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KISSsys application:

Lifetime analysis of a 4MW wind turbine gearbox:

Two planetary stages plus one helical stage



1 Task

In order to compare different gears with respect to their lifetime in short time, a model of a complete gearbox including all relevant parameters shall be established. The lifetime analysis for the gears, shafts and bearings shall be performed for a user defined load spectra (speed and torque versus time on the input shaft). The results are summarized in a report.

2 Solution

Using KISSsys the complete gearbox, consisting of two planetary stages and one helical stage is modeled. Hence, the kinematics of the epicyclic gear is known and the loads on the gears, bearings and shafts can be calculated automatically. Using these loads, the lifetime of the mechanical elements is automatically calculated using KISSsoft in the background.

It is hence sufficient to define the speed and torque acting on the input shaft in order to analyze all mechanical elements present.

The lifetime analysis of the elements is performed in KISSsys using a damage accumulation algorithm. Calculating a damage equivalent load based on the load spectrum is not meaningful in this application since the slope of the S-N curve of the bearings and gears is not equal. The load spectra can either be defined manually or read from a text file.

Since all analysis parameters and results are stored in variables, comprehensive reports can be generated.

3 Description of the model

3.1 Structure of the gearbox

The gearbox consists of two planetary stages and one helical stage.

Technical data of the gearbox (approximate):

Power rating	4MW
Nominal torque	2500kNm
Nominal speed	15Upm
Reduction	1:120
Diameter	2000mm
Length	2800mm
Lifetime required	20years

