From the Single Component Safety Factor to the System Reliability Rating

The Reliability Concept: A New Way to Assess Gears and Gear Drives

Dr. Ing. Ulrich Kissling, KISSsoft AG
Dr. Ing. Michael Stangl, KISSsoft AG
Dr. Ing. Inho Bae, KISSsoft AG

Introduction

Today, proofs of system reliability for plant components such as gear units or complete mechanical systems are in increasing demand in various sectors of the power transmission industry. However, there has been no common language to interpret the reliability for each component, for example, gear engineers prefer the term ‘safety factor’ while bearing engineers use the service life. A component's safety factor or its calculated service life is actually nothing else than a statement of its reliability. Determining the reliability of individual parts makes it easier to ascertain the reliability of the mechanical system as a whole.

Stating a system's reliability is also more comprehensible than listing safety factors for those people without detailed knowledge in mechanical engineering. A statement such as "the probability that gear unit X will fail during its guaranteed service life of 50,000 hours is less than 0.02%." is much easier to understand than "the safety factors of all the gears in gear unit X, calculated for an operating time of 50,000 hours, are all bigger than 1.6.", although both statements mean the same thing.
This paper describes how the probability of failure of the basic gearbox components (shafts, bearings, gears) can be derived from the component’s service life as specified in the standards, according to the Weibull failure criterion. In order to determine system reliability, the gear unit elements are classified according to their significance: if an element fails, does it directly cause the failure of the entire gear unit? Or are redundancies present? The total reliability of the entire system can then be determined by mathematically combining the reliability of the individual components. This method can be applied to all ISO-, DIN- or AGMA-standard calculations that use S-N curves (Woehler lines), either with nominal loads or with load spectra.

1 Calculating the strength of mechanical components

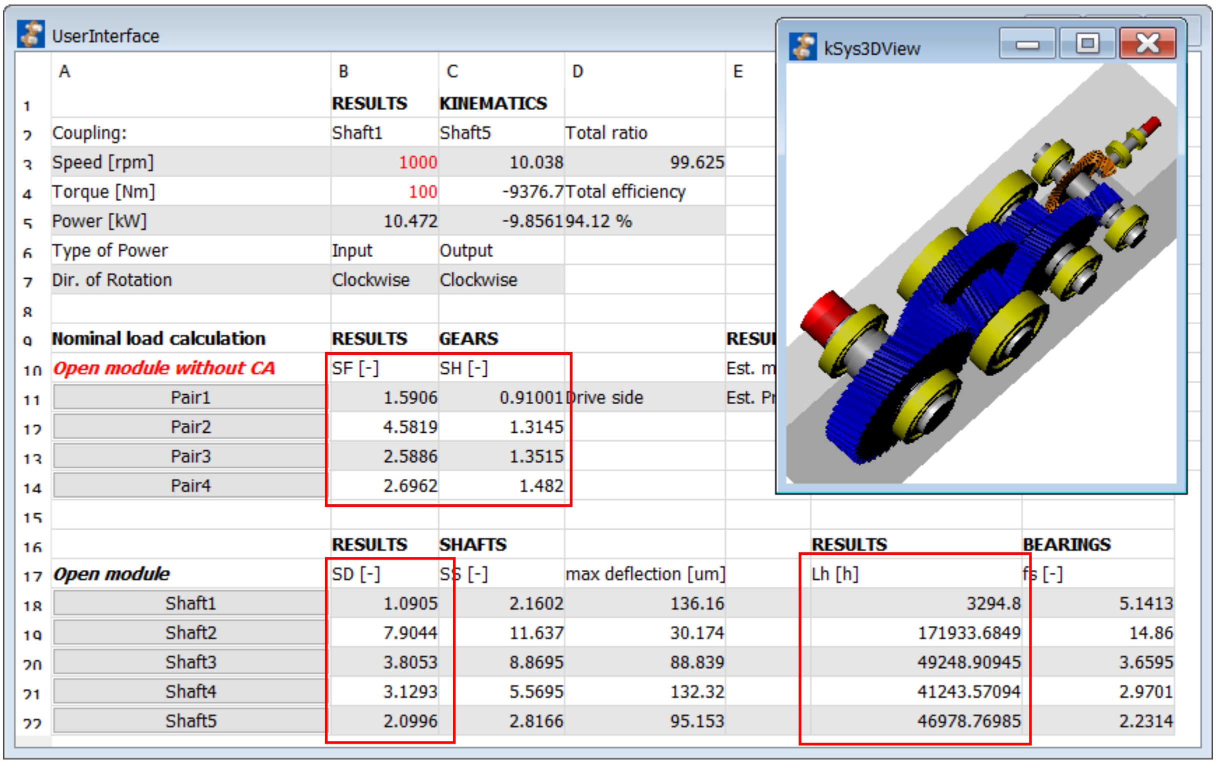
For many years – and increasingly, since the beginning of the 20th century − engineers have striven to develop rules for analyzing the strength of elements used in mechanical engineering. German engineers in particular, used different combinations of basic mechanical engineering formulae in the attempt to define calculation rules for sizing components. This approach has proven to be extremely successful, and is implemented world-wide today. A clear illustration of its success is that every ISO calculation standard published to date is based on this principle.

