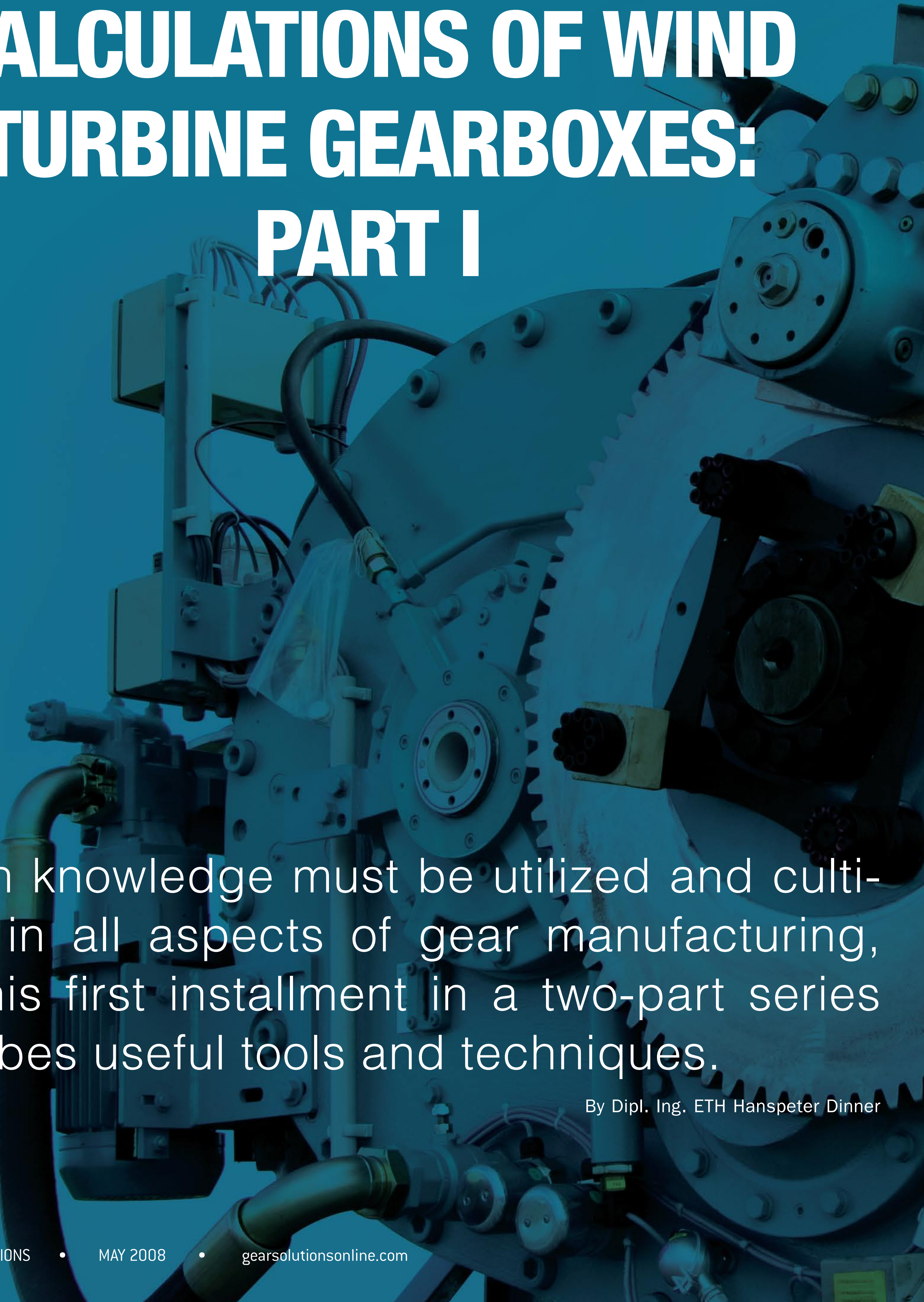
STATIC AND FATIGUE CALCULATIONS OF WIND TURBINE GEARBOXES: PART I



Design knowledge must be utilized and cultivated in all aspects of gear manufacturing, and this first installment in a two-part series describes useful tools and techniques.