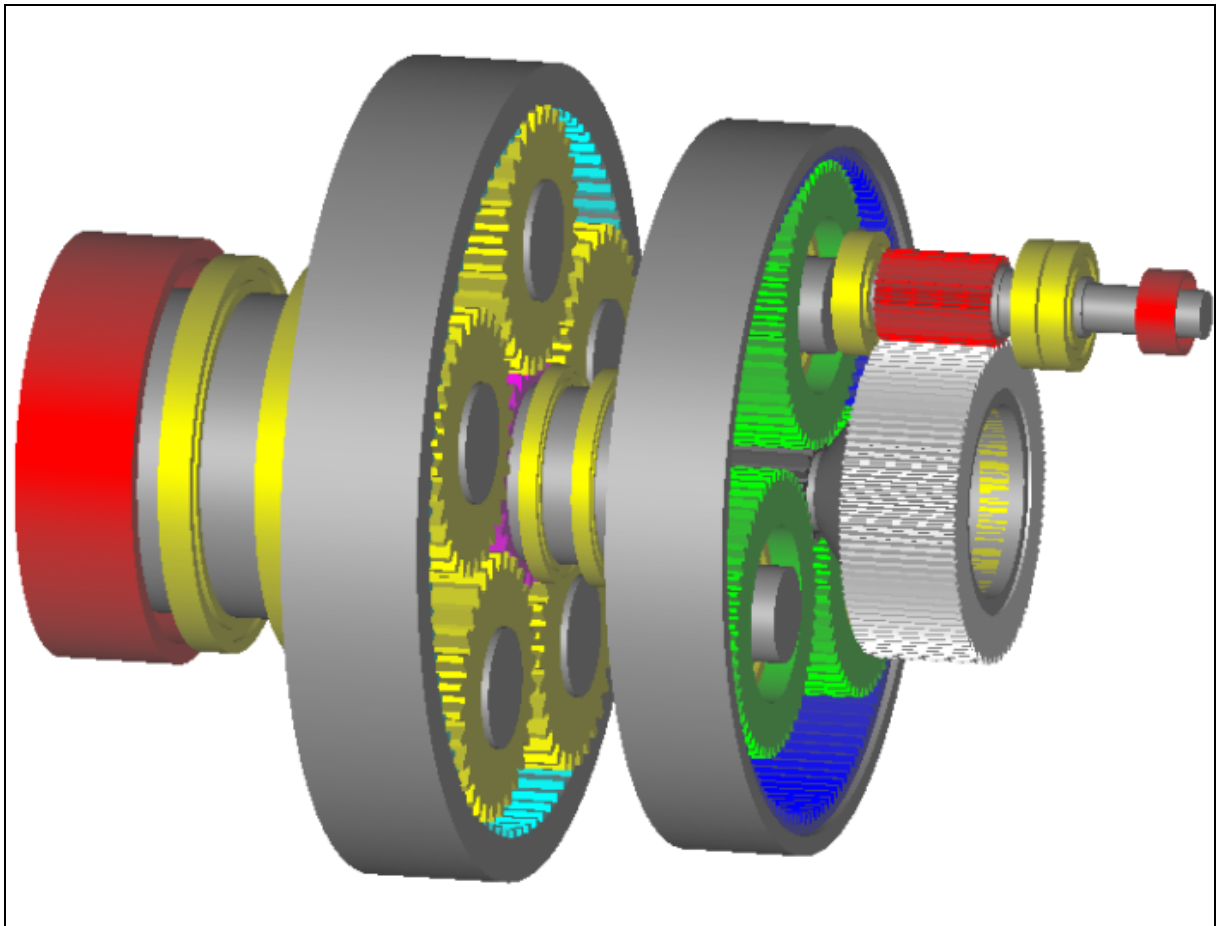


Figure 3.1-1 Structure of the gearbox: Black: shafts, blue: gears, Yellow: bearings, red: couplings. Note that only one planet per stage is shown.

The two rings of the planetary stages are fixed, the first stage has five, the second three planets (only one planet is shown in the 3D view). The constraint of the rings is modeled in KISSsys by setting the rotational speed to zero. The power flow of the gear box is shown below:

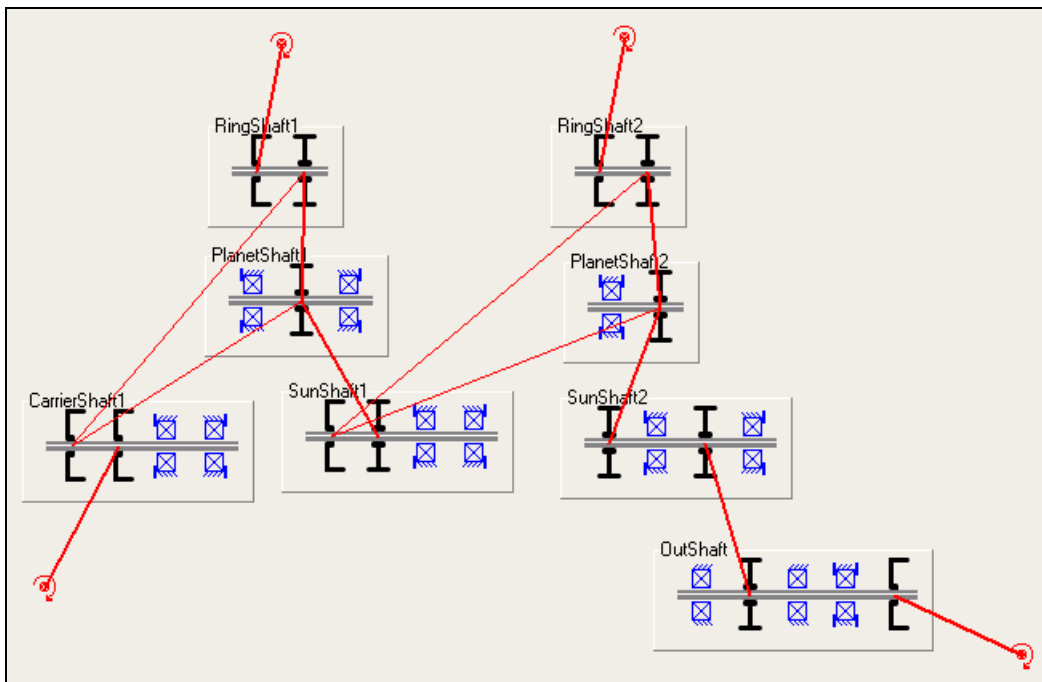


Figure 3.1-2 Schematic of the gearbox with power flow (or forces) (red), mechanical elements (shafts, gears, couplings, bearings) (black) and power input and output.

The three dimensional representation and schematic shown above are based on a tree structure shown below:

Figure 3.1-3 Tree structure of the gearbox consisting of several shafts, power input and output and connections. For the sun shaft of the second stage, the elements arranged on the shaft (bearings and gears) are shown.