Calculation methods for mechanical parts are usually developed by different specialists working at different technical institutions. All these strength analysis methods have one thing in common: 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). It is these differences in the calculation methods that presents us a problem. One might expect that the safety factor, which is the permissible stress divided by the occurring stress, would be sufficient. So that a safety factor greater than 1.0 would means that the part is adequately dimensioned. Unfortunately, this is not the case. For bearings, the service life is determined and not the safety factor. In a gear calculation according to ISO [4], the safeties for the tooth root and flank are determined, giving rise to the question which of the two criteria is decisive, and when. In addition, it is general that different safety factors are recommended, for example, a minimum safety of 1.4 is used for the tooth root, but a minimum safety of 1.0 is used for the flank. The reason for the different required minimum safeties is: if a gear tooth breaks, the entire gear unit will immediately fail, and this is not the case if the flank becomes pitted. When checking the scuffing risk of a gear, a minimum safety of 2.0 is required; this is because the calculation method is considered to be "not yet sufficiently tested". In other hand, the minimum safety that must be achieved in shaft calculation according to the FKM method [9] depends on the component's importance, i.e. the consequences of the shaft breaking. This is obviously a very sensible approach. The VDI bolt calculation [13] for safety against sliding for bolted parts requires a minimum safety of between 1.2 and 1.8, depending on the load. And many similar examples could be added to this list. The conclusion is: depending on the part, "safety factor" is not equal "safety factor".

Assessing the result of a verification therefore involves a great deal of time and effort, and requires a thorough knowledge of the calculation methods and the minimum safeties to be applied. Figure 1 shows the result of a strength calculation performed on all the vital components of a 4-stage bevel helical gear unit in KISSsys [10]. Although the results are clearly displayed, even experienced experts would find it difficult to identify at first glance:

- whether the gear unit is adequately dimensioned to support the nominal torque (100 Nm) during the required service life (5,000 h)
- which part is the weakest element in the gear unit, and may need to be redesigned

For the gear unit shown in Figure 1, the question might be whether the most critical bearing (on the "Shaft1" input shaft) with only 3,300 h service life or the inadequate flank safety of the bevel gear pair ("Pair1" with a safety factor of only 0.91), is more significant and might lead to a premature failure. In this case, the fact that the flank safety is proportional to the square of the torque must be taken into consideration. If the nominal torque is reduced from 100 to 88 Nm, the flank safety only increases from 0.91 to 0.96. In contrast, the service life of the most critical bearing rises from 3,300 to 5,100 h. The bevel gear pair is therefore the weakest element in this gear unit.



	A	B	C	D	E	F	G
1		RESULTS	KINEMATICS				
2	Coupling:	Shaft1	Shaft5	Total ratio			
3	Speed [rpm]	1000	10.038	99.625			
4	Torque [Nm]	100	-9376.7	Total efficiency			
5	Power [kW]	10.472	-9.8561	94.12 %			
6	Type of Power	Input	Output				
7	Dir. of Rotation	Clockwise	Clockwise				
8							
9	Nominal load calculation	RESULTS	GEARS		RESU		
10	*Open module without CA*	SF [-]	SH [-]		Est. m		
11	Pair1	1.5906	0.91001	Drive side	Est. Pr		
12	Pair2	4.5819	1.3145				
13	Pair3	2.5886	1.3515				
14	Pair4	2.6962	1.482				
15							
16		RESULTS	SHAFTS			RESULTS	BEARINGS
17	*Open module*	SD [-]	SS [-]	max deflection [um]		Lh [h]	fs [-]
18	Shaft1	1.0905	2.1602	136.16		3294.8	5.1413
19	Shaft2	7.9044	11.637	30.174		171933.6849	14.86
20	Shaft3	3.8053	8.8695	88.839		49248.90945	3.6595
21	Shaft4	3.1293	5.5695	132.32		41243.57094	2.9701
22	Shaft5	2.0996	2.8166	95.153		46978.76985	2.2314

Figure 1 Overview of the results of a 4-stage bevel helical gear unit (SF: bending safety; SH flank safety; SD: safety of the most critical shaft cross section; Lh: bearing service life)

2 Determining the service life, damage and exposure of machine elements

All the calculation methods that use a material's S-N curve to define the permissible stress can be used to determine the achievable service life. As a consequence, this approach can be applied in every gear and bearing calculation. In the latest versions of DIN 743 (2012) [7] and the FKM guideline [9], S-N curves can be used in shaft calculations; when AGMA 6001 [5] is used, it can only be used to a limited extent. Both of the minimum safety and the load must be specified in the calculation. The service life is then determined with reference to this minimum or required safety. This approach makes it possible to directly compare the calculated service life values of the different parts with each other. The element with the lowest service life is therefore the weakest element in the gear unit.

