New Guideline for Determining the Reliability of Planetary/Spur Gear Units

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In the wind power industry, the reliability of powertrain components plays a major role.

Especially in multi-megawatt offshore applications, an unplanned replacement of drivetrain components can lead to extremely high costs.

Hence, the expectation of wind farm operators is to forecast the system reliability. Under the leadership of the VDMA (Mechanical Engineering Industry Association), the standardization paper 23904 "Reliability Assessment for Wind Turbines" was published in October 2019.

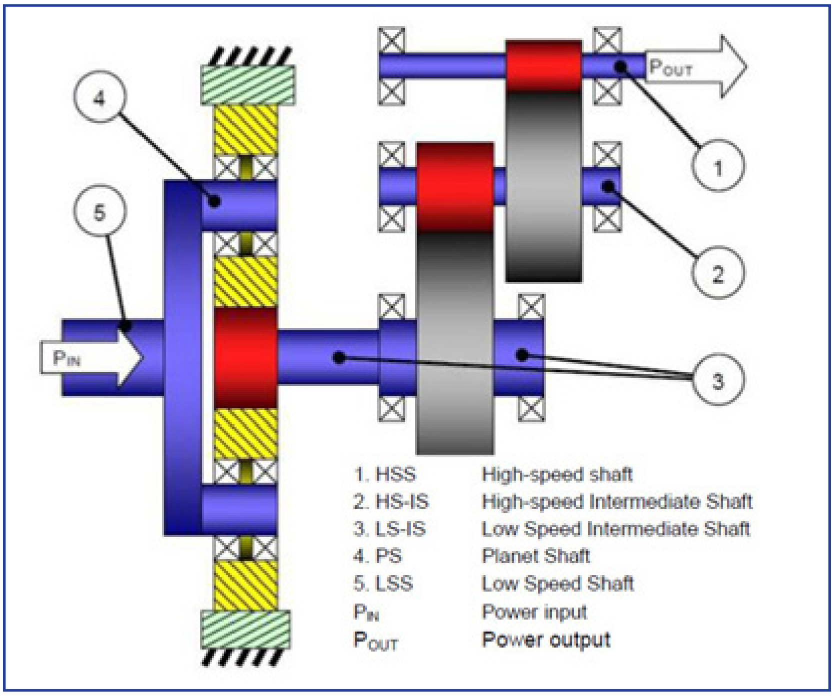

Figure 2 **Determination of the functional elements (Ref. 1).**

Up to now, wind gearboxes have been designed according to IEC 61400-4. This specifies minimum safety requirements for all relevant load-carrying components in the gear unit, which must be fulfilled for the various operating and extreme loads (Ref. 3). For example, the gear teeth are designed in accordance with ISO 6336-3 and ISO 6336-2, with minimum safety factors for the tooth root and flank load carrying capacity, and also the scuffing and micro-pitting load carrying capacity, in accordance with ISO/TS 6336-20 or ISO/TS 6336-21 and ISO/TS 6336-22. The shafts are designed according to DIN 743, bolted connections according to VDI 2230, and structural components are designed according to the FKM guidelines "Dimensioning of Machine Components Made of Steel and Cast Iron" and "Fracture Mechanics" (Ref. 4), specifying the boundary conditions for the calculation. What all calculation methods have in common is that they are based on a safety concept, i.e. — the permissible load is evaluated with the load that occurs in the form of a safety factor. Standardization paper 23904 provides a method for calculating the system reliability of gearboxes in wind turbines (Fig. 1). The method is essentially based on the principles of statistical determination of failure probability according to Bertsche (Ref. 5).