3.2 Definition of forces acting

The user sets the nominal speed and torque on the input shaft (rotor hub) using the following dialog:

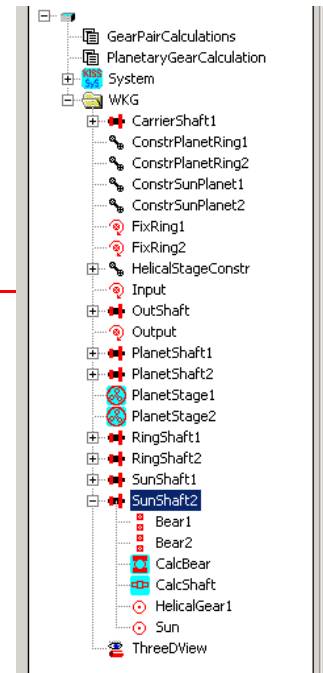
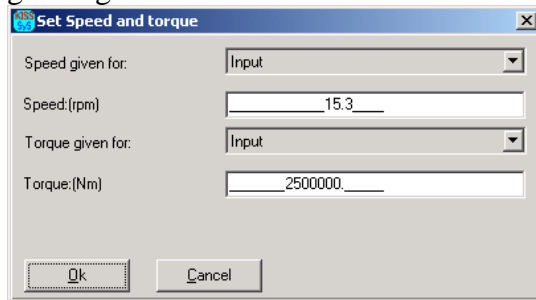


Figure 3.2-1 Dialog for definition of input (or output) speed and input (or output) torque

These nominal values can be modified using a load spectrum (frequency, torque and speed). The load spectrum can be defined manually or read from a text file. The spectrum again is selected / defined using the following dialog:

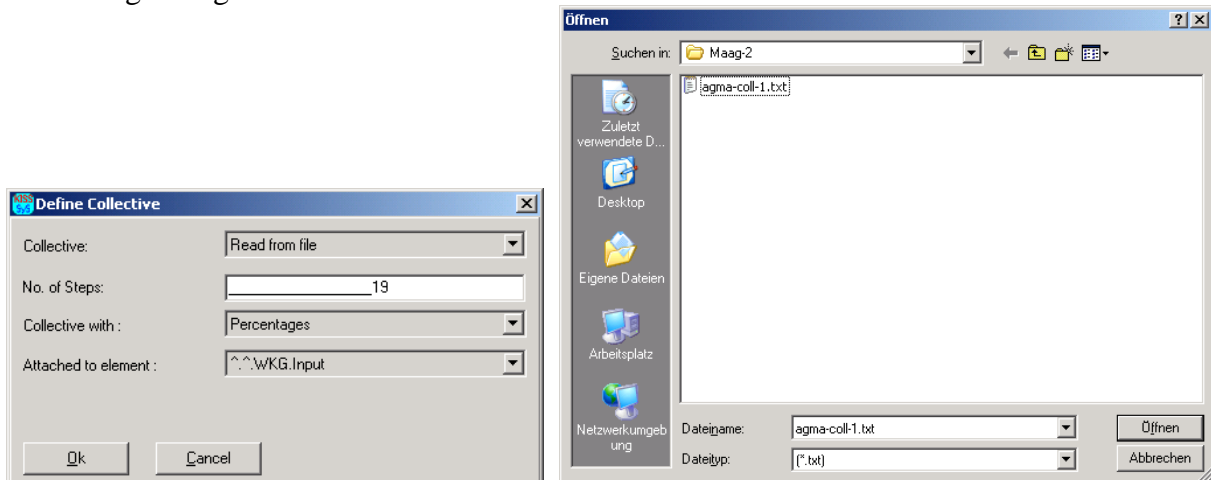


Figure 3.2-2 Selection of the applicable load spectrum through a KISSsys dialog and windows open file dialog

The load spectrum is then read and stored in KISSsys as a table:

Figure 3.2-3 Load spectrum read in KISSsys. The speed remains constant on the nominal value. The left column shows the frequencies (sum of all frequencies=1), middle column shows the relative torque

	Frequency	Torque	Speed
0	7.032e-007	1.4737	1
1	4.2109e-006	1.4474	1
2	4.0558e-006	1.4211	1
3	1.5691e-005	1.3947	1
4	2.8812e-005	1.3684	1
5	8.2002e-005	1.3421	1
6	0.00012854	1.3158	1
7	0.00046542	1.2895	1
8	0.00084218	1.2632	1
9	0.0024379	1.2368	1
10	0.007092	1.2105	1
11	0.011525	1.1842	1
12	0.015514	1.1579	1
13	0.048758	1.1316	1
14	0.082002	1.1053	1
15	0.12854	1.0789	1
16	0.22606	1.0526	1
17	0.27482	1.0263	1
18	0.20168	1	1

Furthermore, the user can define lubrication parameters such as lubrication mode, lubricant and lubricant temperature:

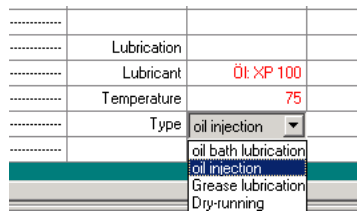


Figure 3.2-4 Selection of lubricant, temperature and lubrication mode

3.3 Execution of analysis, summary of results

In the user interface (see below) after having defined the loads, spectrum and lubrication, the user can start the lifetime calculation on calling the function „Calc. Lifetime“.

