mesh changes its location with the rotation of the planet carrier.

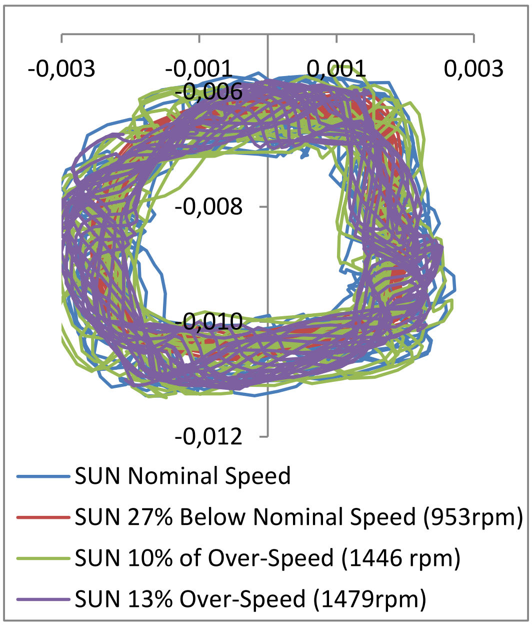


Figure 4: Axis misalign. of sun gear (in *mm*)

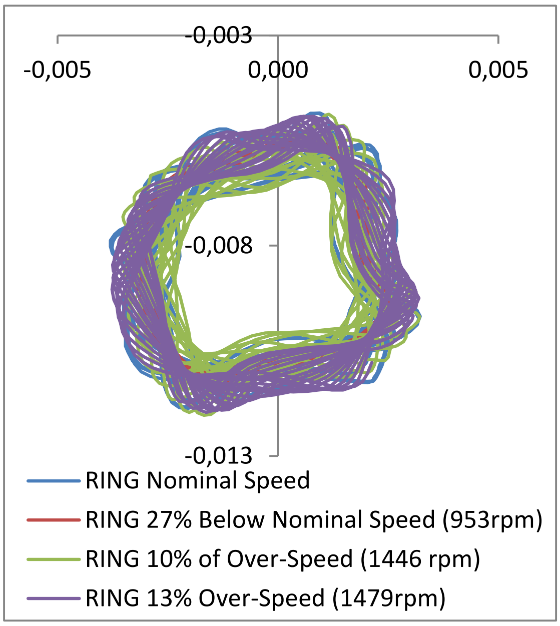


Figure 5: Axis misalign. of internal gear (in *mm*)

Here additional methodology is used. The node on side I of the gear mesh serves as a reference node. If the node on side II lags behind (in the circumferential direction; the planet carrier is twisted), Figure 3 (right), the lag would add to the planet gear axis deviation error, equation (3). If the node on side II is further away in the radial direction, the difference in the radius between them will be added to the axis inclination error equation (4).

$$f_{DEV}^{pl} = \bar{r} \cdot \tan \Delta\varphi$$
$$\Delta\varphi = \tan^{-1}\frac{z_{II}}{x_{II}} - \tan^{-1}\frac{z_I}{x_I} \qquad (3)$$
$$\bar{r} = \frac{\sqrt{x_{II}^2 + z_{II}^2} + \sqrt{x_I^2 + z_I^2}}{2}$$

$$f_{INC}^{pl} = \sqrt{x_{II}^2 + z_{II}^2} - \sqrt{x_I^2 + z_I^2} \qquad (4)$$

In equations (3) and (4), only the displacement of the planet gear nodes is considered. The same assumption as for the sun gear applies when the effect on the internal gear-planet gear mesh is evaluated. If the internal gear is misaligned by a constant value, from the planet carrier's point of view, the planet gear "sees" the internal gear misalignment rotating around the planet carrier's axis. Alternating axis deviation and axis inclination errors are different in phase, but have the same amplitude. The magnitude of the axis inclination and deviation error is exactly the maximum axis misalignment of the internal gear.

The axis deviation and axis inclination error for both the internal gear-planet gear mesh and for the sun gear-planet gear mesh is evaluated using equations (5) and (6) respectively.

$$f_{DEV}^{int-pl} = f_{DEV}^{pl} \pm \max r^{int}$$
$$f_{INC}^{int-pl} = f_{INC}^{pl} \pm \max r^{int} \qquad (5)$$

$$f_{DEV}^{sun-pl} = f_{DEV}^{pl} \pm \max r^{sun}$$
$$f_{INC}^{sun-pl} = f_{INC}^{pl} \pm \max r^{sun} \qquad (6)$$

