Fatigue Life Calculation using winLIFE

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Summary:

- The Software winLIFE, the underlying theory and the most important possibilities are described. A calculation example of a wind farm shows how the software can be used by the user.

1   Introduction

Light weight construction is becoming more and more important and the construction is often limited by fatigue life. winLIFE provides the well known and proved methods such as the Nominal Stress Method, Elastic Stress Method and Local Strain Approach in a user friendly interface to save the user having to carry out complex mathematical operations.

In some industries technical standards for fatigue life design exist (wind-turbines (GL), ship-construction (GL), and civil engineering (FKM)) and the user has to follow these rules to get a certification.
These standards represent the experience of decades and it is recommended to follow these rules. Because of the complexity a software-assist is desired which winLIFE provides.

The application of Finite Element Analysis is commonly used and in many cases indisputable. As a result winLIFE is designed to be connected to finite element and multi body programs running under Windows versions from XP until Windows 7. To avoid any interface problems, the data transfer is done by data-import and data-export files, which are described in detail. Interfaces for FEMAP, SAMCEF are shipped with winLIFE.

Resulting from the powerful macro capabilities of ANSYS Classic many users of ANSYS established their individual data interface to winLIFE without problems. SAMCEF included the data transfer in the native code.

winLIFE is an open program, the data structure is documented to enable the user to access it for his own use or to implement additional features.

Here a short overview is given and an example for fatigue calculation of wind turbines is included.

2   Overview winLIFE

winLIFE deals with the following capabilities and captures a wide range of fatigue life calculation possibilities:

2.1   Calculation Methods for proportional cases:

If there is no change of the direction of principal stresses (proportional case) the analysis can be performed with high calculation speed. The following procedures are available:

    Nominal stress method (HCF and VHCF)
-   Elastic stress method (HCF, VHCF)
    Local Strain Approach (LCF)

2.2   Calculation Methods for multiaxial cases (Critical Plane Approach)

In complex structures such as automotive bodies or gearbox housings there is usually a multiaxial problem. In this case the critical plane approach is used which leads to very time consuming calculations. For each (surface) node of interest in (typically 10…20) planes a fatigue calculation has to be done. In each plane (of the surface) the normal and shear stresses are calculated and a stress time history is created. For this stress time history a fatigue calculation is done and a damage D in each plane is ascertained. The plane with the maximum damage is the relevant plane. The procedure is shown in the following figure.
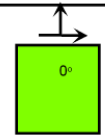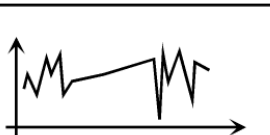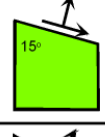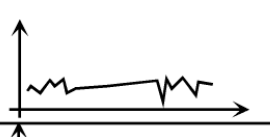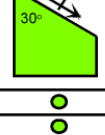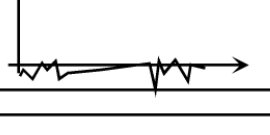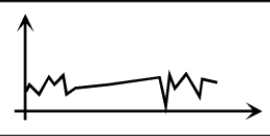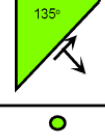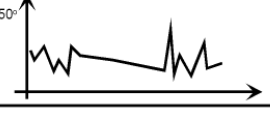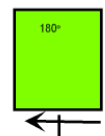
level	comparis.stress-time response	damage
0°		$D(0)$
15°		$D(15)$
30°		$D(30)$
⚬ ⚬ ⚬		
120°		$D(120)$
135° / 150°		$D(135)$
⚬ ⚬ ⚬		
180°		$D(180)$ $=D(0)$

figure 2-1: procedure of the critical plane approach

2.3 Special Modules

The following special modules are available:

- Seam weldings (Nominal stress, structural stresses, 1 concept)
- Gearwheel and Bearing

2.4 Creating of S-N / e-N curves according to the following standards

In many cases the fatigue life curves, S-N curves or e-N curves are not known. The following generators for life curves are included:

- FKM
- GL for Wind turbines
- GL for Ship

- UML (Uniform Material Law)

2.5 Material databases which are included in the software

- FKM-Material database
- Strain life data

2.6 Statistical Analysis

Life curves are mostly given for one failure probability so winLIFE deals with:

- Transforming S-N curves to different failure probabilities
- Calculating the surviving probability under statistical aspects

2.7 Extrapolation and Superposition

In the case of more load cases it is possible to superimpose and/or to extrapolate the results.

3 Integration of Finite Element Analysis and Multi Body Systems

3.1 Superimposing static unit load cases

A very important procedure is to calculate the stress history of a structure by superimposing unit load cases and (measured) load histories according to the following equation:

$$S_k(t) = \sum_i \left\{ S_{ki,e} * \frac{F_i(t)}{F_{Fe,i}} \right\} \quad \begin{array}{l} k=\text{nodenumber} \\ i=\text{load} \\ e=\text{unit load case} \end{array}$$

The results of unit load cases are got from a static calculation. A simple example shows the procedure. The body is loaded by three time-dependent loadings $F_1(t)$, $F_2(t)$, $F_3(t)$ These three loadings change their amount but not their direction. Now a unit load has to be defined on the line of action for each individual load. A static FE calculation is then calculated under these unit loads $F_{E1}(t)$, $F_{E2}(t)$, $F_{E3}(t)$. By scaling according to the equation above, you obtain the stress tensor as a function of the time.

In many cases, for example when there are bearings (contact), this simple procedure no longer works because in the bearing itself it is only possible to transmit pressure but not tensile forces. The solution for a problem like this can be found by replacing one unit load case with 2 unit load cases - in each case a positive and a negative share. Accordingly, a load will also be split into two, one positive and one negative. In this way the contacts or other non-linear effects will be taken into account correctly.
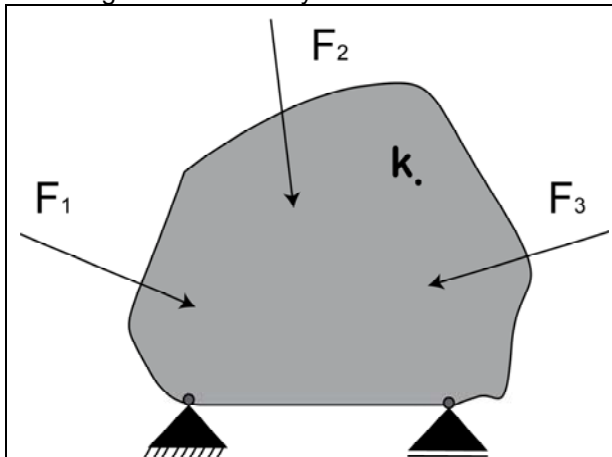

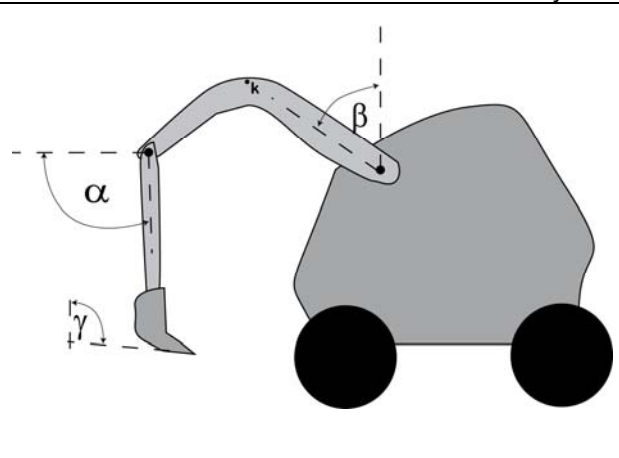| figure 3-1: solid body loaded by time depending loadings which do not change their action line | figure 3-2: multi body structure whose geometry is changing largely by moving the crane arm elements |

The rotation of components e.g. shafts, can also be solved by replacing it with a sequence of static superposition.