By Dipl. Ing. ETH Hanspeter Dinner

1 Abstract

Wind turbine manufacturers have a vital interest in carrying out gearbox calculations on their own, and for multiple reasons:

- Influence on the design proposed by the gearbox supplier
- Plausibility considerations for the calculations submitted
- Quality control in case of less known/experienced suppliers
- Comparison of various standard gearboxes from different providers
- Gearbox recalculation for different load conditions, micrositing

2 Introduction

The wind turbine manufacturer is in the unique position of being able to study gearbox behavior in practice. He can compare field experience against test bench results and calculations. This comparison allows for the adjustment of the theoretical calculations to the hands-on experience and increasing validity of future calculations. These calculations consume considerable resources such as time, knowledge, and tools. The allocation of time is quite difficult because, at the present time, the supply of qualified calculation engineers is rather exhausted. Tools are available, and there is a wide range of commercial solutions being offered. The critical parameter resides in the know-how required to be able to calculate and also standardize to the last detail. This know-how must be respected and cultivated. This article attempts to further the know-how of gear engineers in the wind industry.

2.1 Overview

Using sensitivity analysis, it is investigated how changing the calculations' starting parameters can influence the results, such as service life or strength parameters. The identification of the most-important or less-considered parameters serves the calculation engineer as a guideline for the extension of the existing analysis methodology and calculation standards or helps him check specifications on missing but important data.

To carry out the sensitivity analysis at the gearbox level, a parameterized model of the complete gearbox is used, which includes power flow, component physical distribution, gearing data, shafts, bearings, and shaft-hub connections such as involute splines, keys, or press fit. These sensitivity studies can be carried out automatically and in a very short time with the appropriate programming of the calculation model. The output, in text format, allows for quick processing of the acquired data. This methodology applies to all types of gearboxes.

2.2 Objectives

The engineer is conscious that his assumptions can influence the quality of the calculation results. However, there is only a limited time available to control and/or improve them. That is why he must concentrate on the assumptions, which could bring a really clear improvement to the quality of the calculation. This task is further complicated by the fact that not all assumptions can be appropriately evaluated. Thus, two points must be cleared up:

- Which input values in the calculation can be better evaluated with a minimum of effort (what is the cost of the improvement)?
- How big is the influence of a particular input value upon the result (what brings the improvement)?

Only the second point will be dealt with in the scope of this work.

3 Methodology

3.1 GEARBOX

The 1.5 MW gearbox used in the calculations carried out in this study is based upon suppliers' data after a slight modification (see Figs. 1-2).

3.2 CALCULATION MODEL

With the KISSsys software, commercially available for the past four years, it is possible to display the power flows in the transmission stages and, with a strength calculation, link them to the existing machine components. In this way it is possible to "parameterize" complete gearbox/transmission stages and analyze them in relation to strength and service life. Among other things, KISSsys offers the user the possibility of quickly carrying out a detailed parameterized study of a complete gearbox/transmis-

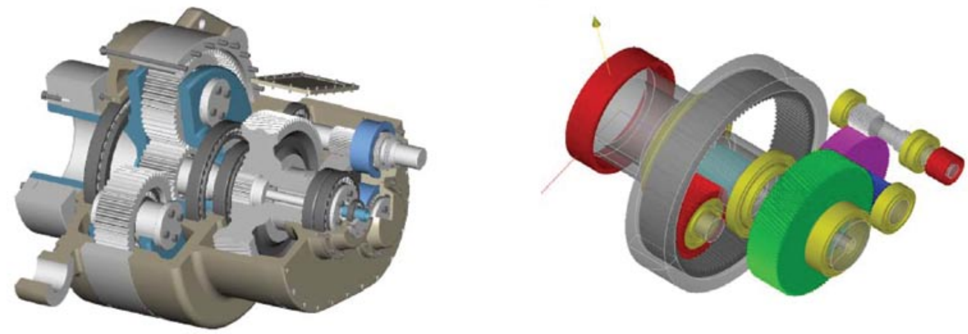


FIG. 1 LEFT: CAD MODEL OF A 1, 5 MW GEARBOX [2]; RIGHT: KISSSYS MODEL OF A 1, 5 MW GEARBOX (SHOWING ONLY ONE PLANET).