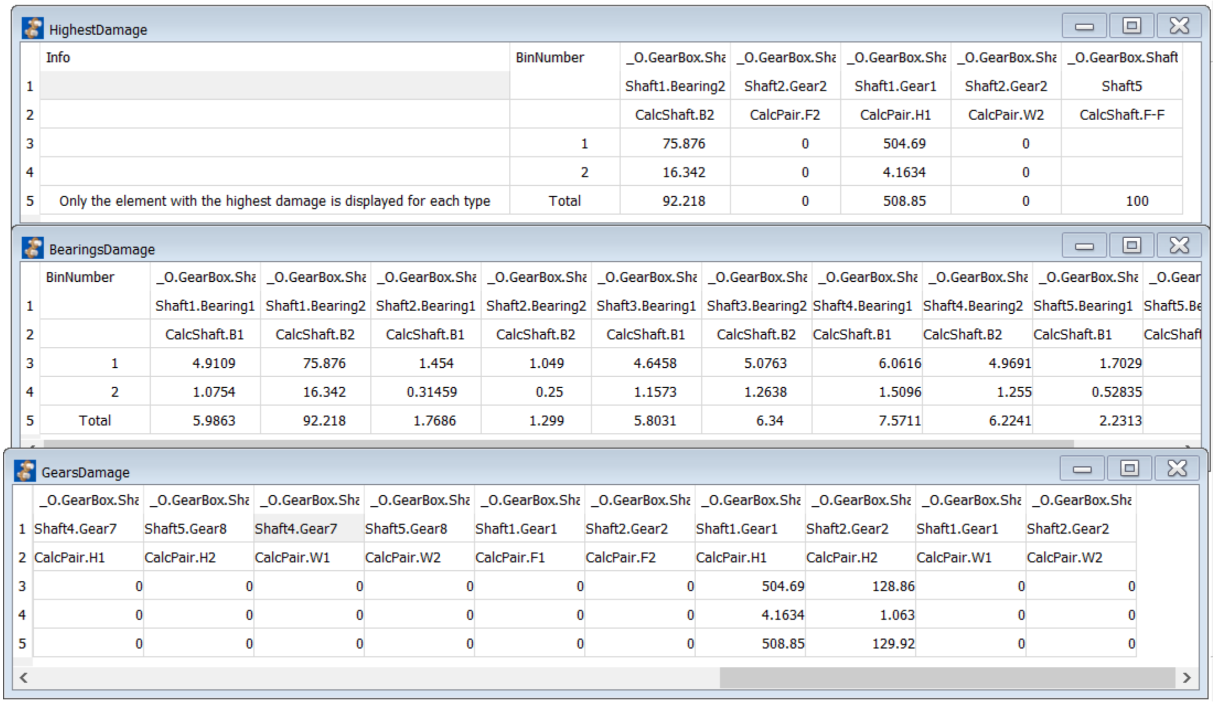
One of the most useful values, especially where load spectra are involved, that can be derived from the service life is the damage of a part. The "damage" is equal to the ratio of required service life to achievable service life. Therefore, the damage of a component progresses proportionally to time (number of load cycles). Figure 2 shows the result of a cylindrical gear pair with a load spectrum. The amount of damage specified for each damage criterion (root/flank, pinion/gear) and for each bin clearly shows which is the dominant criterion for damage, and which is the most damaging bin.

Bin No.	Frequency [%]	Power [kW]	Speed [1/min]	Torque [Nm]	Damage calculated on the basis of required service life (20,000 h)				
					No.	F1%	F2%	H1%	H2%
1	0.00020	175.0000	440.8	3791.1	1	0.08	0.04	0.00	0.00
2	0.00160	172.0250	440.8	3726.6	2	0.54	0.30	0.03	0.01
3	0.02800	166.2500	440.8	3601.5					

4	0.27200	158.9000	440.8	3442.3	3	7.25	3.97	0.41	0.12
5	2.00000	150.1500	440.8	3252.7	4	49.86	26.45	3.02	0.87
6	9.20000	141.4000	440.8	3063.2	5	27.25	121.40	9.58	2.54
7	28.00000	132.6500	440.8	2873.6	6	6.35	35.48	16.63	4.41
8	60.49820	123.9000	440.8	2684.1	7	0.00	4.53	17.96	4.77
					8	0.00	0.00	13.26	3.52
					Σ	91.33	192.18	60.89	16.24

Figure 2 Load spectrum (left) and damage displayed per bin for tooth root (F1: pinion, F2: gear) and flank (H1: pinion, H2: gear). Required service life: 20,000 h. Achievable service life of the gear: 10,400 h (gear tooth root). Therefore, the total calculated damage is 192%.

Summarizing the results of a gear calculation by detailing the damage to all the most important parts (Figure 3) is a quick and direct way to localize the weakest part in a gear unit and obtaining the overall result and to determine whether or not the gear meets the requirements (no single damage value is greater than 100%). Compared to the standard display method described above (as shown in Figure 1), the information shown when damage is taken into account is more uniform (no safety factors for gears and no service life values for bearings). In addition, different predefined minimum safeties are already integrated in the result and therefore do not need to be taken into account again when the results are compared.



HighestDamage

	Info	BinNumber	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Shaft
1			Shaft1.Bearing2	Shaft2.Gear2	Shaft1.Gear1	Shaft2.Gear2	Shaft5
2			CalcShaft.B2	CalcPair.F2	CalcPair.H1	CalcPair.W2	CalcShaft.F-F
3		1	75.876	0	504.69	0	
4		2	16.342	0	4.1634	0	
5	Only the element with the highest damage is displayed for each type	Total	92.218	0	508.85	0	100

BearingsDamage

	BinNumber	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.Gear
1		Shaft1.Bearing1	Shaft1.Bearing2	Shaft2.Bearing1	Shaft2.Bearing2	Shaft3.Bearing1	Shaft3.Bearing2	Shaft4.Bearing1	Shaft4.Bearing2	Shaft5.Bearing1	Shaft5.Be
2		CalcShaft.B1	CalcShaft.B2	CalcShaft.B1	CalcShaft.B2	CalcShaft.B1	CalcShaft.B2	CalcShaft.B1	CalcShaft.B2	CalcShaft.B1	CalcShaft
3	1	4.9109	75.876	1.454	1.049	4.6458	5.0763	6.0616	4.9691	1.7029	
4	2	1.0754	16.342	0.31459	0.25	1.1573	1.2638	1.5096	1.255	0.52835	
5	Total	5.9863	92.218	1.7686	1.299	5.8031	6.34	7.5711	6.2241	2.2313	

GearsDamage

	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha	_O.GearBox.Sha
1	Shaft4.Gear7	Shaft5.Gear8	Shaft4.Gear7	Shaft5.Gear8	Shaft1.Gear1	Shaft2.Gear2	Shaft1.Gear1	Shaft2.Gear2	Shaft1.Gear1	Shaft2.Gear2
2	CalcPair.H1	CalcPair.H2	CalcPair.W1	CalcPair.W2	CalcPair.F1	CalcPair.F2	CalcPair.H1	CalcPair.H2	CalcPair.W1	CalcPair.W2
3	0	0	0	0	0	0	504.69	128.86	0	0
4	0	0	0	0	0	0	4.1634	1.063	0	0
5	0	0	0	0	0	0	508.85	129.92	0	0

Figure 3 Damage values for all the critical elements of a gear unit in KISSsys. The top table shows the damage per element type (gears, shafts, rolling bearings) - the most critical in each case. The middle table shows the damage (for root and flank) for all the gears. The bottom table shows all the bearings.