The following diagram shows a rotating shaft in a fixed coordinate system (above) and in a rotating coordinate system (below). You can therefore consider the shaft as stationary and convert it by superposition of a sequence of four unit load cases in an equivalent loading.
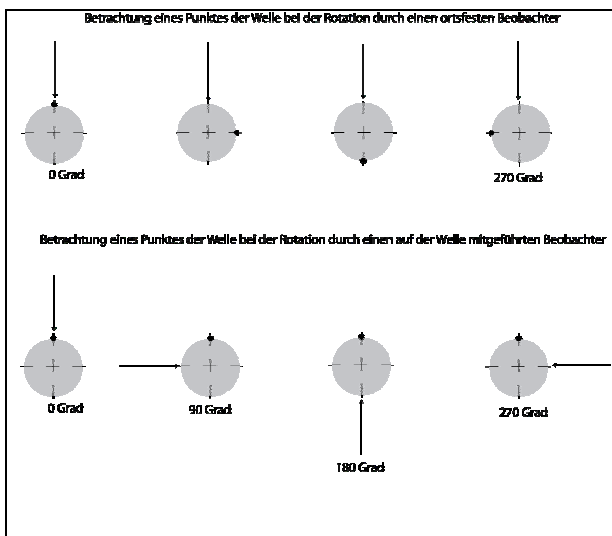

| figure 3-3: rotating wheel in a fixed coordinate system (above) and a rotating coordinate system (below) | figure 3-4: stress versus rotation angle for reality (above) and approximation (below) |
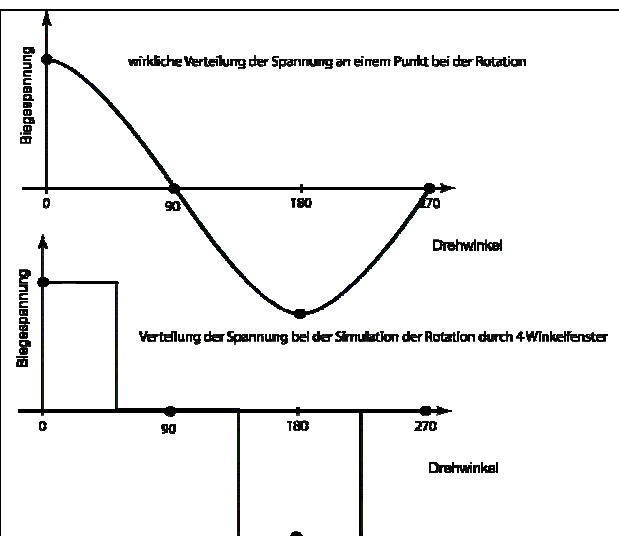
The following shows the result for the stress over the angle of rotation. Since the reversal point is decisive for the fatigue life, the same fatigue life is achieved despite the error due to the approximate angle division of only 4 areas for one rotation. Normally we select 30° rather than the 90° shown here in this example.

This load splitting is done in winLIFE automatically.

The fatigue life calculation for multiaxial problems occurs according to the process of the critical cutting plane. In the majority of cases, the crack initiates from the surface where, generally, there are plane stresses. Therefore the number of nodes to be examined is reduced considerably.

3.2   Export from an MBS-Analysis

In the case of a Multi-Body-System-Simulation, the whole non-linear problem is solved in the Multi-Body-System. A solution with the aid of static superposition would be very complex since many unit load cases would have to be defined.

In the model shown in the next diagram, the angles Alfa, beta and gamma change as a function of time. For the structure points of interest the stress tensors can be calculated straight away as a function of time and transferred to the fatigue life calculation.

Such an interface to winLIFE is provided in the Software SAMCEF. All the stress tensors representing the dynamic behaviour are exported to winLIFE and are immediately available for a fatigue life calculation. This process is used in particular in the field of wind energy with the special SAMCEF-Tool S4WT (SAMCEF for Wind turbines).

4   Interfaces for FEA and MB-Systems

Reliable fatigue life calculations can be carried out based on the results of the FEA and MBS analyses, winLIFE is used as a post processor and picks up the results of such calculations. The results of the FEA/MBS-calculation are described in a file to which winLIFE then has access. The results of the fatigue life calculation are then transferred to the pre-processor and shown there as ISO-lines.
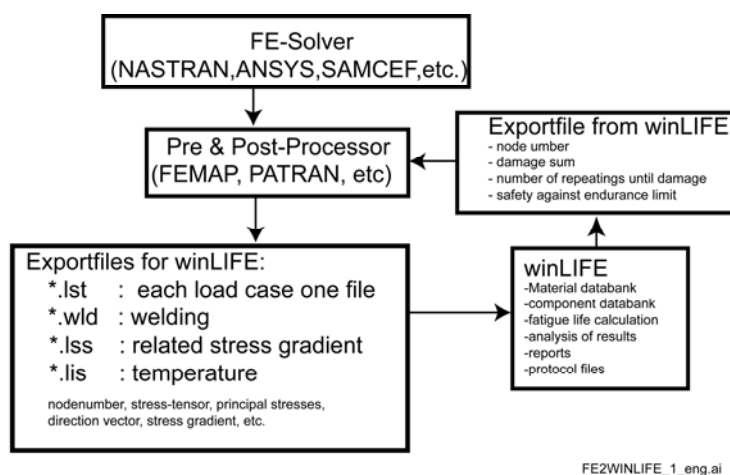

figure 4-1: Data communication between FEA / MBS

5   Example

The example shows the fatigue life calculation of a transmission case and real multiaxial loadings received from measurement readings. The static superposition and scaling method was used.

In this transmission, the drive train support is an integrated system. The load is initiated at the rotor hub. Via the stub shaft which is stored with two bearings in the flange casing, the bearing reactions influence the flange casing. The torque is initiated via the ring gear. The flange casing is supported vertically and horizontally at its torque reaction bar. The flange takes on the axial forces. In the stand-by system the hub loadings are converted to the bearing loads in the casing using equilibrium conditions. Due to several contact problems, each unit load case has been defined separately in a positive and negative direction. The appropriate loads have also been divided into two loads, one only in positive and the other only in negative direction. In this way the different behavior by pull and push has been taken into account simply and correctly.

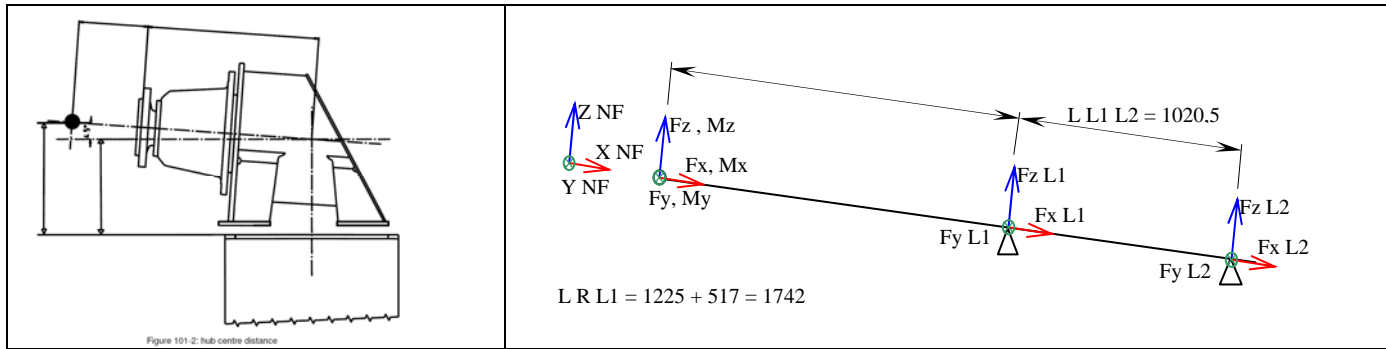## 5.1 Load conversion in the static stand-by system

figure 5-1 : static stand-by system with load initiation in the rotor hub and the bearings

