Documentation of Gearbox Reliability — an Upcoming Demand

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The proof of the reliability of a gear drive is now an additional requirement. In Europe, the acceptance authorities for wind turbines are requesting a system reliability proof from gearbox manufacturers. The AGMA committee reviewing the AGMA 6006 standard for wind turbines is considering adding a chapter about "design for reliability." However, reliability considerations are not new; NASA, for example, was in the 1980s using reliability concepts for gear drives.

Most standardized methods used to assess the strength of gearbox components such as gears or shafts lead to a safety factor that expresses the ratio of permissible stress to effective stress. The permissible stress is determined based on a failure probability of the material strength value determined during the measurement of the S-N-curve.

In mechanical engineering, a safety factor greater than 1.0 does not always imply "safe" or less than 1.0 "not safe." It also depends on other reasons, as, for example, the consequences of a failure if a certain safety factor level is considered as safe or not. Therefore, it is quite difficult to understand the result of a gear strength analysis. For a drivetrain, the situation is even worse in that gear strength is expressed in safety factors for bending, pitting and scoring — but bearing strength is expressed in lifetime. So how is it possible to deduce if a drivetrain is safe?

Although these individual procedures for the design of a single component are very handy, it lacks an easy statement about the failure probability of the full system. This is aggravated by the fact that the intrinsic failure probability of the methods for different components varies. This paper presents a method to transfer the safety factor of each component into a failure probability based on a Weibull distribution — taking material properties into account. After this step it is easy to obtain a probability function for the failure of the full system over lifetime. Based on this, a statement such as "the probability of a failure of the gearbox within 20,000 hours is less than 0.2%" can be made.

The paper describes in detail how for the main gearbox components — gears, shafts, bearings — the strength results according to standards can be transformed in a Weibull failure probability distribution. The method can be applied to all standard calculations according to ISO, DIN and AGMA — which are based on an S-N-curve. It can be used for calculation with nominal load or duty cycles.

To determine the reliability of the drive system, the transmission elements are categorized: Is the failure of an element directly causing the gearbox failure? Is there redundancy? Thus, the system reliability can be determined with the components' reliabilities.

Symbol	Notation	Units
R	Reliability (of a single component)	%
R_s	Reliability of system	%
t	Number of load cycles	
t_0	Number of load cycles without failure (no failure during the to cycles, from the beginning)	
T	Characteristic service life (in cycles) with 63.2% probability of failure (or 36.8% reliability)	
fac	Number of load cycles per hour (conversion of operating hours into load cycles)	1/h
B	Weibull form parameter	
f_{tB}	Factor according to table 2	
H_{att}	Achievable service life of the component (in hours)	h
H_{att10}	Achievable service life of the component with 10% probability of failure	h
F_o	Specific probability of failure (for calculation of Hatt according to table 1)	%

Design Life

Typically, machines are designed for a certain lifetime, as, for example, 20,000 hours. So, the design engineer will lay out all the components of a machine based on such a request. But the methods used to check if a component fulfills the request vary.

For gears, the normal calculation method, e.g. — ISO 6336 (Ref. 4) — will provide, based on the requested life and the applied torque, a safety factor for bending and pitting. For bearings, the calculation method (per ISO281 (Ref.3)) will provide the attainable lifetime. For gears, the obtained safety factor must exceed a larger-than-requested minimum safety. The requested safety could be 1.0; but is often — depending on prescriptions by specific application rules — higher than 1.

Basically, for all other machine elements such as shafts, bolts and housings, the verification methods are also different. Calculation methods for mechanical parts were usually developed by different specialists working at different technical institutions. All these strength analysis methods have one thing in common, i.e. — they determine the stresses created by the applied loads and then compare these stresses with the permitted stresses. However, the calculation procedures differ greatly, depending on what type of machine element is involved (for example — bearings, shafts, gears or bolts). So, a check if all components of the verified object fulfills the requirements needs of specific knowhow.

This paper was first presented at the International VDI Conference on Gears 2017, Garching/Munich (VDI-Berichte 2294, 2017, VDI Verlag GmbH.

Failure Probability of Machine Elements

All of the above methods have one major weakness in common in that they are based on an intrinsic failure probability that differs from method to method (Table 1). So if a shaft has a calculated safety factor of 1.2 and a gear root has a safety factor of 1.3, it is not clear which is the more critical component.