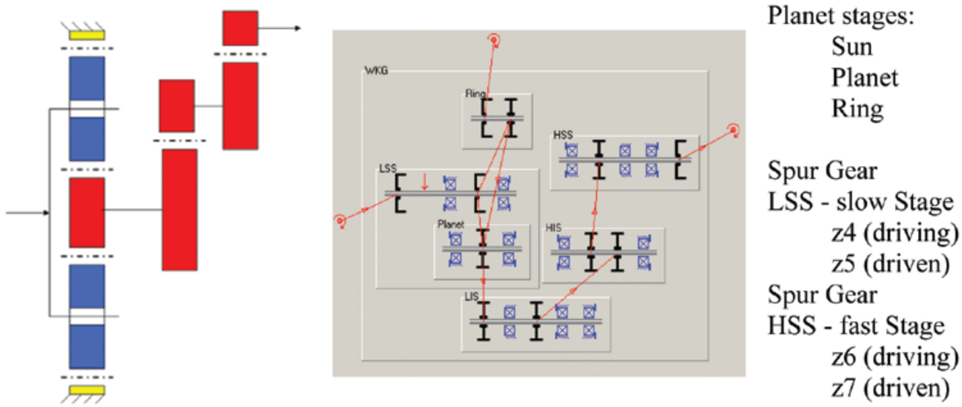


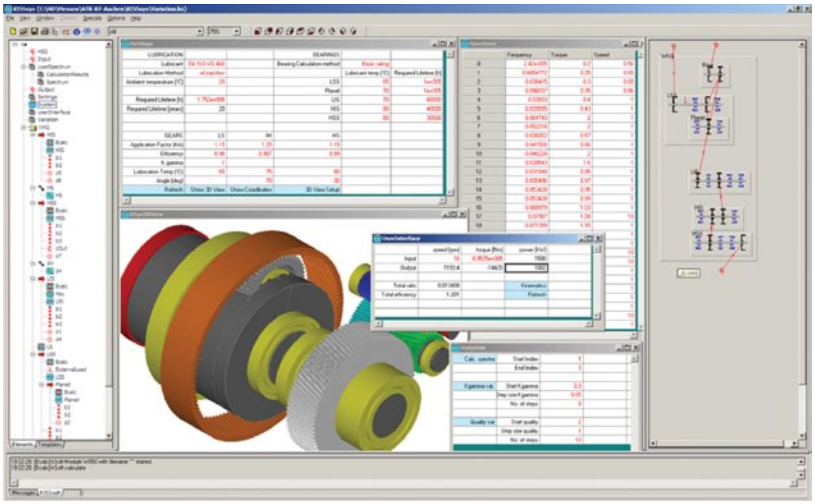
FIG. 2 GEARBOX SCHEMATICS, POWER FLOW IN KISSSYS AND CORRESPONDING NAMES.

sion in order to be able to efficiently compare the several variants of a project design. KISSsys uses KISSsoft to calculate the strength and service life of the different machine components. KISSsoft is a CAE-Software for the quick and safe layout, optimization, and verification of machine components such as gears, shafts, bearings, bolts, shaft-hub connections, and springs. KISSsoft is intended for users in the gearbox production area and is well known for its varied optimization possibilities. The use of KISSsoft for wind turbine gearboxes is described in [6]: Haus der Technik, March 07, "Integrated Layout, Optimization, Verification, and Plan Production for Wind Turbine Gearboxes." KISSsys, as a system complement to KISSsoft, has the following properties (see Fig. 3):

Kinematics Calculation:

- Power flow/rotational speeds with cylindrical, bevel, worm, and crossed axis helical gears
- Modelling of epicyclic drives (Planets, Ravigneaux, Wolfrom…)
- Differential (with bevel or spur gears)
- Chain and belt transmissions
- Clutches can be activated and deactivated, with slippage taken into consideration
- External acting loads are taken into consideration

FIG. 3 KISSSYS MODEL OF A 1.5 MW WIND TURBINE GEARBOX.