F_x, F_y, F_z	Load forces in the rotor hub
M_x, M_y, M_z	Load moments in the rotor hub
F_{xL1}, F_{yL1}, F_{zL1}	Restoring forces in the ball joint bearing
F_{xL2}, F_{yL2}, F_{zL2}	Restoring forces in the cylinder roller bearing, where $F_{xL2}=0$
L_{RL1}	Distance rotor hub to middle of ball joint bearing =1742mm
L_{L1L2}	Distance middle of ball joint bearing to middle of cylinder roller bearing =1020,5mm

Table 1: Formula signs for the load conversion

The calculation of the individual bearing forces is carried out according to the following formulae:

$$F_{xL1} = F_x$$

$$F_{yL1} = -\frac{M_z - F_y(L_{RL1} + L_{L1L2})}{L_{L1L2}}$$

$$F_{zL1} = -\frac{M_y - F_z(L_{RL1} + L_{L1L2})}{L_{L1L2}}$$

$$F_{xL2} = 0$$

$$F_{yL2} = F_y - F_{yL1}$$

$$F_{zL2} = F_z - F_{zL1}$$

The torque M_x is supported by the ring gear.

5.2 Geometry Processing

- Remove all flange casing drillings which are not necessary for the calculation.
- Separate the component into flange casing and flange casing body to obtain a better cross linking (see section **Fehler! Verweisquelle konnte nicht gefunden werden.**).
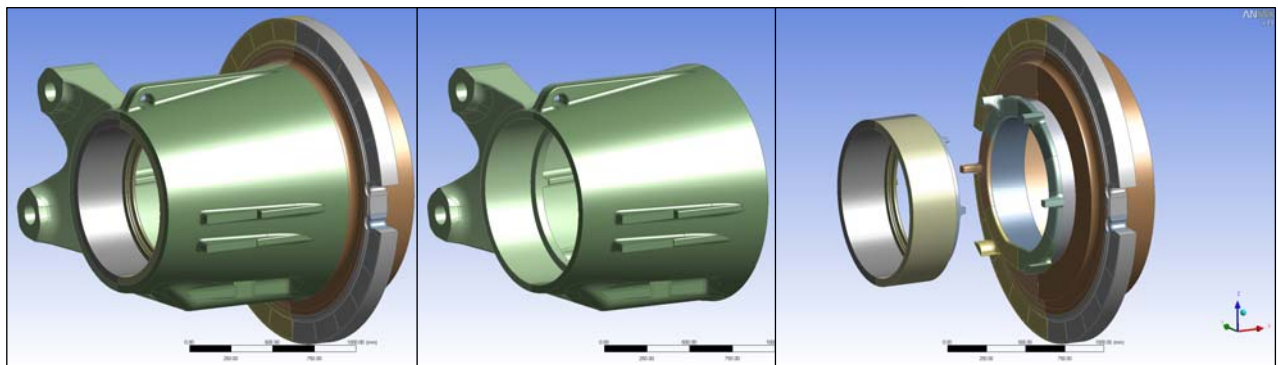

figure 5-2: flange casing – whole (left); flange casing – body (middle); flange casing (right)
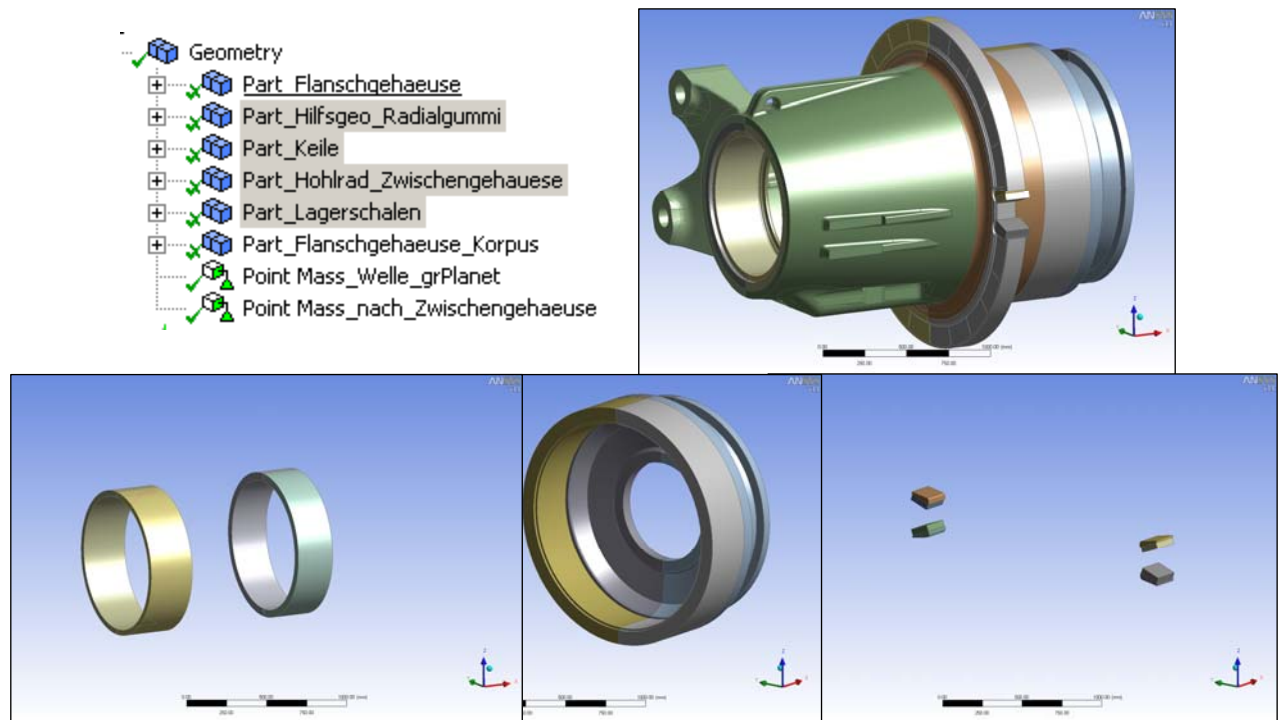
- Create additional geometries

figure 5-3: structure tree (top left); whole model (top right); bearing shell of ball-joint and cylinder roller bearing (bottom left); ring gear and intermediate casing (bottom middle); radial rubbers and wedges (bottom right)