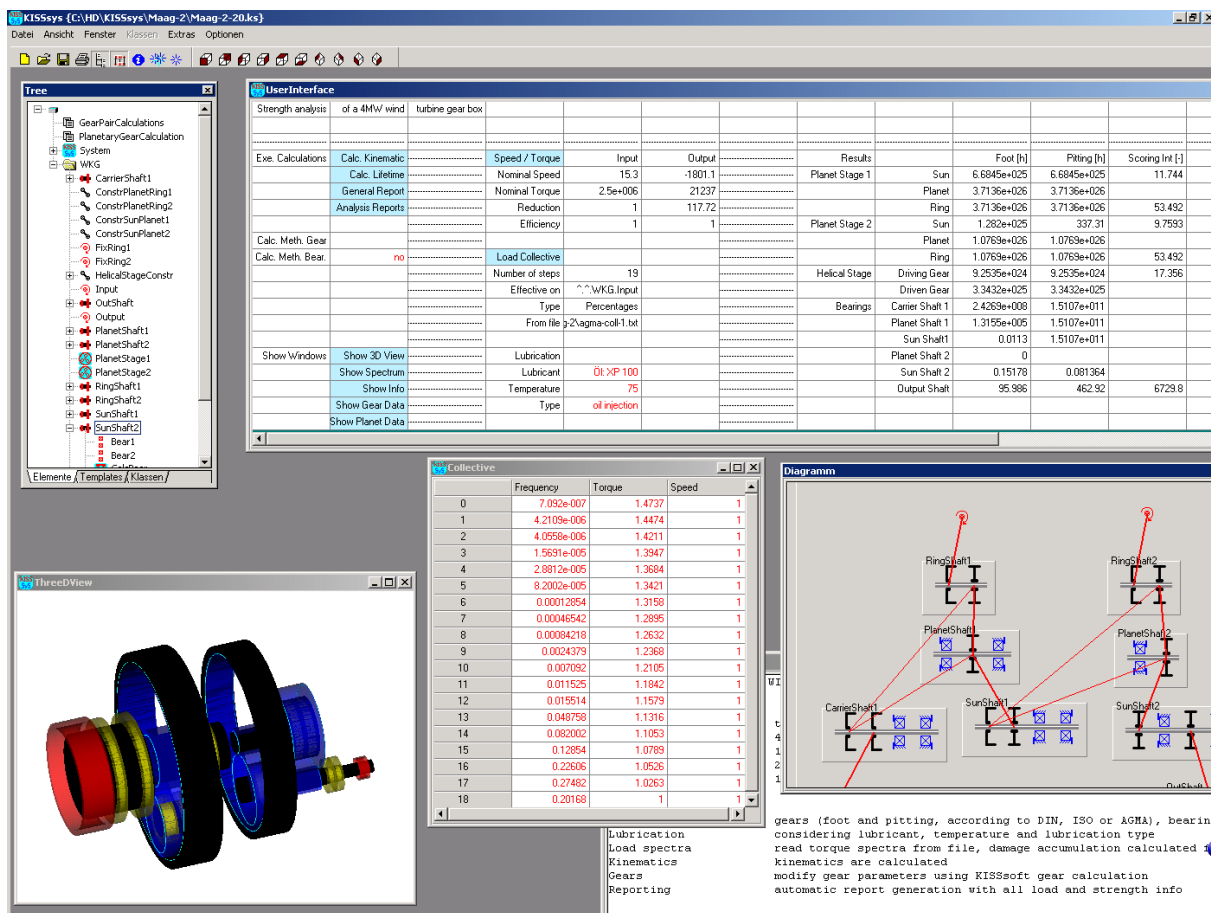


Figure 3.3-1 Overview of the KISSsys interfaces (user interface: large table in top right corner)

This function calculates for each step of the load spectrum a damage for all mechanical elements present using the respective S-N curves (which are all different for all parts). The damage is accumulated continually and being transformed into a lifetime at the end of the analysis. The analysis of the gears is according to DIN, ISO or AGMA standards, the analysis of the bearings according to standard L10 calculation or DIN/ISO 281.

The tooth geometries can be modified and optimized through calling the respective KISSsoft interfaces.

Analysis of a gearbox with modified tooth data takes a few seconds only.