Integrated Strength and Service Life calculation:

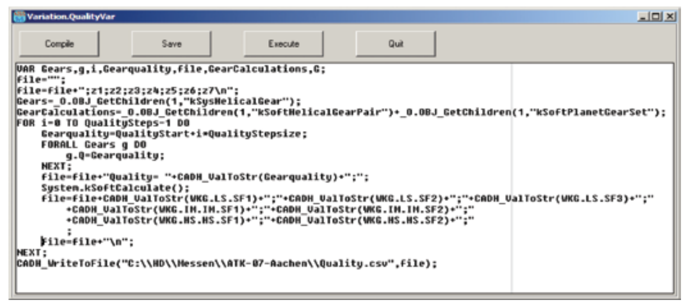
- Here, KISSsys accesses KISSsoft
- Bearing strength, transmission error, profile correction, efficiency 3D model
- Automatic 3D display (based upon the data defined in KISSsoft)
- Export of the 3D model to CAD, import of gearbox casings (step/iges)
- Checking for collisions

Special Features:

- All machine elements in the model, calculated with load spectra
- Several gearbox variants in the same model
- Automatic documentation (strength analysis) for the whole gearbox
- Integrated programming language for implementing special functions

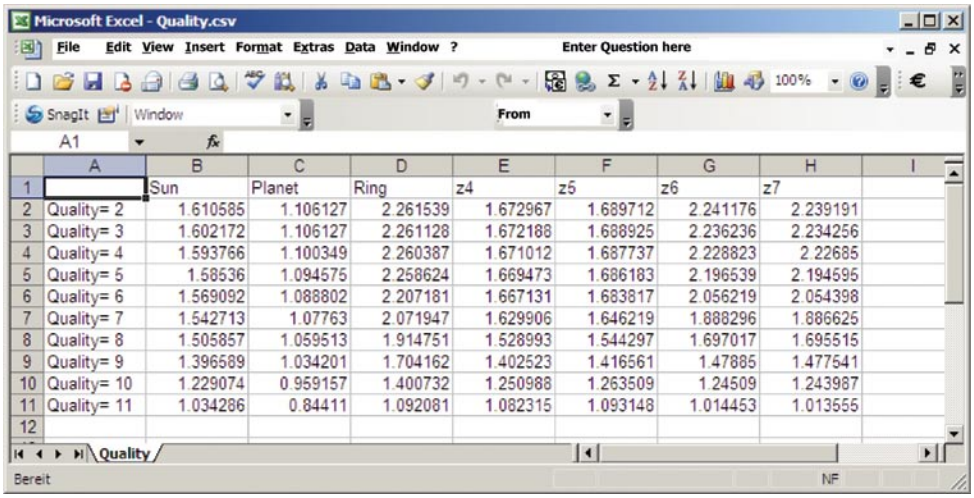
LINE	FUNCTION
1	Local variable definition with the command "VAR"
2	Defines the variable "file" as text type
3	Writes a title line and a line feed in the variable "file"
4	Establishes a "Gears" list which contains all gears available in the KISSsys Model
5	Establishes a list of all gear calculations (spur wheel pairs and planet calculations)
6	Calls a loop which will be executed as many times as desired (Number of runs = Number of gearing qualities)
7	Determines the new gearing quality to be used
8, 9, 10	Attributes the gearing quality to all gears
11	The actual gearing quality is documented in the "file" file
12	The command "System.kSoftCalculate()" executes the strength calculation
13-16	The resultant Safety factors (Root) will be stored in the "file" file, separated by ";".
17	Inserts a line feed
18	The whole procedure will be repeated for the next gearing quality
19	Stores the "file" contents in a *.csv file.

FIG. 4 EXPLANATION OF THE GEARING QUALITY VARIATION FUNCTION.



```
VAR Gears,g,i,Gearquality,File,GearCalculations,G;
File="";
File=File+";z1;z2;z3;z4;z5;z6;z7\n";
Gears=_0.OBJ_GetChildren(1,"kSysHelicalGear");
GearCalculations=_0.OBJ_GetChildren(1,"kSoftHelicalGearPair")+_0.OBJ_GetChildren(1,"kSoftPlanetGearSet");
FOR i=0 TO QualitySteps-1 DO
    Gearquality=QualityStart+i*QualityStepsize;
    FORALL Gears g DO
        g.Q=Gearquality;
    NEXT;
    File=File+"Quality= "+CADH_ValToStr(Gearquality)+";";
    System.kSoftCalculate();
    File=File+CADH_ValToStr(WKG.LS.SF1)+";"+CADH_ValToStr(WKG.LS.SF2)+";"+CADH_ValToStr(WKG.LS.SF3)+";"
        +CADH_ValToStr(WKG.IM.IM.SF1)+";"+CADH_ValToStr(WKG.IM.IM.SF2)+";"
        +CADH_ValToStr(WKG.HS.HS.SF1)+";"+CADH_ValToStr(WKG.HS.HS.SF2)+";"
        ;
    File=File+"\n";
NEXT;
CADH_WriteToFile("C:\\HD\\Hessen\\ATK-07-Aachen\\Quality.csv",File);
```

FIG. 5 KISSSYS GEARING QUALITY VARIATION FUNCTION.