5.3   Cross linking

For a reliable strength calculation it is necessary to have a good stress convergence in the notches. For this reason it is preferable to have hexahedron elements in such places. This is made possible if the flange casing is divided into several volumes (solids) which are then put together again in a component group (part). However, in ANSYS the cross linking methods Hexdominant and Tetrahedrons cannot be used on volumes in the same component group. For this reason it is better to split the casing into two component groups, the tetrahedron cross linked body and the predominantly hexahedron cross linked flange casing. Both these component groups can be joined back into one component with bonded Contacts.
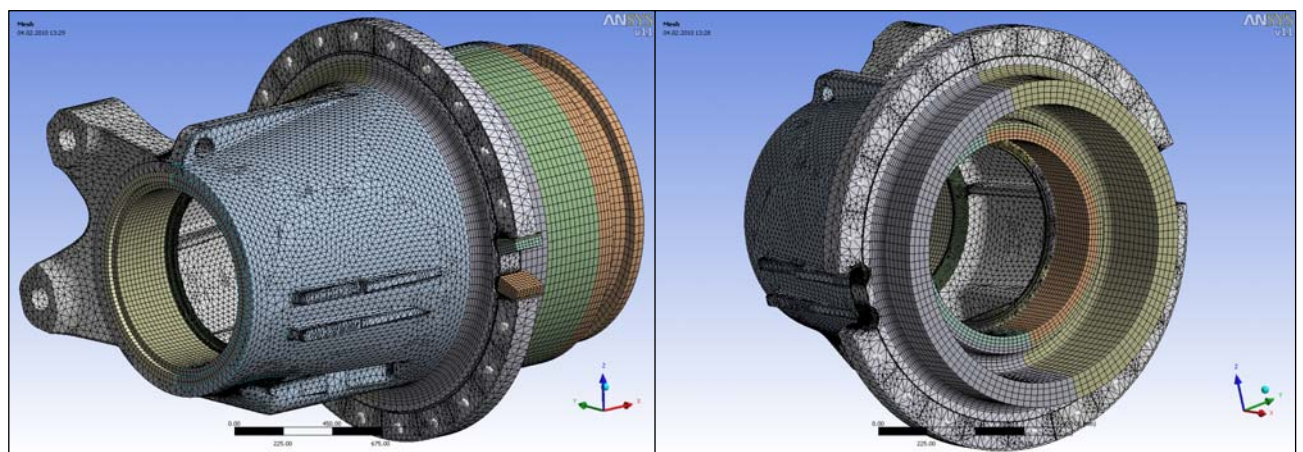

figure 5-4: Cross linking the whole model (left) and the flange casing (right)
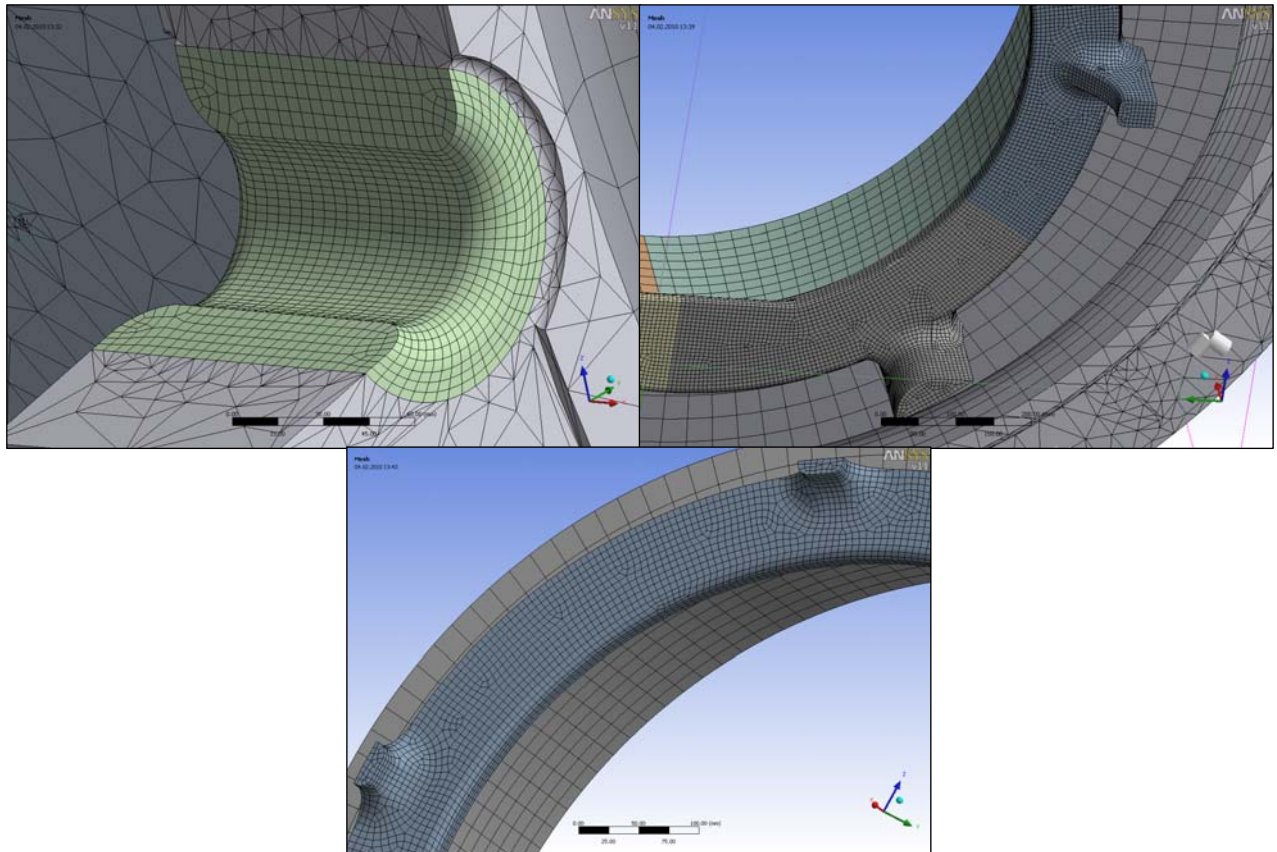
figure 5-5: Hexahedron netting: radial moment support (top left); end of support ribs to cylinder rolling bearing (top right); end of support ribs to ball joint bearing (bottom)

5.4   Boundary conditions and loads

In the boundary conditions we differentiate between the extreme load cases and the unit load cases. For the unit load cases we presume that in normal use the torque supporter does not rise at any time. Therefore an elastic support with the stiffness of the radial rubber is used in the moment support.

- *Elastic Support_Radialgummis*: Only for unit load cases. The radial rubbers are pre-stressed and therefore prevented from twisting.
- *Elastic Support_Axialgummis*: Die axial rubbers are pre-stressed with circle segments and therefore an axial displacement is prevented.
- *Federn*: In order to show the shear stiffness of the axial rubbers, springs were used with the appropriate thrust stiffness for each circle segment.
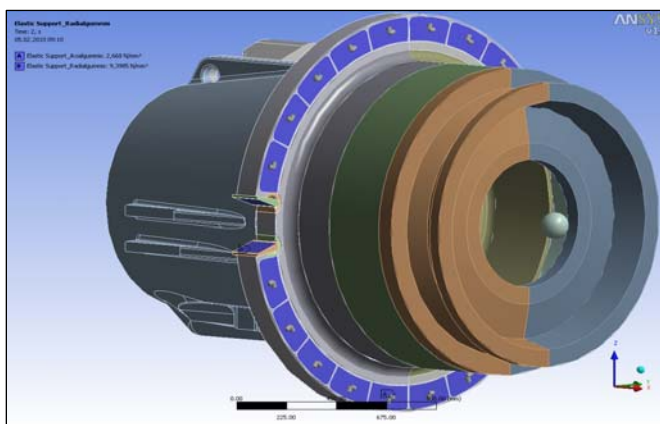

figure 5-6: unit load case with elastic support

The individual loadings are either activated or suppressed, according to the calculation case. Therefore only one load is active per unit load case, except for *Bearing Load_RotorLock1 a*nd *Bearing Load_RotorLock2*.

For the extreme loads all loads except the two rotor locks are loaded.
For the rotor lock loads, no torque is induced.