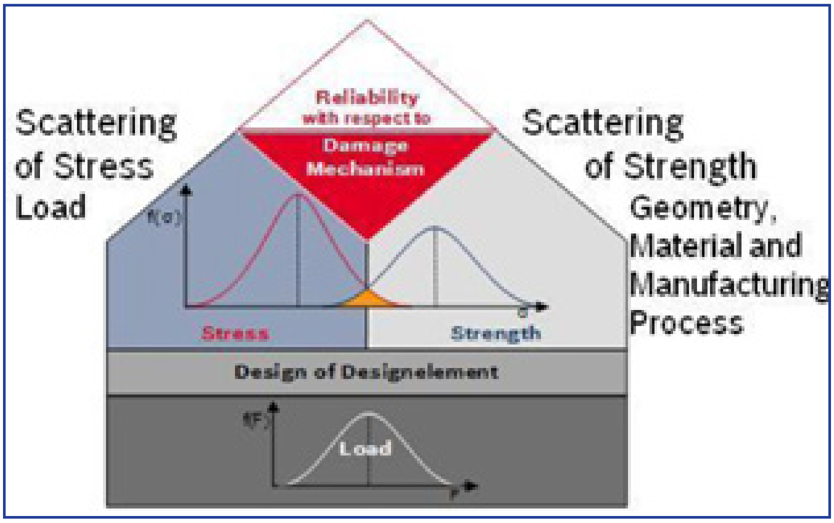

Figure 1 **System reliability.**

Theoretical calculation approaches are not available for all failure mechanisms occurring in real operation. The present method is limited to failure mechanisms for which a fatigue life can be described according to the recognized rules of technology. It is therefore possible to investigate parameters influencing reliability and to compare gear designs. An absolute forecast of the system reliability is not yet possible. For this purpose, calculation approaches for the failure mechanisms that have not been calculable thus far, or associated statistical distributions must be determined in the future.

The method first identifies the functional ealements that are relevant for the determination of system reliability (Fig. 2). Typically, these are the

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power transmitting components and supporting structures.

In the next step, the so-called system elements are determined based on failure mode effect analysis (FMEA); the system elements describe the failure mechanisms of the functional elements. For example, a gear wheel can fail due to a tooth root bending fatigue or pitting damage (Fig. 3).

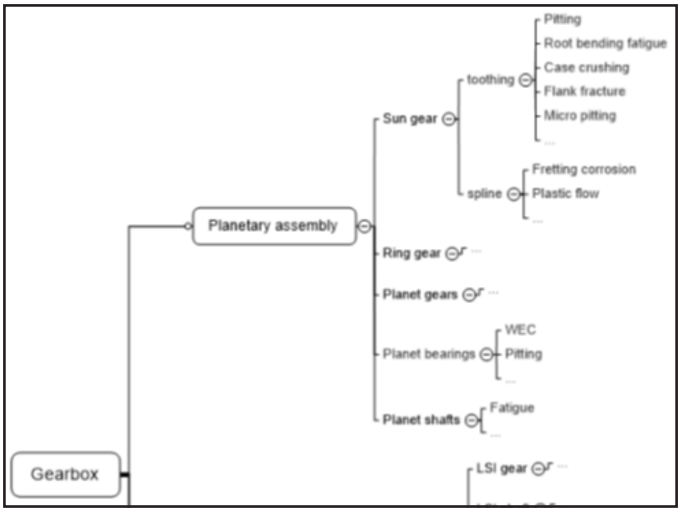


Figure 3 System elements.

The system elements are then classified, whereby system elements are classified as reliability relevant (A1, A2, B) and neutral (C) for the system under consideration (Fig. 4). A1 represents those elements for which calculation methods are available (e.g. — ISO 6336), while A2 refers to elements for which calculation methods are not available. Elements of category B are characterized by non-deterministic error distributions (e.g. — scuffing or smearing). Experience and experiments should therefore be used to predict the reliability of these elements. Category C elements are irrelevant to the reliability of the system and are therefore not considered in the calculations. The A1, and partly A2, system elements are considered in the present reliability calculation. The classification corresponds to the current state of the art and will be adjusted if a recognized calculation approach becomes available for an A2 element.