A	B	C	D	E	F	G	H
	Sun	Planet	Ring	z4	z5	z6	z7
Quality= 2	1.610585	1.106127	2.261539	1.672967	1.689712	2.241176	2.239191
Quality= 3	1.602172	1.106127	2.261128	1.672188	1.688925	2.236236	2.234256
Quality= 4	1.593766	1.100349	2.260387	1.671012	1.687737	2.228823	2.22685
Quality= 5	1.58536	1.094575	2.258624	1.669473	1.686183	2.196539	2.194595
Quality= 6	1.569092	1.088802	2.207181	1.667131	1.683817	2.056219	2.054398
Quality= 7	1.542713	1.07763	2.071947	1.629906	1.646219	1.888296	1.886625
Quality= 8	1.505857	1.059513	1.914751	1.528993	1.544297	1.697017	1.695515
Quality= 9	1.396589	1.034201	1.704162	1.402523	1.416561	1.47885	1.477541
Quality= 10	1.229074	0.959157	1.400732	1.250988	1.263509	1.24509	1.243987
Quality= 11	1.034286	0.84411	1.092081	1.082315	1.093148	1.014453	1.013555

FIG. 6 RESULTING *.CSV IN EXCEL.

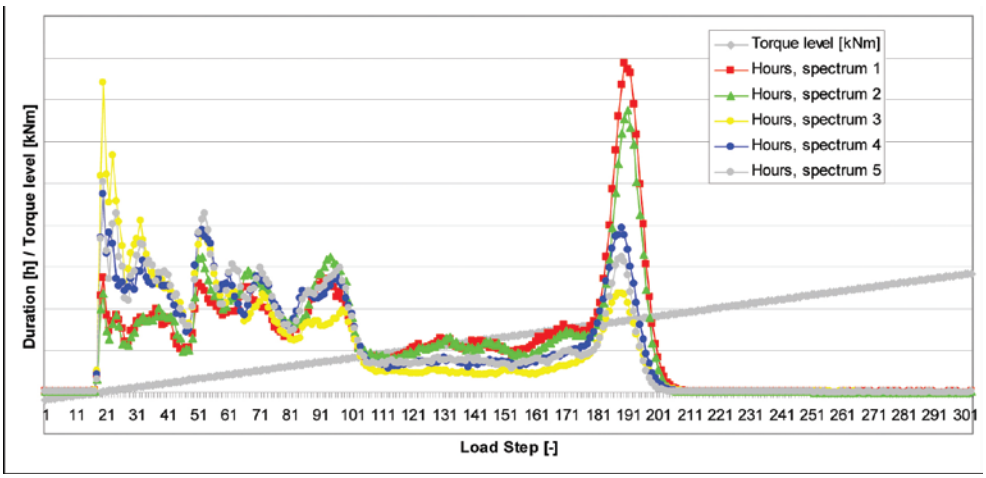


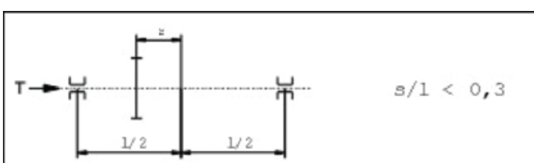
FIG. 7 FIVE LOAD SPECTRA WITH THE SAME NOMINAL TORQUE.

CASE	CALCULATION METHODOLOGY FOR Y_F, Y_S	REMARKS
HR1	Y_F and Y_S according to ISO6336:1996	Uses a 30° Tangent / replacement Rack
HR2	Y_F and Y_S according to ISO6336:2006	Uses a 60° Tangent / Shaping Cutter
HR3	Y_F and Y_S according to graphics methodology	Tooth Profile based on Suppliers' Simulation

FIG. 8 TREE METHODOLOGIES FOR RING WHEEL CALCULATION.