- *Standard Earth Gravity*: Weight force on all modelled components including the additional masses „Point Mass_Welle_grPlanet" and „Point Mass_nach Zwischengehäuse".
- *Bearing Load_Lager_1*: Bearing reaction on the ball joint bearing in the Y and Z directions.
- *Bearing Load_Lager_2*: Bearing reaction on the cylinder rolling bearing in the Y and Z directions.
- *Force_Axialkraft_pos_Lager_1*: Positive axial force on the cylinder rolling bearing has an effect on the outer ring.
- *Force_Axialkraft_neg_Lager_1*: Negative axial force on the cylinder rolling bearing has an effect above the lid on the front of the flange casing.
- *Moment_Mx_Hohlrad*: Moment support above the ring gear.

figure 5-7: bearings loadings, moment loadings and weight force

6 Endurance Calculation

6.1 Procedure

For the endurance calculation, proceed as follows:
1. Calculation of the S-N curve including security for the damage calculation.
2. Conversion of the load time sequences Mx, My, Mz, Fx, Fy, Fz at the rotor hub to load time sequence with the bearings loads according to section 5.1.
3. Calculation of the unit stresses by putting the unit loads into the FE-Model.
4. Exporting the unit stresses in the form of the principal stress of the whole surface.
5. Damage calculation with winLIFE 3.2. The equivalent stress is calculated with the nominal stress hypothesis and the method of the critical cutting plane.
6. Importing the damages of all surface nodes for showing as a graph in ANSYS.

6.2 S-N Curves

6.2.1 Calculating S-N curves

The calculation of a S-N curve occurs according to [3].

The S-N curve calculated in this way is then adjusted downwards by the safety factor. In this way the safety aspects in the endurance calculation are taken into account. The total safety factor $\Sigma\gamma$ is calculated as follows:

$$\sum\gamma = \frac{S_{p\ddot{u}} * S_d}{\gamma_M * \gamma_{srf}}$$

The following diagram shows the areas in which the individual S-N curves are used:
- **Blue**: S-N curve whole model
- **red**: S-N curve flange radius
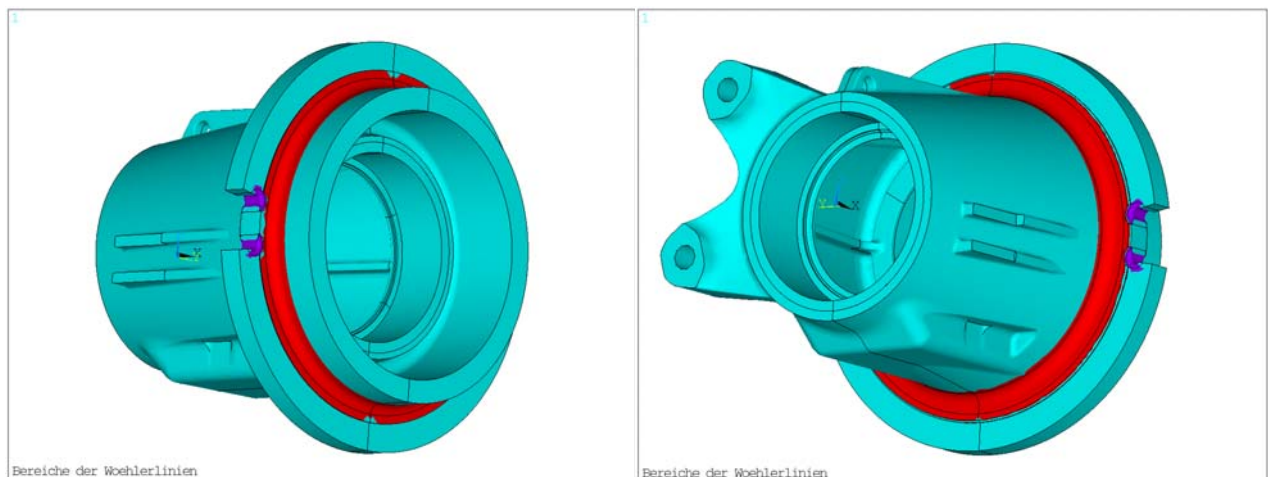- **purple**: S-N curve radius moment support

figure 6-1: Areas in which the individual S-N curves are used

In the following tables the values outlined in red are entry data for winLIFE.

6.3 Conversion of Hub Load to Bearing Load

The load time sequences in the specifications include loads, i.e. forces and moments on the hub. Based on the static stand-by system, each load in the load time sequences with hub load is transferred to a load time sequence with bearings loads. The formulae for converting the hub loads into bearings loads can be seen in section 5.1

6.4 Load Histories in winLIFE

There are up to 36 different load histories (scenarios on the WEA) which when multiplied by the frequency of occurrence sum up to a fatigue life of 20 years. The loads are always positive since the load direction is provided by the direction of the unit load.

In the load histories the diagrams are divided up as follows:
- Diagram 1 = Mx positive; Diagram 2 = Mx negative
- Diagram 3 = Fx Lager1 positive; Diagram 4 = Fx Lager1 negative
- Diagram 5 = Fy Lager1 positive; Diagram 6 = Fy Lager1 negative
- Diagram 7 = Fz Lager1 positive; Diagram 8 = Fz Lager1 negative
- Diagram 9 = Fy Lager2 positive; Diagram 10 = Fy Lager2 negative
- Diagram 11 = Fz Lager2 positive; Diagram 12 = Fz Lager2 negative
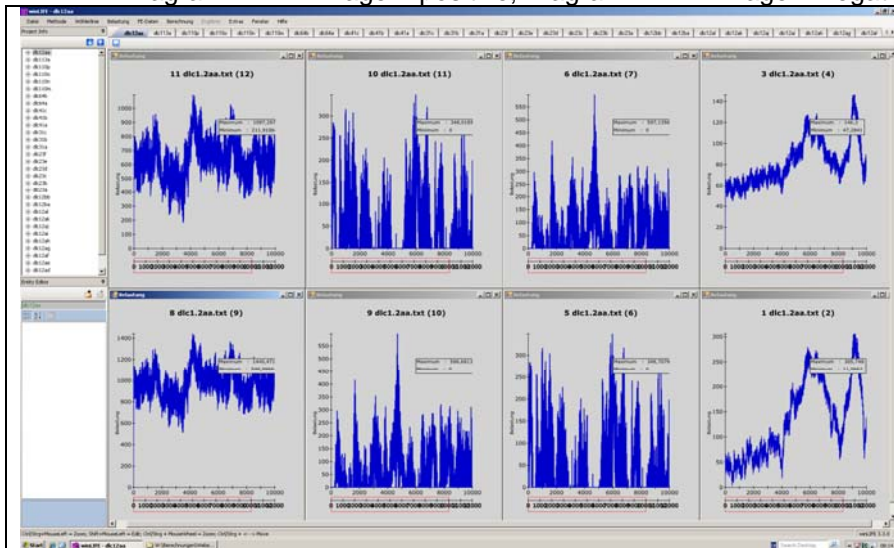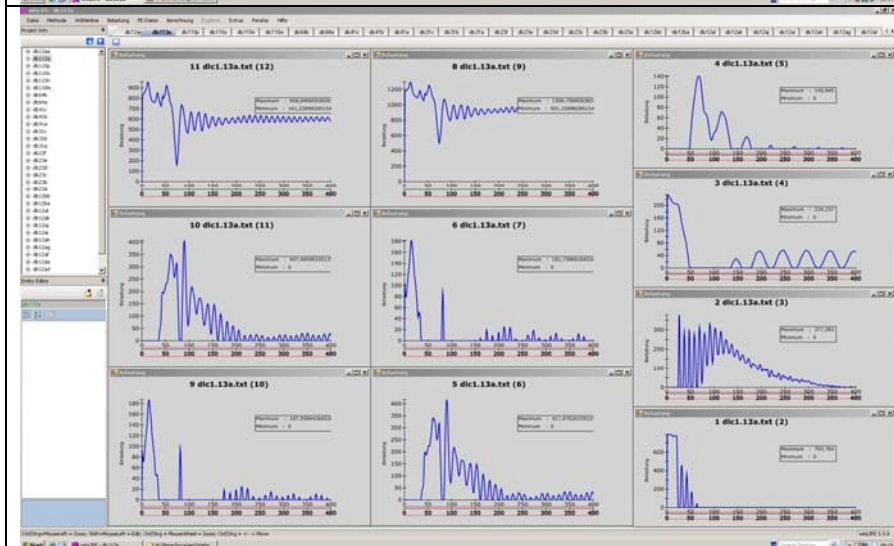