A material strength value with a failure probability of 90% is higher than a material strength value with a failure probability of 99%. Therefore, if the 90% failure probability is applied, the safety factor is greater and the element has both a greater service life and a lower damage rate for its design life; damage that is calculated using methods prescribing different failure probabilities cannot be compared directly. A gear unit may fail because of a part that is not considered to be critical breaks prematurely; this happens quite frequently in real life.

Table 1 Probability of failure used by various calculation methods when determining material properties

Shaft, DIN 743	2.5%	Assumed, not documented
Shaft, FKM guideline	2.5%	
Shaft, AGMA 6001	1%	If $k_c = 0.817$
Bearing, ISO 281	10%	If factor $a_1 = 1.0$
Tooth flank, ISO 6336; DIN 3990	1%	
Tooth bending, ISO 6336; DIN 3990	1%	
Tooth flank, AGMA 2001	1%	If reliability factor $K_R = 1$
Tooth bending, AGMA 2001	1°/0	If reliability factor $K_R = 1$

Table 2 Factors for a Weibull distribution according to Bertsche (Ref. 2)

	factor f_{tB}	Weibull form parameter b
Shafts	0.7 to 0.9	1.1 to 1.9
Ball bearing	0.1 to 0.3	1.1
Roller bearing	0.1 to 0.3	1.35
Tooth flank	0.4 to 0.8	1.1 to 1.5
Tooth root	0.8 to 0.95	1.2 to 2.2

To overcome this problem, the reliability concept can be used. Here the result is a curve that shows the probability of failure of a component or a system over the lifetime. When statistical parameters, such as the scatter of results in a standard distribution, are determined based on measurements on probes, a probability of failure as a function of time (or cycles) can be determined using a statistical approach. The opposite of the probability of failure is called "reliability." Therefore, since the reliability calculation takes into consideration the inherent failure probability (Table 1), the calculated reliability at design life of different parts can be compared effectively with each other. Also, at a given probability level the component with the smallest achievable life is the critical component of the system.

Probability Distributions

In statistics, probability distributions are used to describe stochastic processes (see numerous textbooks, e.g. (Ref. 8)). A probability distribution is a function that gives the likelihood of an event for a specific value of a probability variable. In our case the event is failure (or survival) and the probability variable is the number of load cycles.

The reliability function *R(t)* gives the probability of survival until *t* load cycles. For the definition of a probability distribution the first derivative *R'(t)* is defined, i.e. — the so-called density. The density is a function that defines the probability of the event happening at a given number of load cycles.

The most common distribution for general purposes is the normal distribution. This distribution is defined by the mean value μ and the standard deviation σ. The standard deviation σ controls how wide the distribution is. However, although for small σ the density looks like it becomes zero with enough distance from the mean, it never actually does. So also for negative values of *t*; there is a positive likelihood that failure occurs. Due to these limitations the normal distribution is not very often used in reliability engineering.

A more general approach is the Weibull distribution, in which two variants are possible — the two-parametric and the three-parametric Weibull distribution, where the two-parametric is a special case of the three-parametric.

The two-parametric Weibull distribution leads to the reliability function

$$R(t)=e^{-\left(\frac{t}{T}\right)^b} \tag{1}$$

where T is the characteristic lifetime (defined by the condition $R(T) = 0.632$) and b is the shape parameter.

The three-parametric Weibull distribution has t_0 as a third parameter, which shifts the first occurrence of failure to the point t_0 by the substitution

$$t \rightarrow \tilde{t} - t_0 \tag{2}$$

This substitution gives the reliability function

$$R(t)=e^{-\left(\frac{t-t_0}{T-t_0}\right)^b} \tag{3}$$

Determining the Reliability of Machine Elements

There are currently no mechanical engineering standards that include the calculation of probability. A classic source for this calculation is Bertsche's book (Ref. 2), in which the possible processes have been described in great detail. Bertsche recommends the use of the 3- parameter Weibull distribution.