CASE	DYNAMIC FACTOR VALUE	REMARKS
$K_{H\beta}$ 1	Fix 1.05 for all spectrum steps	
$K_{H\beta}$ 2	Fix 1.15 for all spectrum steps	
$K_{H\beta}$ 3	Fix 1.25 for all spectrum steps	
$K_{H\beta}$ 4	Fix 1.35 for all spectrum steps	
$K_{H\beta}$ 5	Variable for each load step	According to ISO 6336, B
$K_{H\beta}$ 6	Fix 1.44 for all spectrum steps	According to ISO 6336, B, at nominal load

FIG. 9 SIX Kγ CASES.



l=600
s=150
dsh=150
With helix crowning and helix angle correction

FIG. 10 CALCULATION DATA FOR THE LOAD-DEPENDENT KHγ ACCORDING TO ISO 6336, METHOD B.

3.3 PARAMETER VARIATIONS, EDITION OF THE RESULTS

KISSsys has an object-oriented programming language allowing for control of the calculations, which can read data from text files and export the results, for instance, to Excel. It is thus possible to automate parameter variations and swiftly execute them.

It is shown in Fig. 4 how such a parameter variation can be programmed in KISSsys. The function in the example shows how, for an established starting value and a defined number of steps, the gearing quality varies and, how the gearing is verified for nominal load for each condition. The resulting safety factors, separated by ";", are written to a file and can be displayed in Excel (see Fig. 5). The sequence of operations is shown in Figs. 4, 5. Excel creates a file "Quality.csv" displaying the information seen in Fig. 6.

4 Influencing Values

4.1 DIFFERENT LOAD SPECTRA WITH THE SAME T_N

The gearbox will be calculated for five different load spectra (e.g., for different locations of the wind turbines), but for an equal nominal torque of about 800 kNm. The rotor speed for all stages stays constant at 16 rpm. The influence on the resulting root and flank safety has to be investigated (see Fig. 7).

4.2 RING GEAR CALCULATION

One of the known weaknesses of the 1996 edition of the ISO 6336:1996 (the DIN 3990 has the same problem) is the calculation of the tooth-root stress for ring gears. This is now calculated in a completely different way, in which the tooth profile is determined by the cutter wheel used for the manufacturing. With it, there are more practical data (force application point, lever arm tooth root cross section, and rounding radius) than by the previous assumptions for the replacement rack. The tooth profile values and the stress correction factors Y_F, Y_S change with it in the new edition of ISO 6336:2006. In the graphical method, the factors Y_F, Y_S are calculated along the whole root, which is a more precise method to calculate the root strength as the highest resulting stress is considered (see Fig. 8).

4.3 K FACTORS

A summary of K factor values to be used, according to several guidelines and standards, is given in [4]: Antriebstechnik 5/2006-Participating Dialog as a Success Solution, Gear Calculation for Wind Turbine Gear Boxes. Here, selected K factors will be modified. Especially interesting is the comparison between the two following cases:

- For each load spectrum step, a K factor will be separately calculated/modified.

CASE	DYNAMIC FACTOR VALUE	REMARKS
K_γ 1	Fix 1.00 for all spectrum steps	According to GL Guidelines
K_γ 2	Fix 1.10 for all spectrum steps	According to EC 61 400
K_γ 3	Fix 1.20 for all spectrum steps	According to AGMA 61 23 (1.23)
K_γ 4	Fix 1.25 for all spectrum steps	In comparison with K_γ 5
K_γ 5	Variable between 1.25 and 1.00	According to ISO 6336, B

FIG. 11 FIVE DIFFERENT $K_{H\beta}$ CASES.

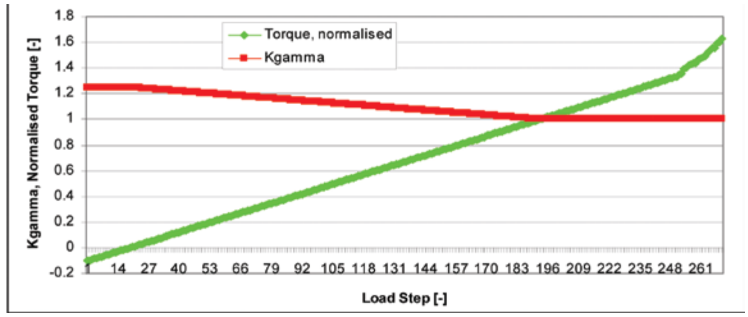


FIG. 12 Kγ CASE 5: $K_{H\beta}$ VARIATION WITH THE LOAD SPECTRUM.