In recent years, strength calculation methods such as the FKM guideline [9] for shafts or the draft of an ISO standard for flank fracture [12], use the term ‘material exposure’ to express the final result of the calculation procedure. Exposure is, in itself, the reciprocal value of the mathematical safety factor, but also already includes the required minimum safety. Exposure is therefore proportional to the load and, for this very reason, cannot be proportional to the damage. As the load and service life are linked by the logarithmic S-N curve, a 10% increase in exposure - depending on the inclination of the S-N curve– will result in an increase of 100% or more in damage. Only if the results are considered more from a load-oriented viewpoint, the use of exposure can be given priority over damage and service life.

3 Failure probability of machine elements

The use of damage as a criterion to quantify the reliability of gear unit components (as described above) would appear to be the ideal method for determining the service life of gear unit components. However, there is a problem: material properties, such as the S-N curve, are measured by taking samples. The measurement results scatter. In order to obtain a characteristic value for the calculation, it is usually assumed that the measurement values are scattered in a standard distribution. The probability of damage for which the strength values used in the calculation must then be defined, and once again this will differ according to the method used (Table 1).

Calculation procedure	Probability of damage Fo			
	1%	10%	Other	Comment
Shaft, DIN 743			2.5%	Assumed, not documented
Shaft, FKM guideline			2.5%	
Shaft, AGMA 6001	*			If kC = 0.817
Bearing, ISO 281		*		If factor a1 = 1.0
Tooth flank, ISO 6336; DIN 3990	*			
Tooth bending, ISO 6336; DIN 3990	*			
Tooth flank, AGMA 2001	*			If reliability factor KR = 1
Tooth bending, AGMA 2001	*			If reliability factor KR = 1

Table 1 Probability of damage used by different calculation methods when determining material properties

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 required service life. Damage that is calculated using the methods prescribing different failure probabilities cannot be compared directly. Therefore, calculated values for damage are not exact. They are subject to the scatter of material property values and other effects that are ignored by the calculation methods and, as a consequence, are themselves subject to a statistical distribution. A gear unit may fail because of the part that is not considered to fail critically and to break prematurely. This happens quite frequently in real life.

Therefore, when an achievable service life (or a damage value for the required service life) is documented, the associated failure probability must also be added. Now, when statistical parameters, such as the scatter of results in a standard distribution, are determined on the basis of 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, as the reliability calculation takes into consideration of the inherent failure probability (Table 1), the calculated reliability at a required service life of different parts can be compared effectively with each other.

Reliability is expressed as a percentage, from 0% to 100%, and psychologically also has an important side-effect, because safety factors give the impression of being absolute values: a gear unit with high safety factors cannot fail. In contrast, displaying the same results as reliability, even if it is 99.99%, shows that there is always an element of uncertainty.

4 Determining the reliability of machine elements

Methods for calculating reliability are still not widely used. However, they gain increasing interest, for example, in the wind energy sector, where there is a demand for accurately determining system reliability [1]. There are currently no mechanical engineering standards which include this type of rule. A classic source for this calculation is Bertsche's book [2], in which the possible processes have been described in great detail. For this reason, the various different methods will not be discussed in this paper. The most commonly used approach, and one which is well suited to the results that can be achieved in "traditional" mechanical engineering calculations, is the "Weibull distribution". In this case, Bertsche recommends the use of the 3-parameter Weibull distribution. The reliability, R, of a machine element is calculated as a function of the number of load cycles, t, using Equation (1).

$$R(t) = e^{-\left(\frac{t-t_0}{T-t_0}\right)^{\beta}} * 100\% \qquad (1)$$

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

$$T = \left(\frac{H_{att} - f_{tB} * H_{att10}}{\sqrt[\beta]{-ln\left(1 - \frac{F_o}{100}\right)}} + f_{tB} * H_{att10} \right) * fac \qquad (2)$$

$$t_0 = f_{tB} * H_{att10} * fac \qquad (3)$$

with

$$H_{att10} = \frac{H_{att}}{(1 - f_{tB}) * \sqrt[\beta]{\frac{ln\left(1 - \frac{F_o}{100}\right)}{ln(0,9)}} + f_{tB}} \qquad (4)$$

	factor f_{tB}	Weibull parameter in form of β
Shafts	0.7 to 0.9 (0.8)	1.1 to 1.9 (1.5)
Ball bearing	0.1 to 0.3 (0.2)	1.1
Roller bearing	0.1 to 0.3 (0.2)	1.35
Tooth flank	0.4 to 0.8 (0.6)	1.1 to 1.5 (1.5)
Tooth root	0.8 to 0.95 (0.875)	1.2 to 2.2 (1.8)

Table 2 Factors for a Weibull distribution according to Bertsche, recommended values given in brackets

Equation (1) for R(t) can now be used to display the progression of reliability over time (or number of cycles) as a graphic. The load cycle values t_0 and T can then be calculated after a service life calculation. Equations (2) to (4), using the achievable service life H_{att}, can be used for this purpose.