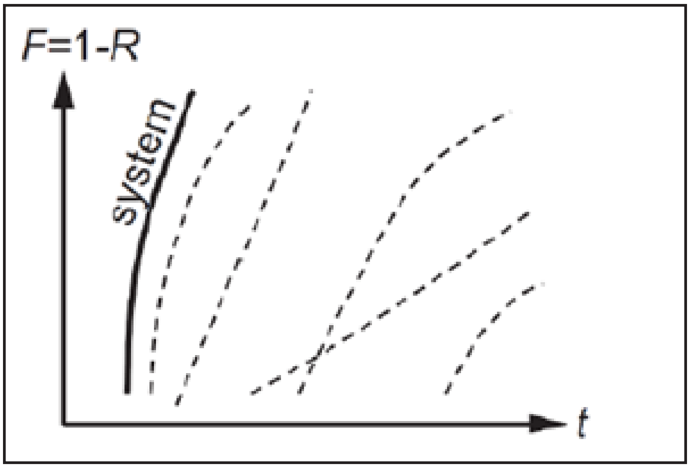
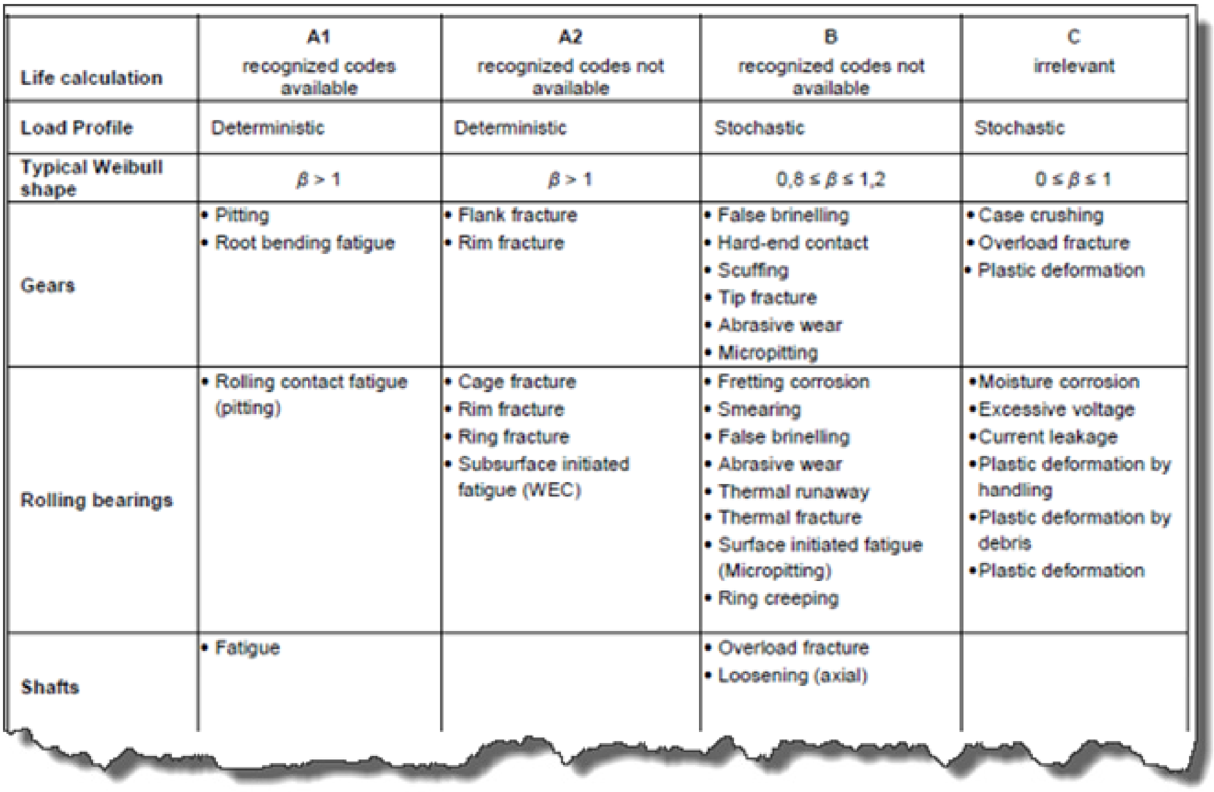


Figure 5 Calculation of the system reliability (Ref. 6).

The system reliability is determined by multiplying the reliability of the system elements. This assumes that the failure modes are independent of each other and that a failure leads to the failure of the functional element (Boolean condition) (Fig. 5).

$$R_s(t) = R_{C1}(t) \cdot R_{C2}(t) \cdot \ldots \cdot R_{Cn}(t) = \prod_{i=1}^{n} R_{Ci}(t)$$

The method provides calculation approaches for the A1



Life calculation	A1 recognized codes available	A2 recognized codes not available	B recognized codes not available	C irrelevant
Load Profile	Deterministic	Deterministic	Stochastic	Stochastic
Typical Weibull shape	$\beta > 1$	$\beta > 1$	$0{,}8 \leq \beta \leq 1{,}2$	$0 \leq \beta \leq 1$
Gears	• Pitting • Root bending fatigue	• Flank fracture • Rim fracture	• False brinelling • Hard-end contact • Scuffing • Tip fracture • Abrasive wear • Micropitting	• Case crushing • Overload fracture • Plastic deformation
Rolling bearings	• Rolling contact fatigue (pitting)	• Cage fracture • Rim fracture • Ring fracture • Subsurface initiated fatigue (WEC)	• Fretting corrosion • Smearing • False brinelling • Abrasive wear • Thermal runaway • Thermal fracture • Surface initiated fatigue (Micropitting) • Ring creeping	• Moisture corrosion • Excessive voltage • Current leakage • Plastic deformation by handling • Plastic deformation by debris • Plastic deformation
Shafts	• Fatigue		• Overload fracture • Loosening (axial)	

Figure 4 Classification of the system elements.

system elements.