- The same K factor will be kept as a fix for each load spectrum step, typically at the value issued from the calculation with the nominal load.

4.3.1 UNIFORM LOAD FACTOR $K_{H\beta}$

Calculation of gear safeties for six different assumptions of $K_{H\beta}$ (see Fig. 9). In the fifth case ($K_{H\beta}$ 5), $K_{H\beta}$ will be separately calculated for each individual load step according to ISO 6336, Method B. In this case the values used are displayed in Fig. 13 for the fast stage considered. In the sixth case ($K_{H\beta}$ 6), the calculation is carried out in comparison to a fix $K_{H\beta}$ value, determined in the verification for a nominal load.

4.3.2 LOAD DISTRIBUTION FACTOR Kγ

Various guidelines, standards, and specifications from wind turbine manufacturers consider different implicit load distribution factors Kγ depending upon the number of planets. Measurements are documented, for instance, in [3] and [5]. The comparison is carried out with different Kγ values coming from different standards and guidelines. Of particular interest is the comparison between cases Kγ 4 and Kγ 5, between a value set by the spectrum as a constant and a spectrum variable value (according to Fig. 12).

4.3.3 DYNAMIC FACTOR K*v*

This will be calculated according to ISO 6336, but following pertinent regulations must not be less than 1.05. The fast stage will be studied in the following cases. Again of interest is the comparison between cases K*v* 5 and K*v* 6; i.e. one with a value set by the spectrum as fix and one with a separately calculated K*v* value for each load step.

4.4 S-N CURVE MODIFICATIONS, Z_{NT} AND Y_{NT}

As for the calculations, the S-N curve can be modified in terms of three different levels of endurance limit. For the so-called

CASE	DYNAMIC FACTOR VALUE	REMARKS
Kv 1	Fix 1.00 for all spectrum steps	
Kv 2	Fix 1.05 for all spectrum steps	
Kv 3	Fix 1.10 for all spectrum steps	
Kv 4	Fix 1.15 for all spectrum steps	
Kv 5	Fix 1.20 for all spectrum steps	
Kv 6	Individually calculated for each step (according to ISO 6336)	

FIG. 13 CALCULATED *Kv* CASES FOR THE FAST STAGE.

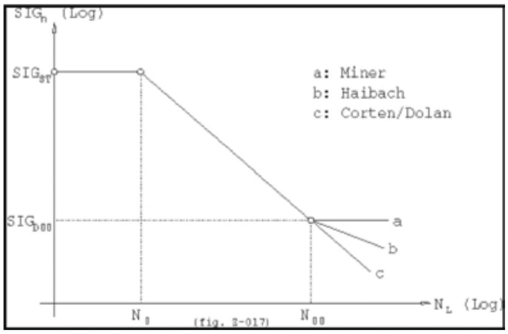


FIG. 14 S-N CURVE MODIFICATIONS FOR THE FAST STAGE A): MINER ORIGINAL, B): HAIBACH, C) MINER ELEMENTARY.

Haibach modification, the fatigue limit line, with approximately half inclination (2k-1), continues after the first inflexion point. With this the loads below the endurance limit are also considered, and the calculated service lives will stay lower than with the original S-N curve lines (see Fig. 14).

Additionally, the gear service life and the root and the flank safety factors are examined for different material qualities (influence of Z_{NT} and Y_{NT} for 10^{10} cycles). Different S-N curves result from this can be seen in Fig. 15. The calculation will be carried out once with a load spectrum and once with a nominal load.

4.5 GEARING QUALITY

The gearing safety factors are calculated for different gearing qualities. The range of qualities considered in DIN 2 to 11 cover the normal quality ranges very generously.

4.6 Kγ INFLUENCE ON THE PLANET BEARING

The Kγ factor is used in the calculation of the planet stage gearing. Since it represents a system variable it will also have to be considered in the calculation of bearings (also in the calculation of the planet bolts). It should vary from 0.90 to 1.25 in steps of 0.05. Values below 1.00 should show in how much the calculated planet bearing service life will change in the less loaded path. Values greater than 1.00 are relevant, for instance, for solutions with more than three planets.