Calculation of the factors of reliability R(t) according to B. Bertsche with Weibull distribution:
R(t) = 100 * Exp(-((t*fac - t_0)/(T - t_0))^^{b}) %; t in hours (h)

Gear		fac	B	t_0	T	R(H)%
1	Tooth root	1000	1.7	1.667e+007	2.562e+007	82.99
1	Tooth flank	1000	1.3	3.543e+007	1.688e+008	100.00
2	Tooth root	329	1.7	2.416e+006	3.713e+006	0.07
2	Tooth flank	329	1.3	3.826e+007	1.823e+008	100.00

Reliability of calculation at given required service life (%): 0.06 (Bertsche)

Figure 4 List of the factors used in the Weibull equation for calculating reliability

5 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 can 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: if the element fails, we should determine if it directly cause the failure of the entire gear unit or redundancies are 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. Although this appears to be complicated at first glance, it is usually quite straightforward for most gear units: if any one of the vital elements in a standard gear unit (bearing, shaft, gear) breaks, 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:

$$R_S(t) = \frac{R_{C1}(t)}{100} * \frac{R_{C2}(t)}{100} * \ldots * \frac{R_{Cn}(t)}{100} * 100 \text{ or } R_S(t) = 100 * \prod_{i=1}^{n} \frac{R_{Ci}(t)}{100} \quad (5)$$

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

5.1 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 subsystems 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:

$$R_{pair}(t) = \frac{R_{f1}(t)}{100} * \frac{R_{h1}(t)}{100} * \frac{R_{f2}(t)}{100} * \frac{R_{h2}(t)}{100} * 100 \qquad (6)$$

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. So, theoretically, 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 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):

$$R_{pstage}(t) = \frac{R_{f1}(t)}{100} * \frac{R_{h1}(t)}{100} * \left(\frac{R_{f2}(t)}{100} * \frac{R_{h2}(t)}{100}\right)^p * \frac{R_{f3}(t)}{100} * \frac{R_{h3}(t)}{100} * 100 \qquad (7)$$

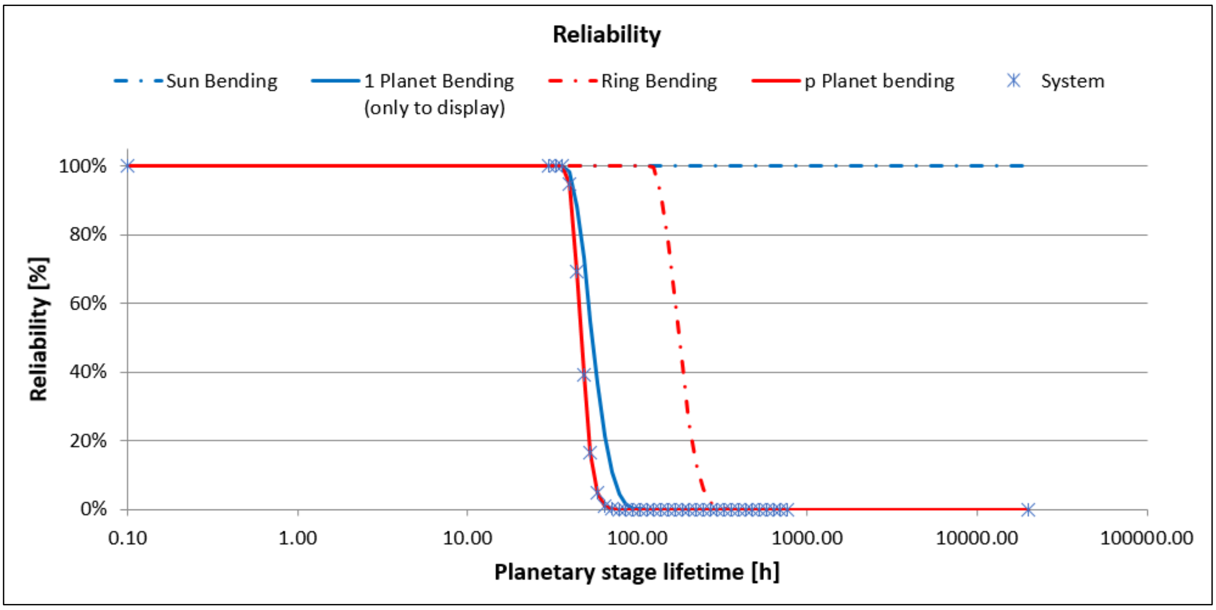
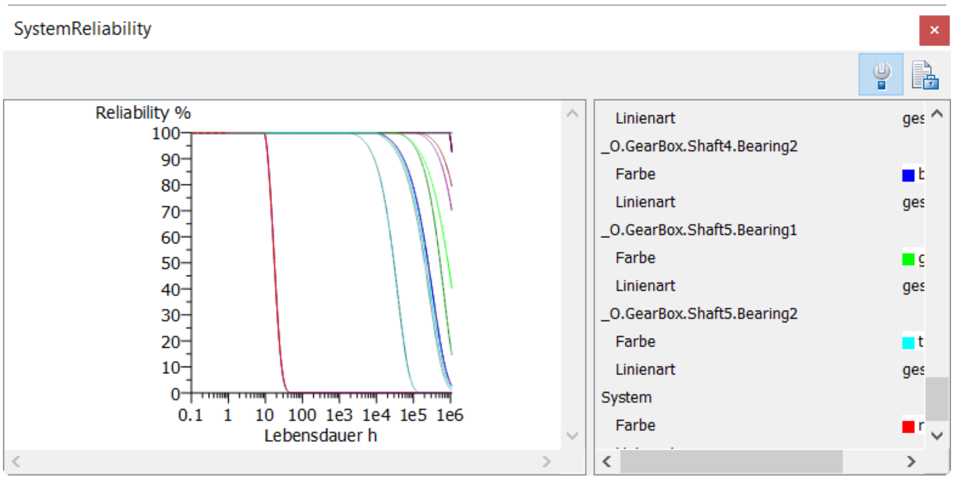