Parameters T and t_0 can be derived from the achievable life of the component, H_{att}, as follows (with failure probability F_o according to the calculation method from Table 1, b and f_{tB} from Table 2 according to Bertsche):

$$T=\left(\frac{H_{att}-f_{tB}\times H_{att10}}{\sqrt[\beta]{-ln\left(1-\frac{F_o}{100}\right)}}+f_{tb}\times H_{att10}\right)\times f_{ac} \tag{4}$$

$$t_0=f_{tb}\times H_{att10}\times f_{ac} \tag{5}$$

with

$$H_{att10}=\frac{H_{att}}{(1-f_{tB})\times\sqrt[\beta]{\frac{ln\left(1-\frac{F_o}{100}\right)}{ln\,(0.9)}}+f_{tB}} \tag{6}$$

Equation 1 for $R(t)$ can now be used to display the progression of reliability over time (or number of cycles) as a graphic.

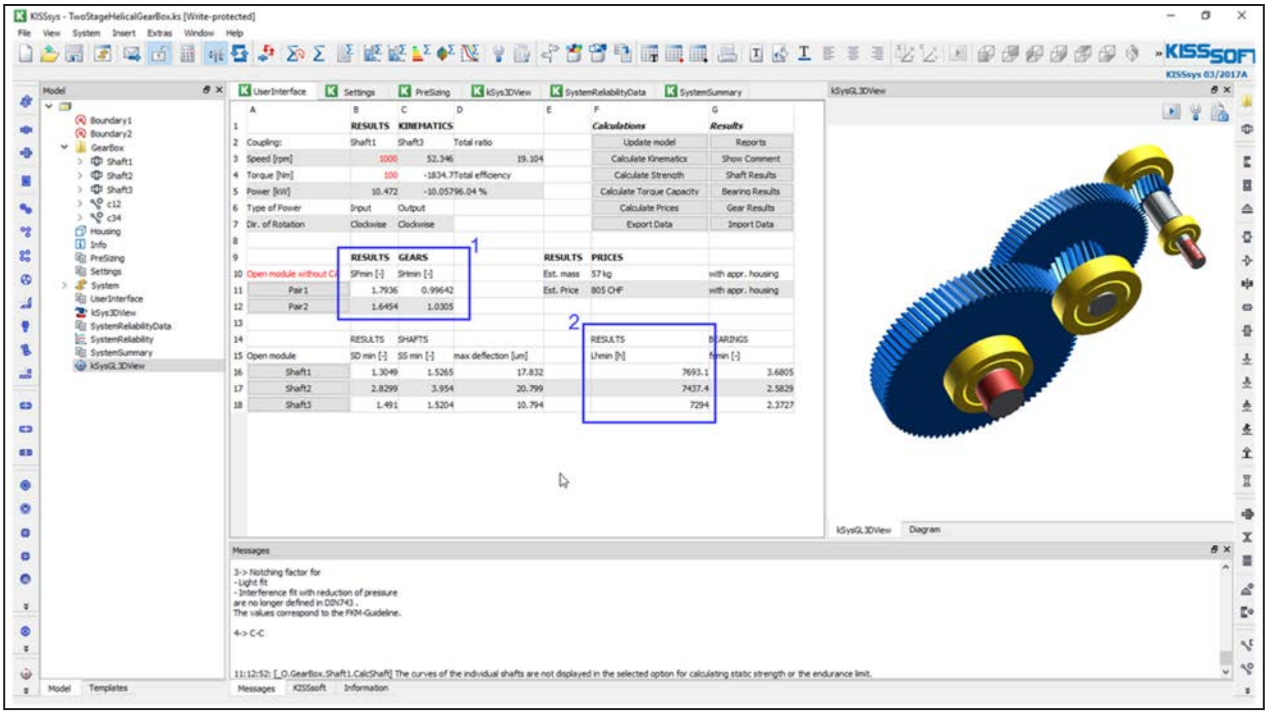

Figure 1 Example of gearbox modeled in *KISSsys*.

The load cycle values t_0 and T can then be calculated based on a service life calculation. Equations 4-6, using the achievable service life H_{att}, can be used for this purpose.

An Example

To illustrate the differences between the concept of safety factors and reliability an example is shown. Figure 1 shows the model of a two-stage gearbox in *KISSsys* (Ref. 6). The design life is 5,000 h, so the critical component appears to be gear 1 with a flank safety factor of slightly below 1.0 (0.996) — see box "1." The bearings have a calculated lifetime of above 7,000 hours — see box 2 — and thus seems to be on the safe side.