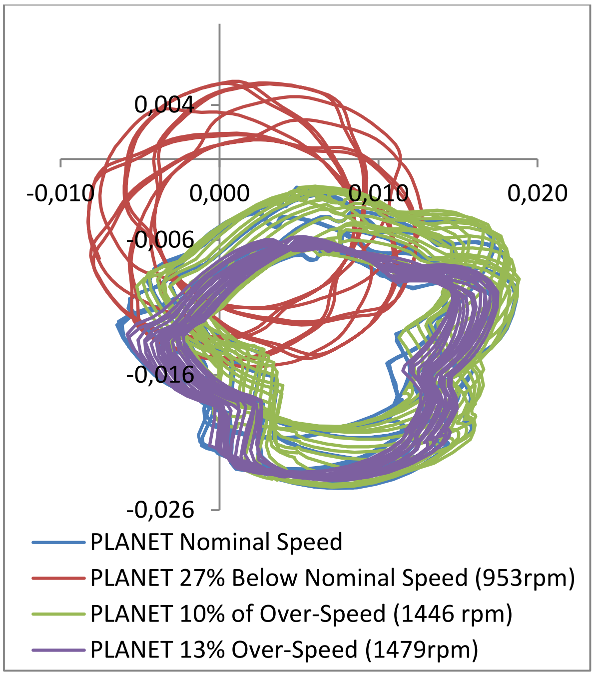


Figure 6: Planet gear axis deviation (horizontal) and inclination (vertical) error (in *mm*)

In general, evaluation and contact analysis can be performed using other values like mean, median or even values with certain survivability (at least 99%) instead of maximum or minimum.

Table 1: Axis deviation & inclination errors

		Sun Gear		Planet Gear	
		Value	Alternation	Value	Alternation
Rated speed	Dev.	0 µm	±7 µm	19 µm	±12 µm
	Inc.	0 µm	±7 µm	-24 µm	±12 µm
27% below rated speed	Dev.	0 µm	±5 µm	13 µm	±11 µm
	Inc.	0 µm	±5 µm	-15 µm	±11 µm
10% over-speed	Dev.	0 µm	±6 µm	19 µm	±12 µm
	Inc.	0 µm	±6 µm	-24 µm	±12 µm
13% over-speed	Dev.	0 µm	±6 µm	18 µm	±12 µm
	Inc.	0 µm	±6 µm	-24 µm	±12 µm

3 MBS Model Validation

The new method for determining axis misalignment depends solely on the accuracy of the MBS model and the accuracy of the input data; hence it directly affects the estimation of the micro-pitting risk. To improve our model and the accuracy of the properties of various joint elements, the model needs to be validated.

At SET's 3MW testing facility, several validation procedures were performed, such as comparing torque, speed and acceleration at the observation points. They also include a special contact pattern test. The test provides us with an exact rating regarding axis misalignment and the accuracy of the MBS model. Figure 7 illustrates a painted part portion of the gear in another system after the contact pattern test.