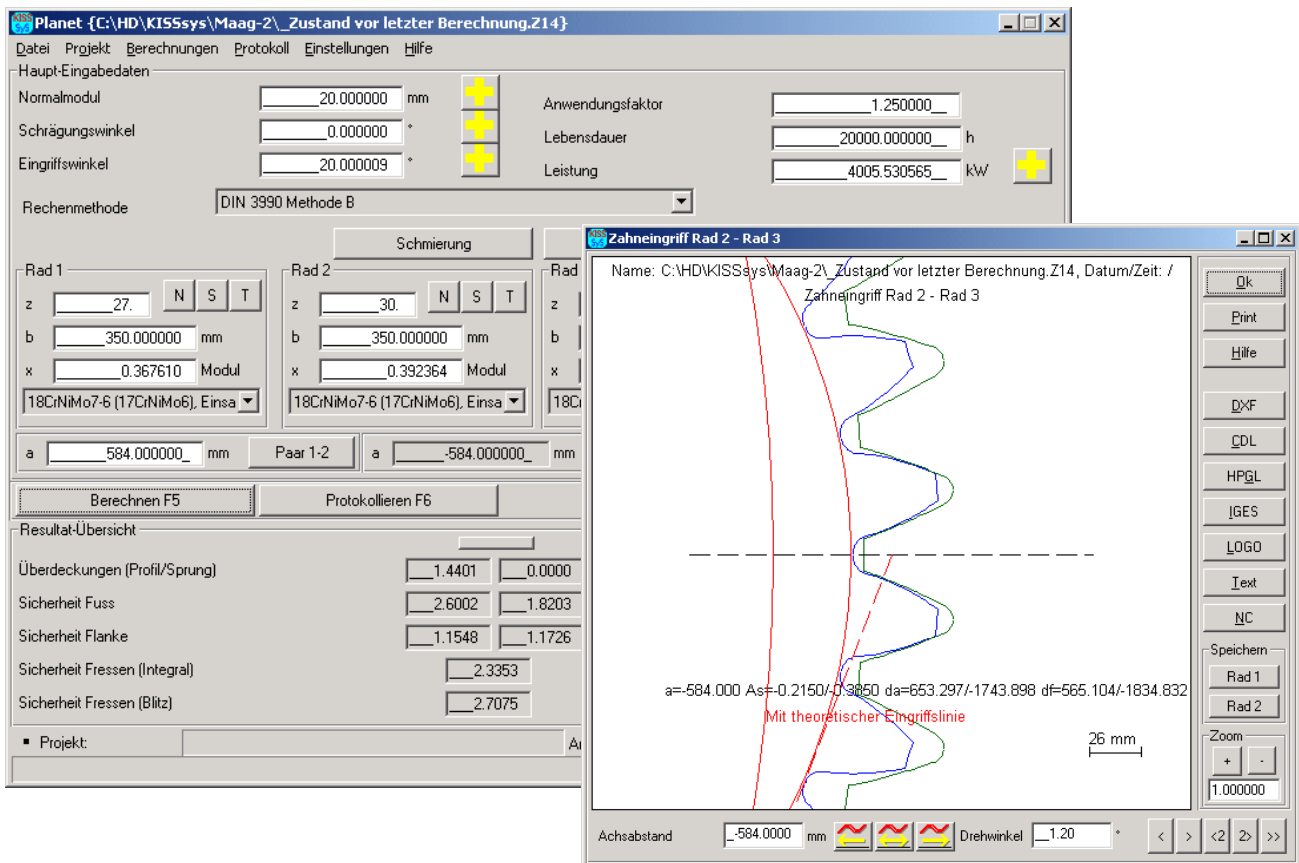


Figure 3.3-2 Optimization of tooth form using KISSsoft, automatic exchange of data between KISSsoft and KISSsys ensures that the analysis of the gearbox uses the correct tooth geometry.

4 Conclusion

The great advantage of this systematic approach to gearbox analysis is the gain in time when comparing different gear designs or when performing parametric studies.

While the modeling of such a system does require time and knowledge of the software, the time saved afterwards when optimizing the gears and for reporting is great, especially if the analysis is to be performed for load collectives with a high number of steps.

The reporting functions delivering comprehensive reports for all members analyzed helps a lot when documenting the analysis and the design of the gearbox respectively. The not so well liked work of writing analysis reports is accelerated considerably.

The model shown above can be extended to other epicyclic gears. A possible next step would be to introduce additional forces and bending moments on the input shaft, greatly affecting the bearings lifetime.