The reliability of a component R is calculated using a 3-parametric Weibull distribution. These are the shape parameter β, the characteristic lifetime η for the failure probability $F(\eta) = 63.2\%$ and the location parameter γ, which is often interpreted as failure-free time in fatigue analysis. The reliability $R(t) = 1 - F(t)$ is the complement of the failure probability. If the component lifetime B_x is specified for another failure probability $F(B_x) = x\%$, the lifetime B_{10} is calculated as follows:

$$B_{10} = \frac{B_X}{\frac{\gamma}{B_{10}} + \left(1 - \frac{\gamma}{B_{10}}\right) \cdot \sqrt[\beta]{\frac{\ln(1-x)}{\ln(1-0{,}1)}}}$$

$$\eta = \frac{B_X - \gamma}{\sqrt[\beta]{-\ln(1-x)}}$$

$$R(t_d) = \begin{cases} 1 & \text{If } t_d \le \gamma \\ e^{-\left(\frac{t_d - \gamma}{\eta}\right)^{\beta}} & \text{If } t_d > \gamma \end{cases}$$

Recommendations for the form parameters and f_{tb} are given in the present paper.

The method provides in addition an extended calculation approach for the error modes tooth root breakage and pitting of involute gears. Based on ISO 6336-6 (Ref. 2), the damage sum for a certain load spectrum is determined and compared with the underlying Wöhler curve. Iteratively, the spectrum is expanded over time and the corresponding failure probability is calculated for each calculation step. By this, the failure probability of the system element over the operating time is obtained (Fig. 6).

The essential content of the method has been transferred to IEC 61400-4. The publication of Edition 2 will contain a chapter dealing with the determination of gearbox reliability, and there will be a reference to the VDMA paper. The IEC 61400-4 Edition 2 will be available as of 2021.

For more information.

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

References

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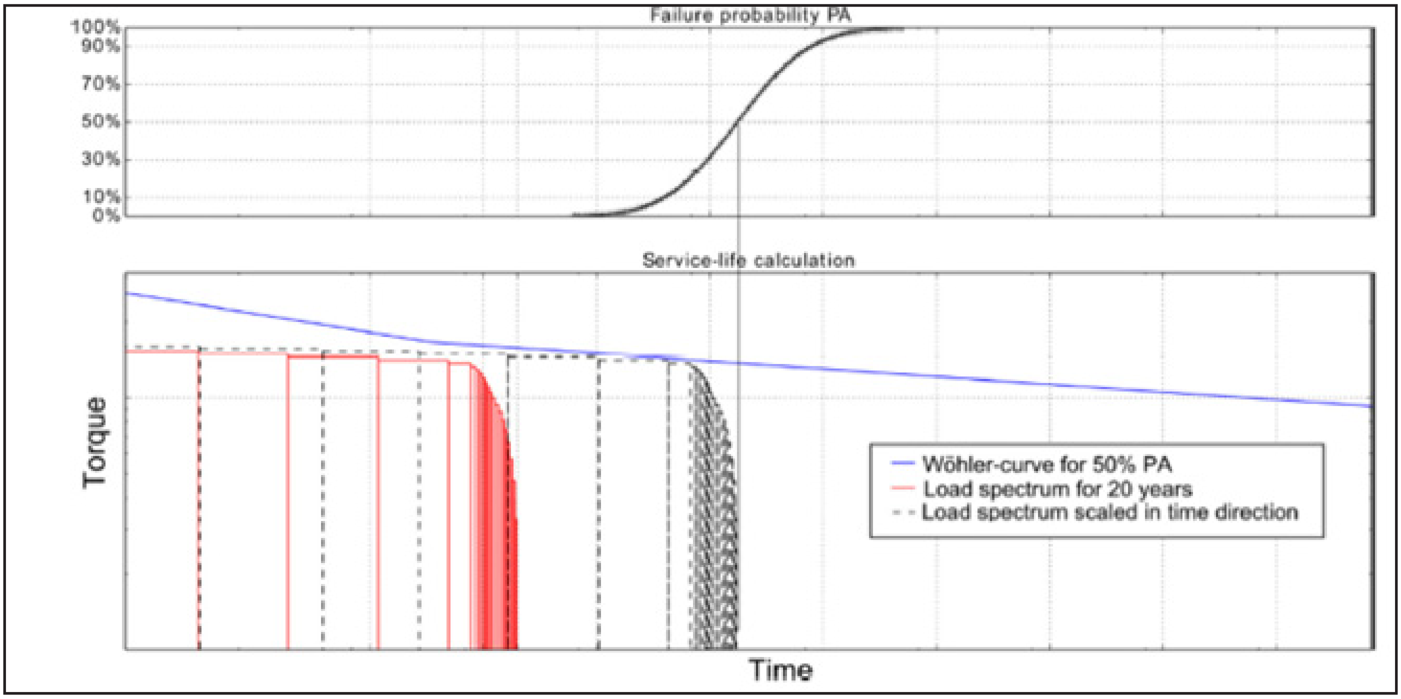

Figure 6 Iterative determination of the probability of failure based on the accumulation of damage according to ISO 6336-6.