figure 6-2: load time histories
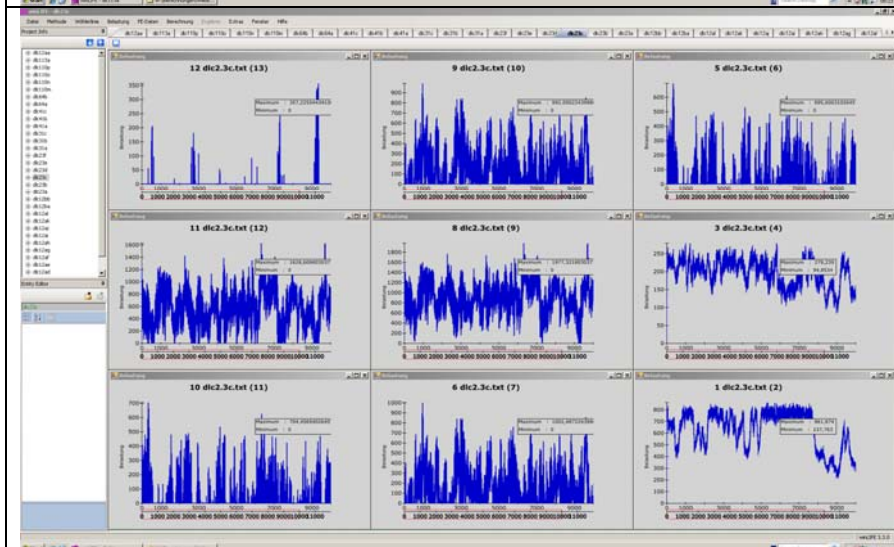

figure 6-3: load time histories
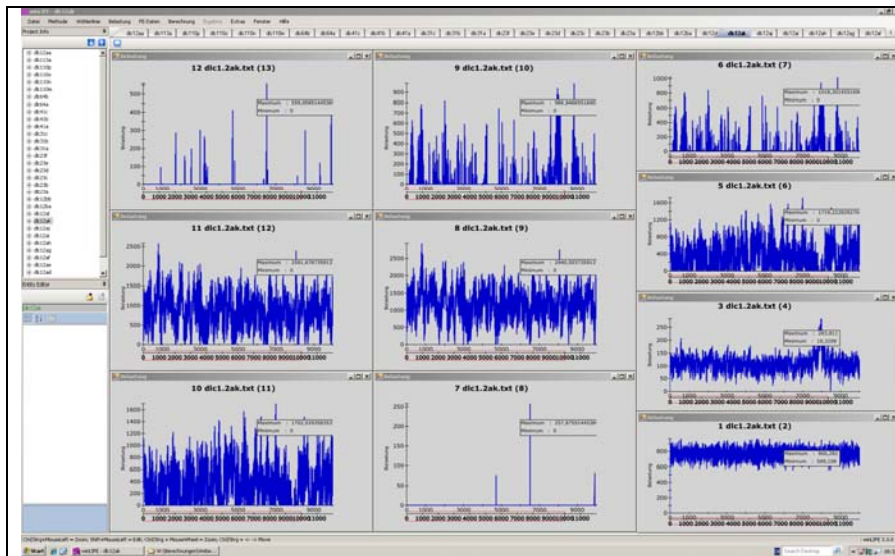

figure 6-4: load time histories

figure 6-5: load time histories
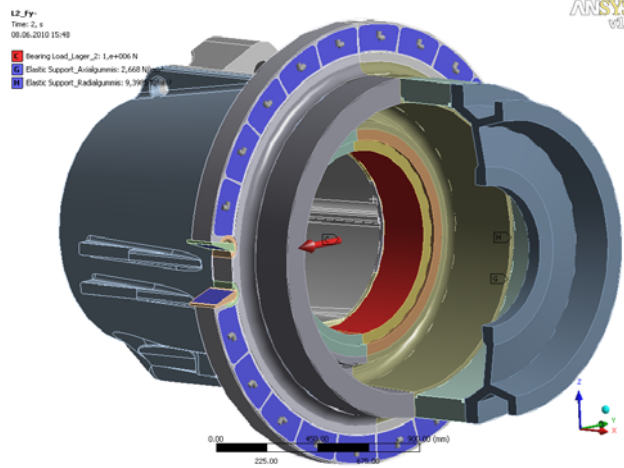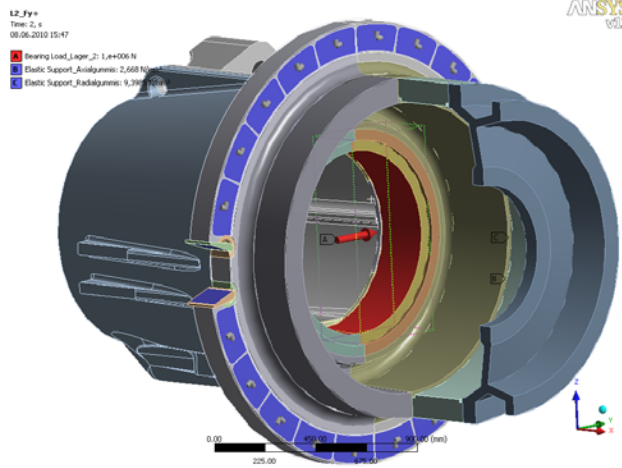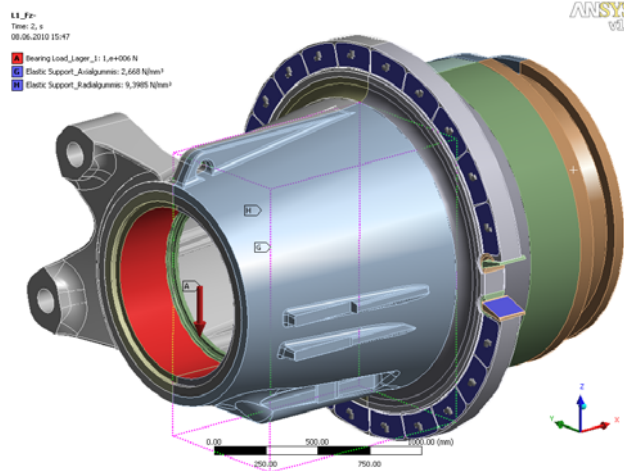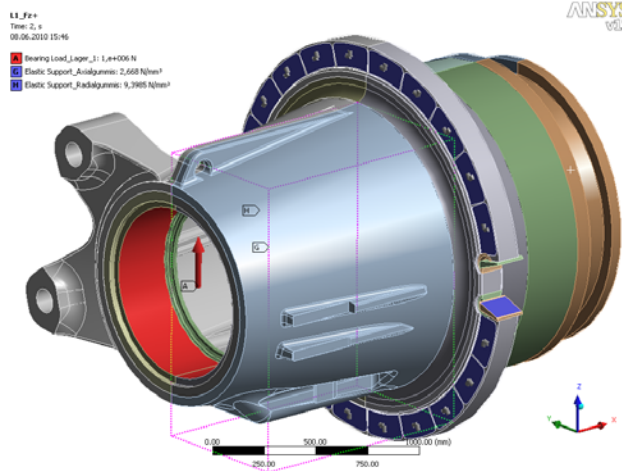
6.5 Unit Loads

6.5.1 Loads in the model

The moments of the unit load amount to 1000kNm and the forces to 1000kN, so that the calculation results from the FEM are sizes which are comprehensible.