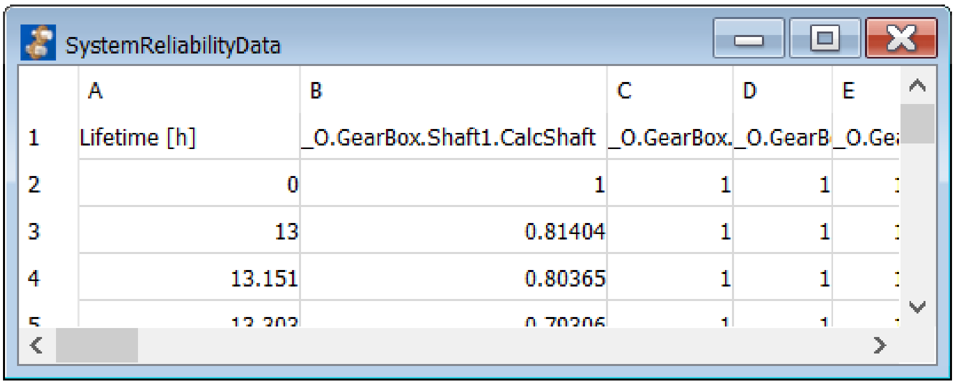
Figure 5 Reliability diagram for a planetary stage with 3 planets: the 3 serially cumulated planets are critical. The system reliability is virtually identical to the reliability of these 3 planets, because the ring and the sun have a significantly higher degree of reliability.

A publication by NASA [[11], Equation 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 why they use the serial formula for the planets.

5.2 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 [10], the achievable service life is also calculated every time a verification is performed. As a consequence, the data for each individual element of the gear unit is automatically available. This data is then forwarded to the KISSsys [10] system program. System reliability can therefore be determined at system level and, if required, a service life reliability diagram can also be displayed (Figure 6). In addition to showing overall reliability, the weakest elements in a gear unit are also clearly displayed in this type of diagram.





	A	B	C	D	E
1	Lifetime [h]	_O.GearBox.Shaft1.CalcShaft	_O.GearBox.	_O.GearB	_O.Ge
2	0	1	1	1	1
3	13	0.81404	1	1	1
4	13.151	0.80365	1	1	1
5	13.303	0.79306	1	1	1

Figure 6 Service life reliability diagram for a 4-stage bevel helical gear unit in KISSsys. (Top: system reliability, displayed with a logarithmic scale. Bottom: table of results.)

In the case of vehicle gear boxes, the calculation for components must be performed with a complex load spectrum which also takes into account the shift setting (shifted gear, time, torque and speed) (Figure 7). 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 situation.

System reliability is of critical importance for gear units used for wind turbines (Figure 8), 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 [1]. AGMA 6006 [6], a US 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.

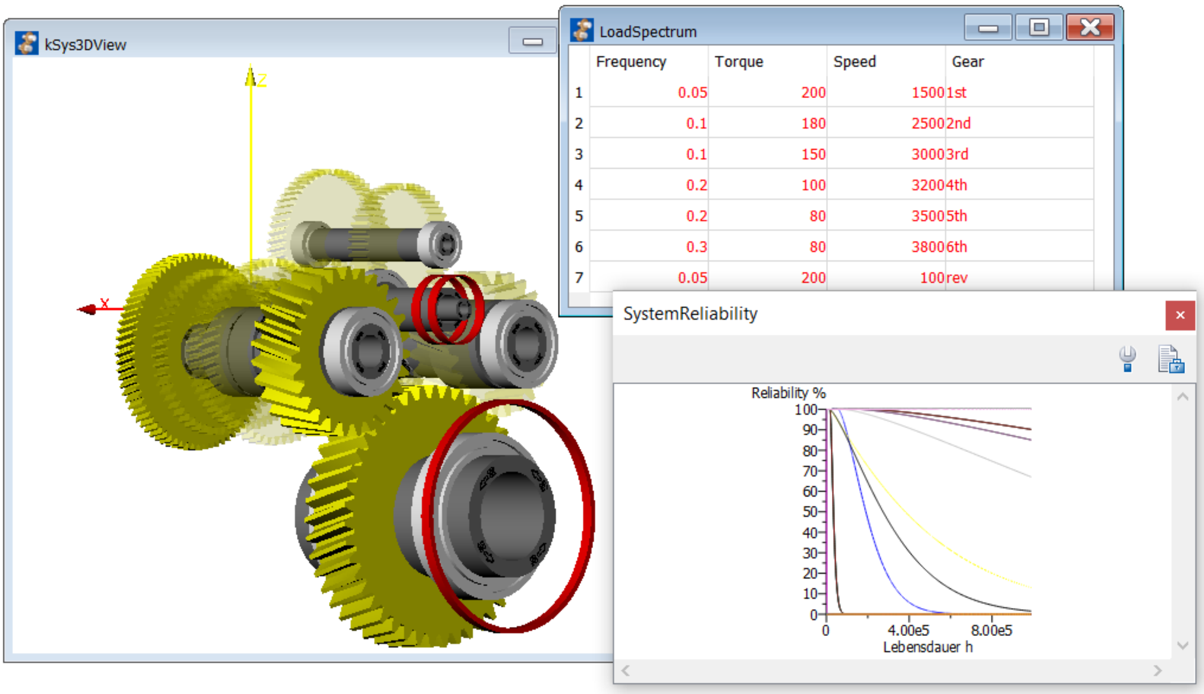


Figure 7 Modern dual clutch gear unit with a load spectrum (6 forward gears and one reverse gear) (System reliability displayed with a linear scale)