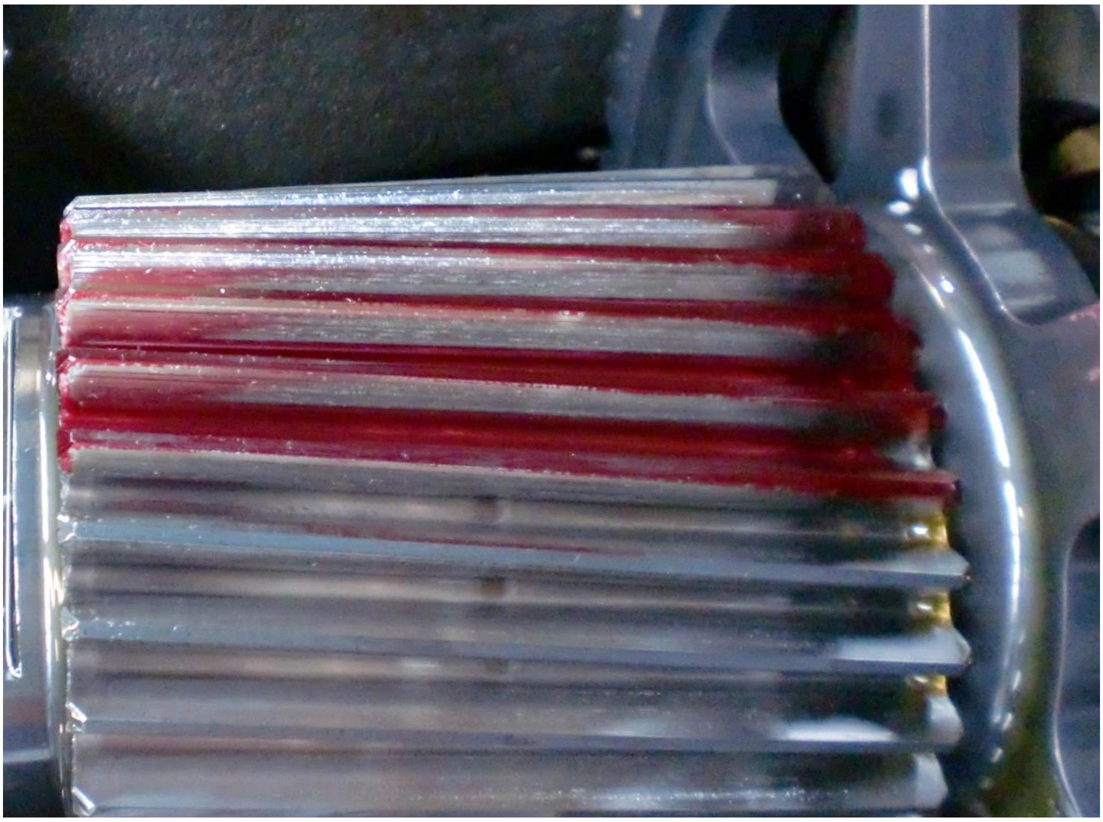


Figure 7: Contact pattern test on a planet gear

4 Definition of micro geometry with KISSsoft

Using the axis deviation and inclination errors between the planet gear and the sun gear the optimal micro geometry has to be defined. The axis deviation was selected at rated speed/load, since the overload speeds didn't show any higher values (see table 1). The alternation values of the axis represent the manufacturing tolerances. The contact between sun and planet gear without any corrections is shown in picture 8. As the gears have rather high face width (ratio b/m_{nm} = 30) the torsion on the sun gear leads to high deformation in lengthwise direction which has to be compensated with helix angle modification and crowning.

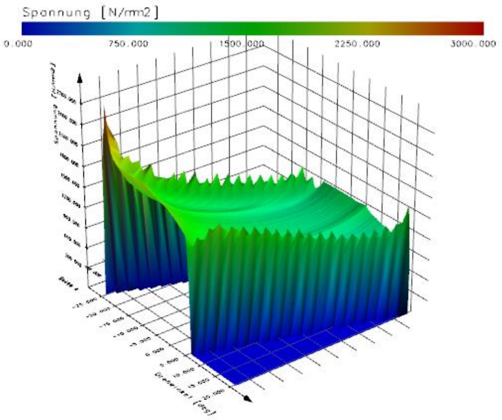


Figure 9: Contact situation under load without any lead and profile modification

The optimization was performed in two steps: in the first step the face load distribution was balanced including the manufacturing tolerances, whereas the goal of the second step the optimization of profile correction was done for minimizing the risk of micro pitting, but however, also considering the main parameters as Hertzian pressure etc

The modification regarding face load distribution was done using helix angle correction (on the sun gear) as well as lengthwise crowning to compensate the deformation due to torsional load on sun and planet carrier. The manufacturing tolerances (toothing and axis alignment tolerance f_{ma} with ±7μm for the sun and ±12μm for the planet gear) are both compensated with additional crowning, however this would lead to concentrated contact under light duty. So often the modification with end relief (with arc like transition) is preferred and applied here as well with 10μm (both ends, results in totally 20μm). The calculation of the load distribution as described in ISO6336-1, annex E, is a very accurate method, and much faster in terms of calculation time than a complet contact analysis. Therefore the optimization was quickly guided by calculating the $K_{H\beta}$ factor based on this annex E and finally checked using the graphic with the pressure distribution in the contact analysis (see figure 10).

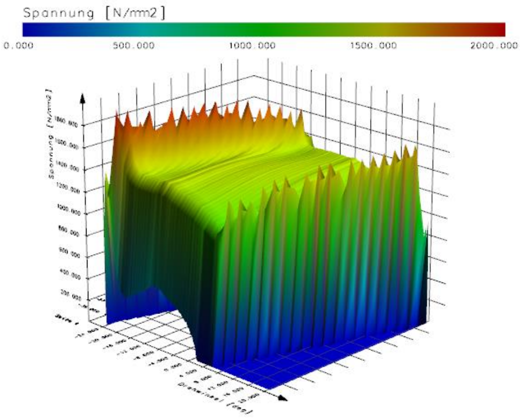
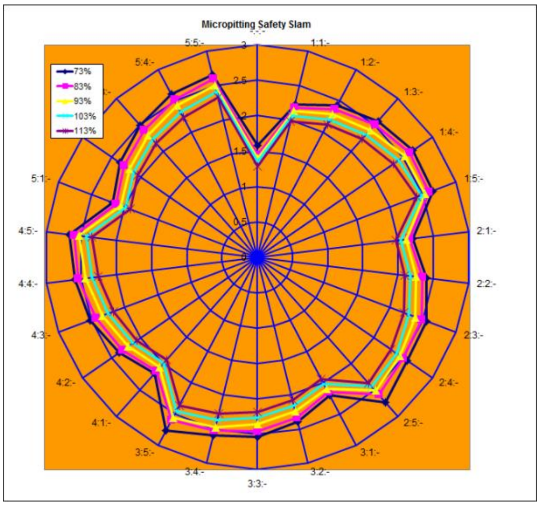