Load case	Mx pos	Mx neg	Fx bearing 1 pos	Fx bearing 1 neg	Fy bearing 1 pos	Fy bearing 1 neg	Fz bearing 1 pos	Fz bearing 1 neg	Fy bearing 2 pos	Fy bearing 2 neg	Fz bearing 2 pos	Fz bearing 2 neg	Eigengew. shaft Fz neg
	Nmm	Nmm	N	N	N	N	N	N	N	N	N	N	N
1	1,E+09	0	0	0	0	0	0	0	0	0	0	0	0
2	0	-1,E+09	0	0	0	0	0	0	0	0	0	0	0
3	0	0	1,E+06	0	0	0	0	0	0	0	0	0	0
4	0	0	0	-1,E+06	0	0	0	0	0	0	0	0	0
5	0	0	0	0	1,E+06	0	0	0	0	0	0	0	0
6	0	0	0	0	0	-1,E+06	0	0	0	0	0	0	0
7	0	0	0	0	0	0	1,E+06	0	0	0	0	0	0
8	0	0	0	0	0	0	0	-1,E+06	0	0	0	0	0
9	0	0	0	0	0	0	0	0	1,E+06	0	0	0	0
10	0	0	0	0	0	0	0	0	0	-1,E+06	0	0	0
11	0	0	0	0	0	0	0	0	0	0	1,E+06	0	0
12	0	0	0	0	0	0	0	0	0	0	0	-1,E+06	0
13	0	0	0	0	0	0	0	0	0	0	0	0	1

Table 2: Unit loads

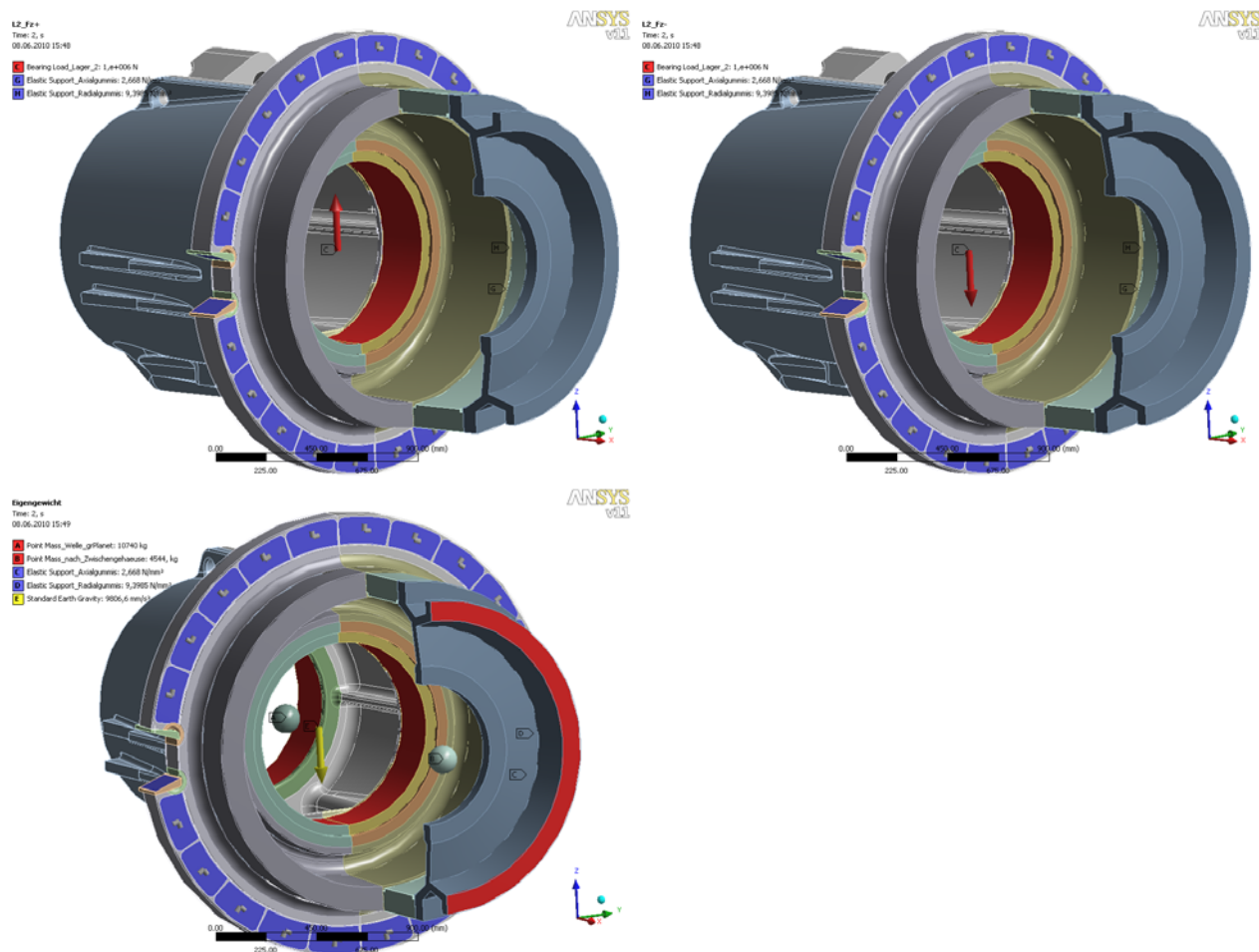In the following diagrams the unit loads are shown.
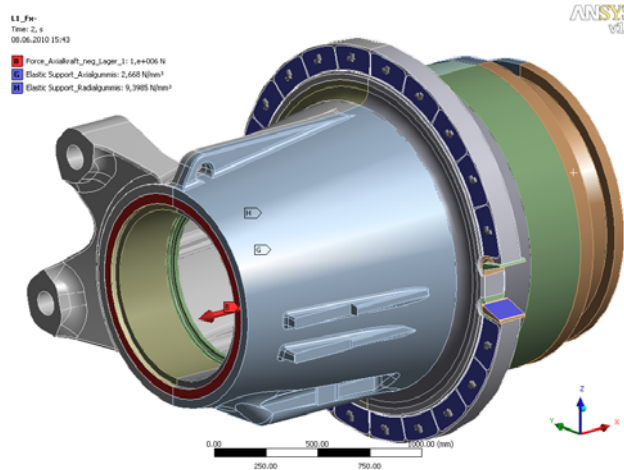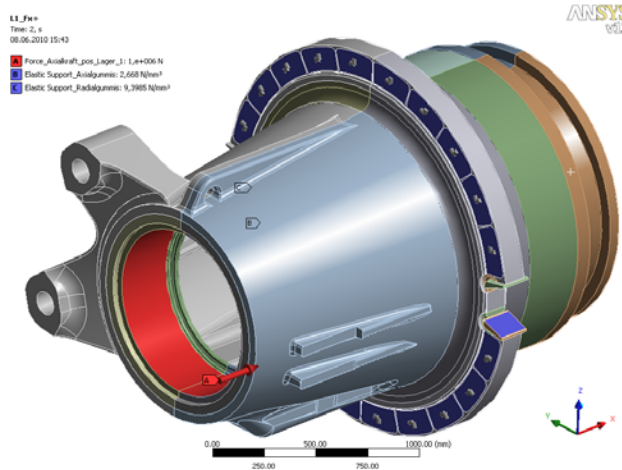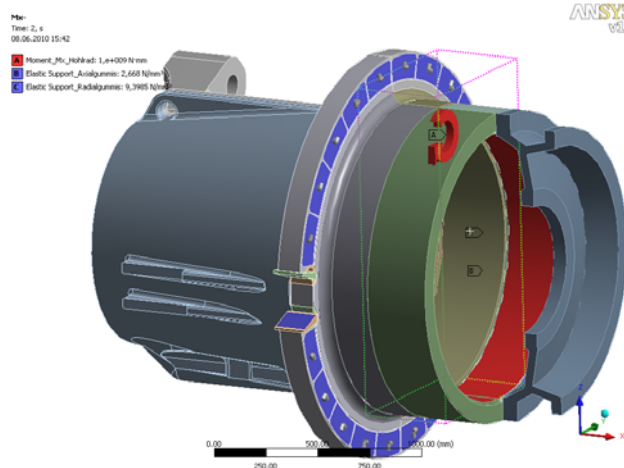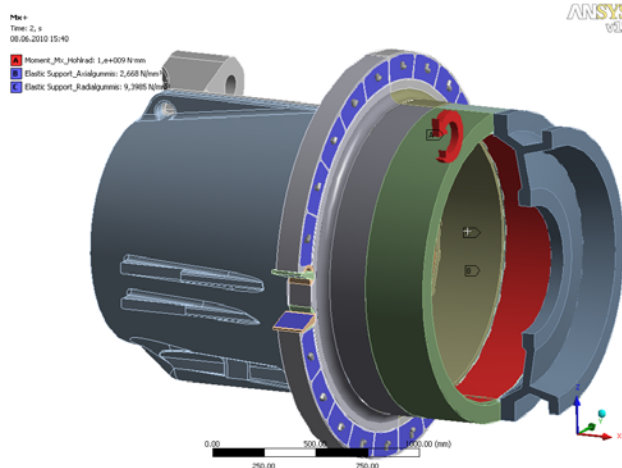
figure 6-6: unit loads (starting from top) Mx+, Mx-, Fx+, Fx-, Bearing 1 Fy+, Bearing 1 Fy, Bearing 1 Fz+, Bearing 1 Fz-, Bearing 2 Fy+, Bearing 2 Fy-, Bearing 2 Fz+, Bearing 2 Fz-, own weight