However, looking at the reliability graph in Figure 2, the situation is different. The left-most first curve of individual components is the one for gear 1, confirming the previous assessment. But this is only true for relatively low probabilities; the lower horizontal red line is on the 90% probability level. Here, gear 1 has indeed the shortest lifetime. Still, this is above the required 5,000h design life — which is marked with the vertical grey line.

At 99% probability, the bearing life is much lower — about 3,000 hours. This is marked with the upper horizontal red line; indeed, the most critical components are the bearings.

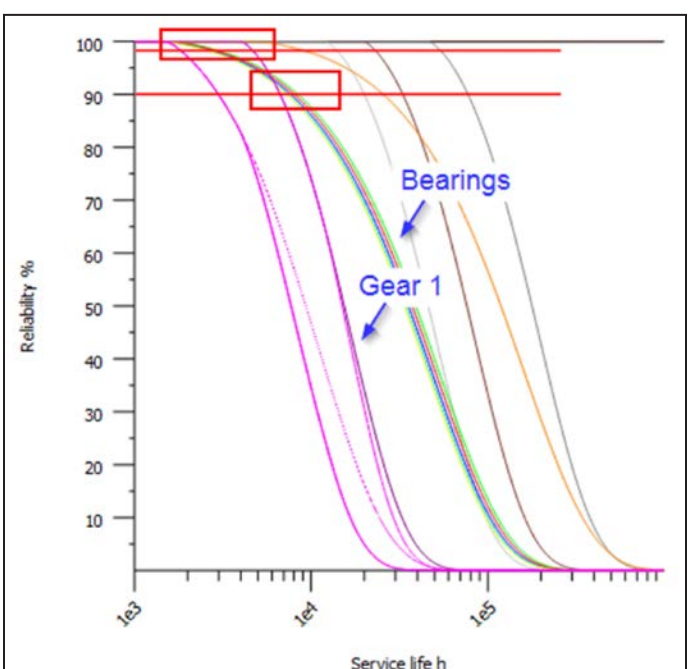

Figure 2 Calculated reliability curves.

Determining System Reliability

Determining the overall reliability of a gear drive is of primary concern for all important drives. In particular, people who are not technical specialists are not particularly interested in knowing which is the critical bearing in a drive; they are much more concerned about the drive's service reliability over a predefined period of operation. However, the reliability of individual elements in a gear unit must be used to determine the reliability of the overall system.

The functional block diagram of the gear unit must be analyzed before the reliability of individual components is used to calculate overall reliability. In order to determine system reliability, the gear unit elements are classified according to their significance, i.e. — if the element fails, does it directly cause the failure of the entire gear unit? Or are redundancies present? The overall reliability of the entire system can then be determined by mathematically combining the reliability of the individual components.

In particular, a distinction must be drawn as to whether the significant components are connected in series or in parallel.