As an alternative to displaying safety factors, Figure 9 now shows system reliability and the damage to particular elements. As the safety against pitting of the first stage is inadequate, the result is a damage value of 1010% and a system reliability of 0%. The disadvantage of displaying the data in this way – in contrast to specifying safeties – is that, although it is obvious that elements with a damage of 0% do not cause a problem, it is not possible to see how much reserve they have before they become critical.

A critical note must be added here in conclusion: A comparison of the display of results from a gear unit verification (for example, comparing Figure 1 with Figure 9 and also Figure 6) clearly shows, that actually there is no "optimum way to display the data". A different type of display may be preferable, depending what is actually of interest: an overview, the critical elements, oversized elements etc. For this reason, it makes good sense to provide different ways in which results can be displayed, so that technical experts can select the one they prefer.

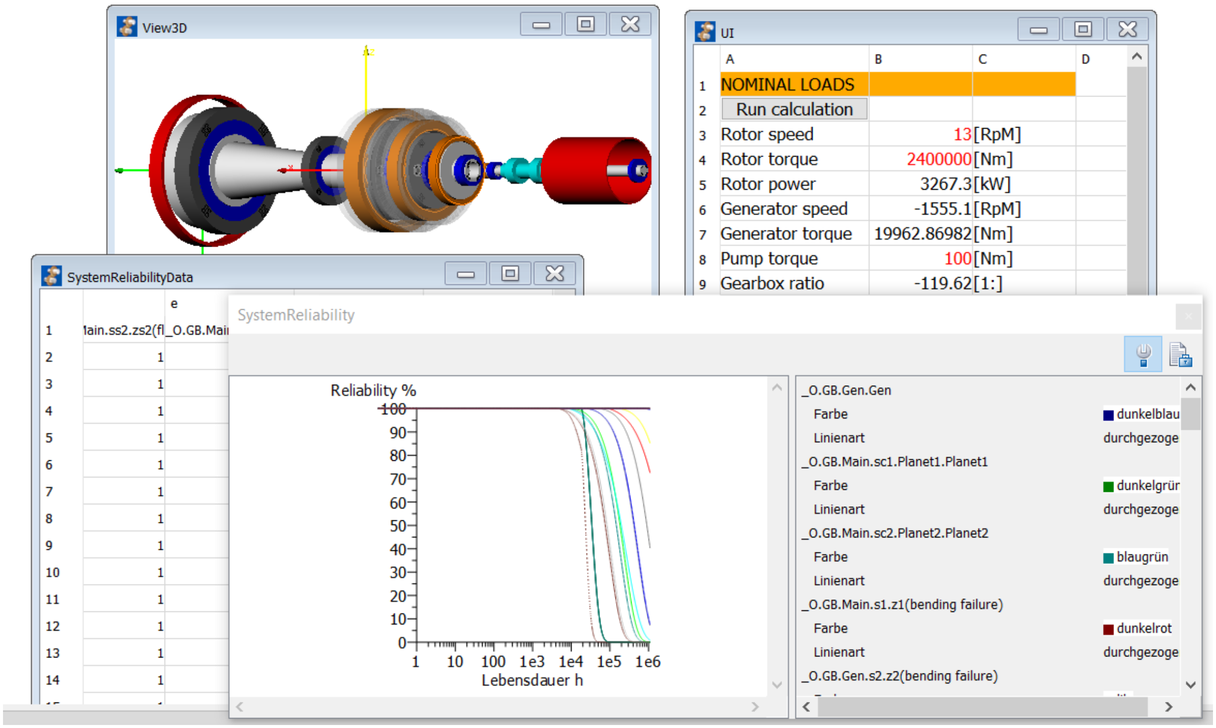
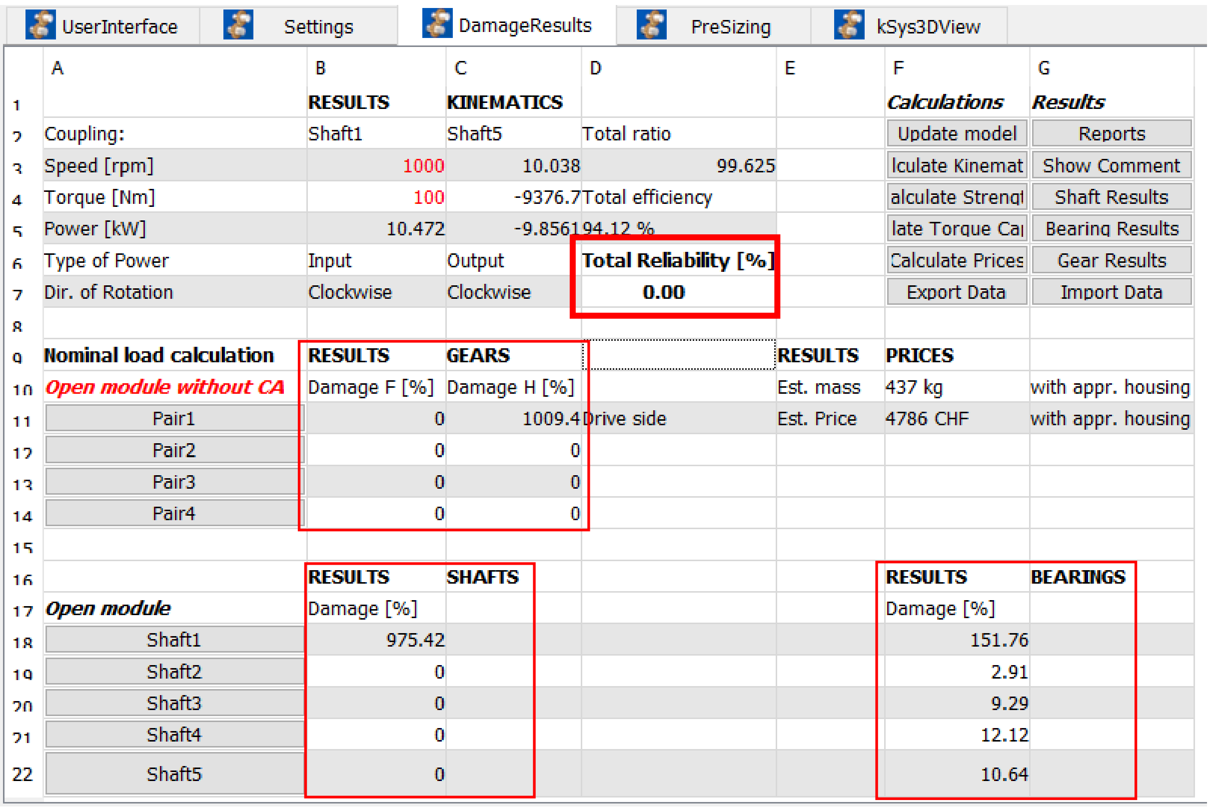