Figure 10: Contact analysis after definition of lead modification

In a second step the profile modifications were defined, focusing on micropitting but as well as Hertzian pressure. The risk of micropitting is highly influenced by the optimal tip relief. This was achieved using the KISSsoft sizing tool for micro geometry by automatically varying different combinations of profile modifications and calculating all the results with contact analysis method. With the specific evaluation table the optimal result was selected, considering the safety factor against micropitting and maximum Hertzian stress. This calculation procedure corresponds to method A of ISO/TR 15144-1:2010 (see earlier publication [1]).

For the optimal profile modification of the sun and planet gear the tip relief C_a was varied from 40 to 50μm, with length variation from factors 1.2 to 2.3 in totally 5 steps. The length factors correspond to the short resp. long tip relief and both the amount of tip relief and length factors were derived by using the KISSsoft sizing functionality. The load was varied between 73% and 113%, which are given in table 1. Using these inputs, all possible combinations were checked, combined with 5 torque levels. The results regarding highest safety against micropitting and lowest Hertzian pressure are shown in figure 11, whereas the first solution (marked as "-:-:-") represents the completely unmodified gear.



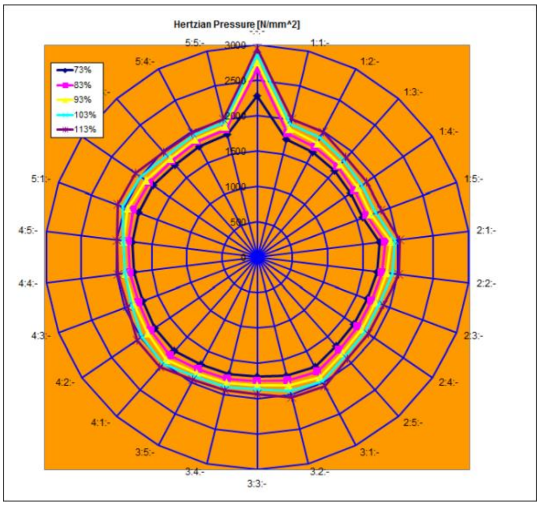


Figure 11: Evaluation of different profile modifications (left: Safety against micropitting; right: Maximum Hertzian pressure in the contact)

It is clearly visible that the safety against risk of micropitting is highest with the combination "1:5:-" which achieves a safety factor 2.5. The Hertzian pressure is lowest with the solutions "1:5:-", which means 1781 N/mm2. So this results is to be preferred and applied to the gears (sun gear with C_a = 40 and factor 1.2, planet gear with C_a = 50µm and length factor = 2.3). A careful checking of the contact situation from planet gear to ring gear finalizes the optimization (fig. 12).

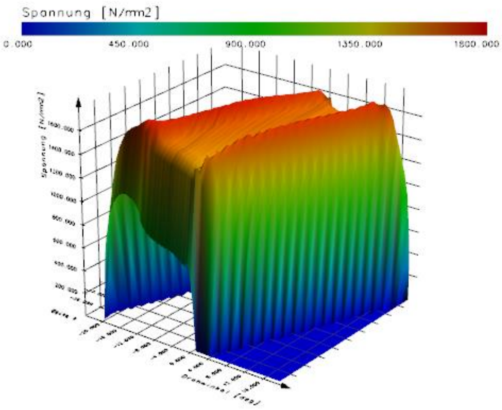


Figure 12: Final stress distribution using flank and profile modification

5 Conclusion

Micropitting in gears is a phenomenon which was detected and analyzed mainly in wind turbine gearboxes. SET uses innovative calculation tools in order to provide a detailed analysis of planetary gear sets to get an optimal design. The MBS software AVL EXITE determines the deformation within the planetary system, whereas with KISSsoft the optimal gear micro geometry is defined. The safety against micropitting could be increased up to 2.5, which means a perfect result with very low risk of failure.

Literature:

[1] U. Kissling: The application of the first international calculation method for Micropitting, AGMA FTM fall technical meeting, 2012.