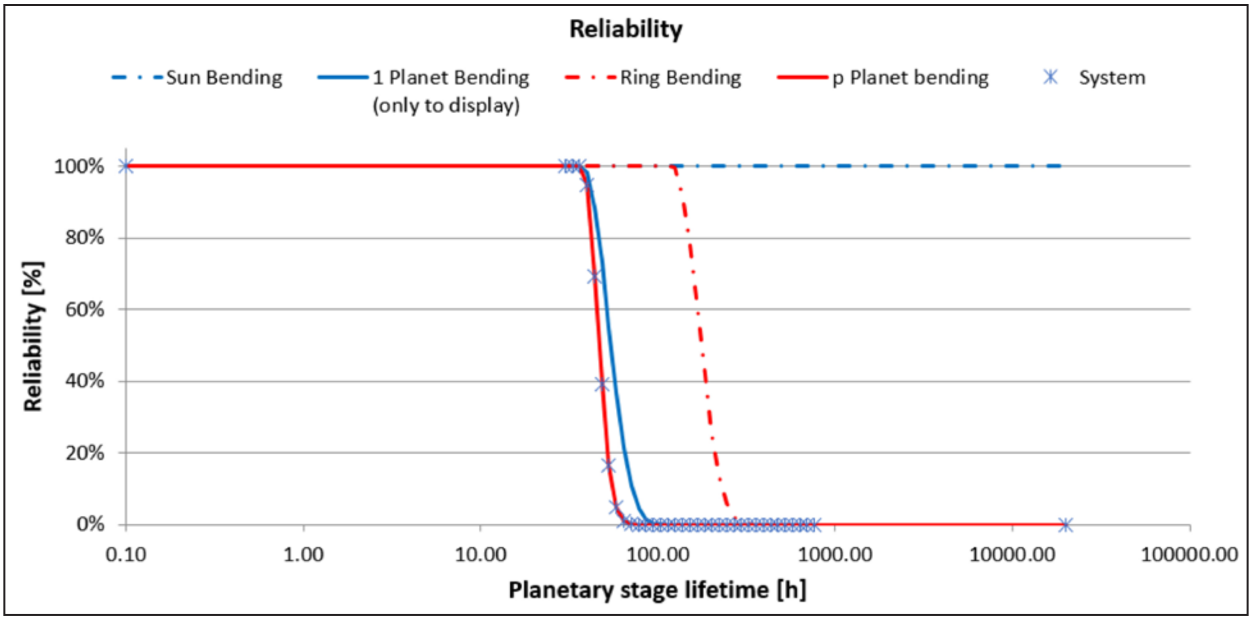

Figure 3 Reliability diagram for a planetary stage with 3 planets; the 3 serially accumulated planets are critical. The system reliability is virtually identical to the reliability of these 3 planets because the ring and the sun have a significantly higher degree of reliability.

Although this appears to be complicated at first glance, it is usually quite straightforward for most gear units. If any of the vital elements in a standard gear unit (bearing, shaft, gear) break, this will cause the entire gear unit to fail. This means that all these elements are connected in series. Gear units designed with redundancies are not commonly found in practice. In this design type, the power flow runs through two parallel branches within the gear unit. If an element within one of the branches fails, the other branch continues to run the unit as a whole.

The following equation can be used to determine system reliability for serial functions:

$$R_s(t) = \frac{R_{c1}(t)}{100} \times \frac{R_{c2}(t)}{100} \times \ldots \times \frac{R_{cn}(t)}{100} \times 100 \text{ or } R_s(t) = 100 \times \prod_{i=1}^{n} \frac{R_{ci}(t)}{100} \quad (7)$$

Bertsche (Ref. 2) has also developed formulae for the less-commonly found cases for units with redundancies (parallel branches).

Reliability for Gear Pairs and Planetary Stages

Gear pairs and planetary stages will be discussed here as an introduction to examine entire systems; these types of configurations are sub-systems in themselves. The procedure for a classic gear pair is quite straightforward: the overall reliability is the product of the four "elements" — tooth root (*f*) and tooth flank (*h*), for the pinion (1) and the gear (2) in each case:

$$R_{pair}(t) = \frac{R_{f1}(t)}{100} \times \frac{R_{h1}(t)}{100} \times \frac{R_{f2}(t)}{100} \times \frac{R_{h2}(t)}{100} \times 100 \quad (8)$$

In planetary stages, the power flow is distributed across the planets. Theoretically, the planetary stage could continue working — even if one planet fails — because of the built-in redundancy of this design. Theoretically, therefore, the planet stage is connected in parallel. However, in practice the failure of one planet (gear or bearings) usually means that metallic fragments penetrate the tooth meshings and bearings, and thus cause other parts to fail. For this reason, these elements have to be considered as connected in series. The reliability of the planetary stage can therefore be determined as follows (p: number of planets):

$$R_{pstage}(t) = \frac{R_{f1}(t)}{100} \times \frac{R_{h1}(t)}{100} \times \left(\frac{R_{f2}(t)}{100} \times \frac{R_{h2}(t)}{100}\right)^{p} \times \frac{R_{f3}(t)}{100} \times \frac{R_{h3}(t)}{100} \times 100 \quad (9)$$

A publication by NASA (Ref. 7, Eq. 43) about the reliability of planetary stages confirms the proposed method. The authors use the same approach for calculating overall reliability, but without providing justification as to why they use the serial formula for the planets.