6.6   Endurance calculation

The endurance calculation has been carried out using the software winLIFE 3.2.
Beforehand the principal stresses of all unit load surface nodes were exported from ANSYS in a file with the extension ".lst".
The load time sequences referring to the rotor hub are provided. These are then converted to load time sequences according to the stand-by system which have an effect in the bearings.
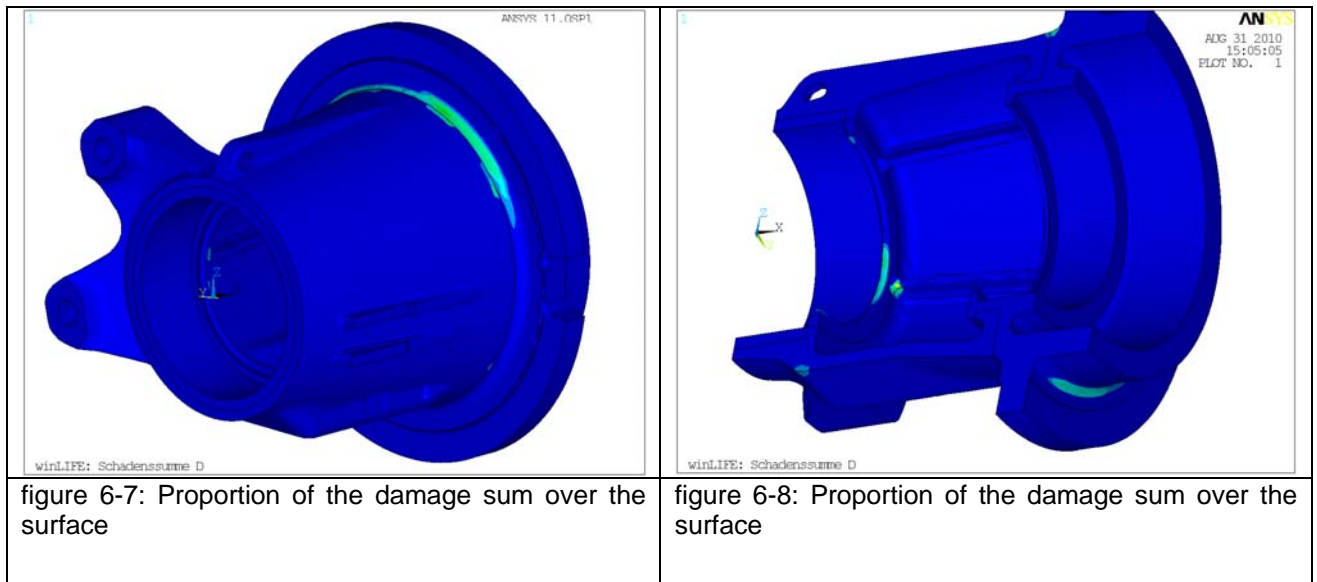In winLIFE the notch stress method is selected and a multiaxial loading is presumed. The S-N curve is provided based on the values from section **Fehler! Verweisquelle konnte nicht gefunden werden.**.
The safety factors are already included in the S-N curves. In the program the calculated load time sequences in the bearings are linked to the unit loads, taking into consideration their size. When calculating the endurance of cast materials the normal stress hypothesis is used. The mean stress influence is taken into account by the amplitude transformation. The damage calculation is carried out using the critical cutting plane method, looking for the maximum from 20 cutting planes. With the help of the rainflow count damage is calculated for each node and each cutting plane. 100 classes were selected for the rainflow classification.

winLIFE calculates the damage for each load time sequence separately. In order to obtain the total damage, the damage of each individual load time sequence is multiplied by the respective frequency of occurrence. The sum is then obtained from this. The total damage of the cutting plane with the largest damage of each individual node is shown in a file with the extension".sum". This is then imported to ANSYS and shown graphically.

The damage for the S-N curve valid locally in each case is shown (see section 6.2.1).

| figure 6-7: Proportion of the damage sum over the surface | figure 6-8: Proportion of the damage sum over the surface |

7   Future outlook

Fatigue life calculation is becoming more and more important and is already part of the development process in many industries. Especially in the field of wind turbines a dramatic increase of the power of the turbines leads to highly loaded components and the need for fatigue life optimisation.

The integration of fatigue calculation in the development process is beneficial for reducing the time of the development phase because different components can be compared reliably.

Despite all our enthusiasm, we have to keep in mind that a reliable quantitative fatigue life prediction is not possible by calculation alone and that parts critical to safety must always be tested additionally.

This means that the real benefit of fatigue life calculation will be achieved when it is performed over a long period of time and if the results are partially compared with test results. By doing this the user will be able to calculate a more reliable fatigue life prediction.

8   References

[1] Haibach, E., Berger, C., Hänel, B., Wirthgen, G., Zenner, H., Seeger, T.: Rechnerischer Festigkeitsnachweis für Maschinenbauteile, Heft Nr. 183-1, 1994, Forschungskuratorium Maschinenbau, Lyonerstr. 18, Frankfurt/M.

[2] FKM Richtlinie: Rechnerischer Festigkeitsnachweis für Maschinenbauteile, 4. Extended Edition 2002, Forschungskuratorium Maschinenbau, 1998

[3] Gudehus, Zenner: Leitfaden für eine Betriebsfestigkeitsrechnung, Empfehlung zur Lebensdauerabschätzung von Maschinenbauteilen. 3rd Edition, ISBN 3-514-00445-5, Publisher: Stahleisen, Düsseldorf.

[4] Guideline for the certification of wind turbines, Edition 2005, Germanischer Lloyd,

[5] Bäumel A.; Seeger, T.: Materials Data for Cyclic Loading, supplement 1, Elsevier Science Publishers B.V.1987, ISBN 0-444-88603 6

[6] Hobbacher, A.: Recommendations for fatigue design of welded joints and components, International Institute of welding, IIW document IIW-1823-07 December 2008