Figure 8 Wind power gear unit with it's reliability



	A	B	C	D	E	F	G
1		RESULTS	KINEMATICS			*Calculations*	*Results*
2	Coupling:	Shaft1	Shaft5	Total ratio		Update model	Reports
3	Speed [rpm]	1000	10.038	99.625		lculate Kinemat	Show Comment
4	Torque [Nm]	100	-9376.7	Total efficiency		alculate Strengt	Shaft Results
5	Power [kW]	10.472	-9.856	94.12 %		late Torque Ca	Bearing Results
6	Type of Power	Input	Output	**Total Reliability [%]**		Calculate Prices	Gear Results
7	Dir. of Rotation	Clockwise	Clockwise	**0.00**		Export Data	Import Data
8							
9	**Nominal load calculation**	**RESULTS**	**GEARS**		**RESULTS**	**PRICES**	
10	***Open module without CA***	Damage F [%]	Damage H [%]		Est. mass	437 kg	with appr. housing
11	Pair1	0	1009.4	Drive side	Est. Price	4786 CHF	with appr. housing
12	Pair2	0	0				
13	Pair3	0	0				
14	Pair4	0	0				
15							
16		**RESULTS**	**SHAFTS**			**RESULTS**	**BEARINGS**
17	***Open module***	Damage [%]				Damage [%]	
18	Shaft1	975.42				151.76	
19	Shaft2	0				2.91	
20	Shaft3	0				9.29	
21	Shaft4	0				12.12	
22	Shaft5	0				10.64	

Figure 9 Display containing an overview of the same results as shown in Figure 1, but with overall reliability. This type of display also has its own problems: although it is clear which elements are critical, there is a complete lack of information about how much elements with damage 0% are oversized.

6 Outlook

Displaying an analysis of a 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 gear box 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 becomes increasingly popular and widespread. However, 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.

AGMA 6006 [6], a US 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 [6] 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 gear box 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 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 is 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 which 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.

Symbols:

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 t_0 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 H_{att} according to Table 1)	%

Literature

[1] Falko, T.; Strasser, D et al.: Determination of the Reliability for a Multi-Megawatt Wind Energy Gearbox; VDI report no.2255, 2015

[2] Bertsche, B.: Reliability in Automotive and Mechanical Engineering; Berlin, Heidelberg: Springer Verlag, 2008

[3] ISO 281, Rolling bearings — Dynamic load ratings and rating life, ISO, 2007

[4] ISO 6336, Part 1-6: Calculation of load capacity of spur and helical gears, ISO, 2006

[5] AGMA 6001-D97: Design and Selection of Components for Enclosed Gear Drives; AGMA, 1997

[6] AGMA 6006-B??: Revision of the Standard for Design and Specification of Gearboxes for Wind Turbines; AGMA, 2003, Restricted document

[7] DIN 743, Tragfähigkeitsberechnung von Wellen und Achsen, 2012. (Calculation of load capacity of shafts and axles)

[8] DIN 3990: Tragfähigkeitsberechnung von Stirnrädern; 1987 (Calculation of load capacity of cylindrical gears)

[9] FKM Guideline, Rechnerischer Festigkeitsnachweis für Maschinenbauteile, 2012

[10] KISSsoft/KISSsys; Calculation Programs for Machine Design; www.KISSsoft.AG

[11] M.Savage, C.A.Paridon, Reliability Model for Planetary Gear Trains, NASA Technical Memorandum, 1982, http://www.dtic.mil/dtic/tr/fulltext/u2/a119165.pdf

[12] ISO/PDTS xxxxx-1: Calculation of tooth flank fracture load capacity of cylindrical spur and helical gears – Part 1: Introduction and basic principles, ISO, 2015

[13] VDI 2230: Systematische Berechnung hochbeanspruchter Schraubenverbindungen (Systematic calculation of highly stressed bolted joints), VDI, 2015