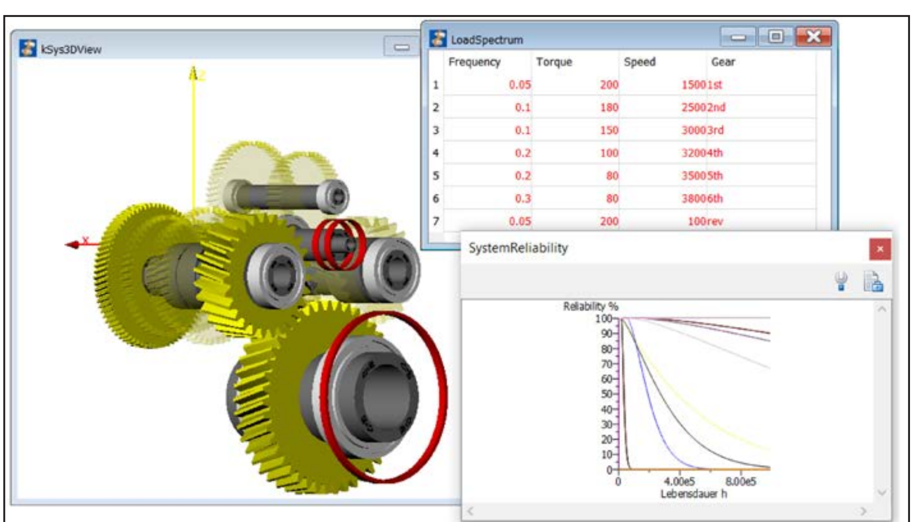

Figure 4 Modern dual-clutch gear unit with a load spectrum (6 forward gears and one reverse gear); system reliability displayed with a linear scale.

System reliability. The major benefit of using reliability as a parameter for qualifying the gear elements is that it is a quick and relatively simple method for determining system reliability. In *KISSsoft* (Ref. 6) the achievable service life is also calculated every time a verification is performed. Consequently, the data for each individual element of the gear unit is automatically available. This data is then forwarded to a system program as *KISSsys* (Ref. 6). System reliability can therefore be determined at system level. In addition to showing overall reliability, the weakest elements in a gear unit are also clearly displayed in this type of diagram.

In the case of vehicle gearboxes, the calculation for components must be performed with a complex load spectrum that also takes into account the shift setting (shifted gear, time, torque and speed) (Fig. 4). This calculation determines the service life of all the components, and the reliability can be derived from these values. The calculation of system reliability also assumes that the components are switched in series. Obviously, if, for example, the second gear fails, the vehicle can still be driven in a different gear. However, this should be regarded as a hypothetical scenario that would apply in an emergency.

System reliability is of critical importance for gear units used for wind turbines (Fig. 5), because any repairs are very expensive. Wind turbine manufacturers therefore require their gear unit suppliers to provide very extensive proofs. Proofs of system reliability are already a requirement in this sector (Ref. 1). AGMA 6006 (Ref. 5), a U.S. standard for wind power gear units, is currently under revision. It is likely that this revised version of AGMA 6006 will include a new method for calculating system reliability — the very first mechanical engineering standard to do so.

Outlook

Displaying an analysis of gear drive strength in terms of system reliability can easily be understood by people who do not have a detailed knowledge of the modern calculation methods used for gearbox components. It is also the only method that can be statistically evaluated and used to make a comprehensive assessment (gear unit will stop/will not stop) with a corresponding level of probability. This method has become increasingly popular and widespread, but a number of problems still remain; for example: should the inclination of the S-N curve in the limited life range affect the Weibull form parameter β? As yet no reliable approaches to this problem have been documented and additional research is needed.

As mentioned, AGMA 6006 (Ref. 5), a U.S. standard for wind power gear units, is currently under revision. Since the first version of this standard appeared in 2003, it has been used as the basis for the currently valid international IEC/TC 88 standard for gear units used in the wind power generation industry. It is likely that this revised version of AGMA 6006 (Ref. 5) will include a new method for calculating system reliability — the very first mechanical engineering standard to do so. We can then presume that the AGMA will propose this type of method in the IEC/TC 88 workgroup as a supplement to IEC 61400 "wind turbines" regulation.

Summary

Modern calculation methods based on S-N curves can be used to analyze every essential element in a gear unit. These methods determine the achievable service life of the gearbox elements, which in turn can be used to calculate the Weibull distribution for reliability.

The reliability of a gear drive can be determined by calculating the reliability of the gear unit components. The use of reliability as a parameter for assessing a gear unit is currently

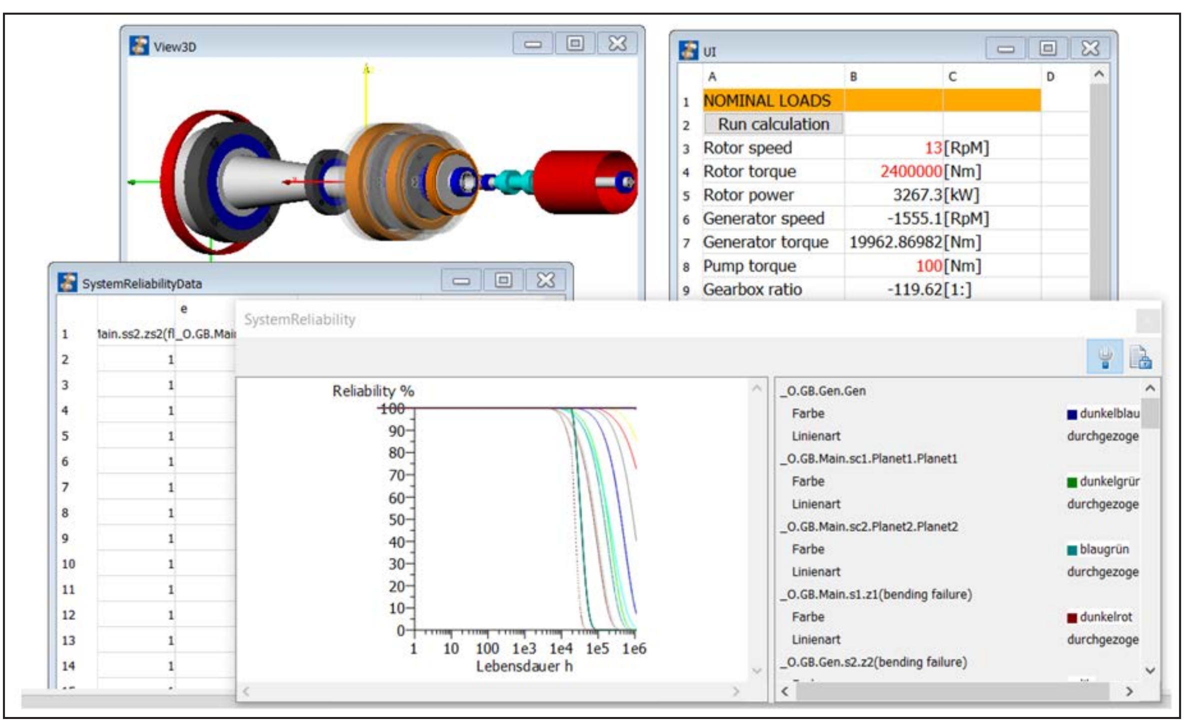

Figure 5 Wind power gear unit with its reliability.

becoming a popular method, and could well be a requirement in the near future for gear units used in wind power generation.

People who are not technical experts will find this method of displaying reliability much easier to understand than a table of achieved safeties for gears and service life values for rolling bearings. They do not need to understand that material properties that comply with ISO 6336 have a 1% failure probability, or that the calculated service life of bearings has 10% failure probability. Nor do they need to know that a higher minimum safety is usually prescribed against tooth bending than against pitting. All these different approaches can be used together to provide a well-balanced statement of reliability, with values that really can be compared with each other. However, when a design review expert is provided with these types of calculations he must (maybe even more so than before) still check exactly which conditions, for example — which minimum safeties — have been used to determine reliability. **PTE**

For more information.
Questions or comments regarding this paper?
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Dr. Ulrich Kissling studied machine engineering (1976-1980) at the Swiss Technical University (ETH), where he also completed his doctoral thesis — "Pneumatic Weft Insertion on Weaving Machines. In 1981 he started his professional career as calculation engineer for a gearbox manufacturing company in Zurich, progressing there to technical manager and ultimately managing director. As a calculation engineer for gearbox design, he began developing software for gear, bearing and shaft layout. In 1985 he branded this software 'KISSsoft' and started to market it, selling its first license in 1986. In 1998 he founded his own company — KISSsoft AG — concentrating on software development and growing staff from three people in 1998 to workers in 2017. Today, aided by the contributions of partner and managing director Dr. Stefan Beermann, KISSsoft is the leading drivetrain design software, used by more than 3,000 companies on all continents. An internationally respected gear expert, Dr. Kissling is chairman of the TK25 committee (gears) of the Swiss Standards Association (SNV) and a voting member for Switzerland in the ISO TC 60 committee. He actively participates in different work groups of ISO for the development of international standards.

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