4.7 BEARING STIFFNESS INFLUENCE ON THE PLANET BEARINGS

Should more than two bearings be used for the support of the planets, and these have helical gearing, the tilting torque will be spread over them depending upon the bearing stiffness. The forces acting on the bearing in case of helical gearing consist of the circumferential forces (transmitting the planet torque) as well as the planet tilting torque. This tilting torque produces additional forces on the planet bearings depending upon the number of bearings, their span and stiffness. The stiffness are assumed as infinitely high and arithmetically estimated. The arithmetical estimation is then increased or reduced by one order of magnitude in order to find in what extent an error affects the stiffness estimation (see Figs. 17, 18).

4.8 DAMAGE DISTRIBUTION: GEARING

It should display which load spectrum steps contribute to the total damage. Should it be

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determined that certain steps do not provoke damage they could, for instance, be neglected in a test bench essay.

4.9 DAMAGE DISTRIBUTION: BEARINGS

Same objectives for the gearing see Chapter 4.8. The conclusion of this article will appear in the June 2008 issue of Gear Solutions magazine. Both parts will be downloadable at [www.gearsolutionsonline.com].

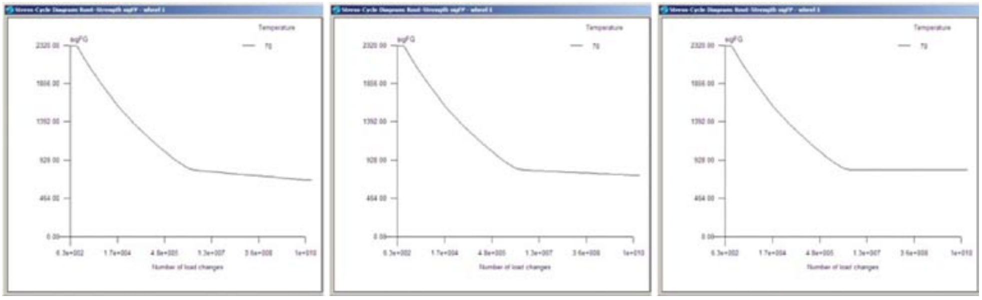

FIG. 15 (FORMERLY FIGURE 0-2) S-N CURVES (ROOT, 18CRNIMO7-6), FOR MATERIAL QUALITIES “ML,” “MQ,” AND “ME.”

CA	CALCULATED WITH LOAD SPECTRUM	
NT 1	ML Quality, $Z_{NT} = Y_{NT} = 0.85$ Calculated with load spectrum	According to ISO 81400
NT 2	MQ Quality, $Z_{NT} = Y_{NT} = 0.92$ Calculated with load spectrum	High Material Quality
NT 3	ME Quality, $Z_{NT} = Y_{NT} = 1.00$ Calculated with load spectrum	Highest Material Quality
NT 4	Haibach modification Calculated with spectrum	
NT 5	ML Quality, $Z_{NT} = Y_{NT} = 0.85$ Calculated with nominal load	According to ISO 81400
NT 6	MQ Quality, $Z_{NT} = Y_{NT} = 0.92$	High Material Quality
NT 7	ME Quality, $Z_{NT} = Y_{NT} = 1.00$ Calculated with nominal load	Highest Material Quality
NT 8	Haibach modification Calculated with nominal load	

FIG. 16 STUDY CASES: ZNT AND YNT VALUE MODIFICATION FOR 1010 CYCLES.

References:

1) R. Grzybowski, B. Niederstucke, Betriebsfestigkeitsberechnung von Getrieben in Windenergieanlagen mit Verweildauerkollektiven, Allianz Report 2004

2) R. Poore, T. Lettenmaier, Alternative Design Study Report: Wind PACT Advanced Wind Turbine Drive Train Designs Study, NREL/SR-500-33196

3) U. Giger, G.P. Fox, Leistungsverzweigte

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CASE	BEARING STRENGTH	REMARKS
LS1	Unlimited high Strength	
LS2	934 N/μm	Calculated
LS3	93.4 N/μm	1/10
LS4	9349 N/μm	*10

FIG. 17 VARIOUS BEARING STIFFNESSES TO CALCULATE THE STATIC CONFORMITY OF A PLANET SUPPORT.

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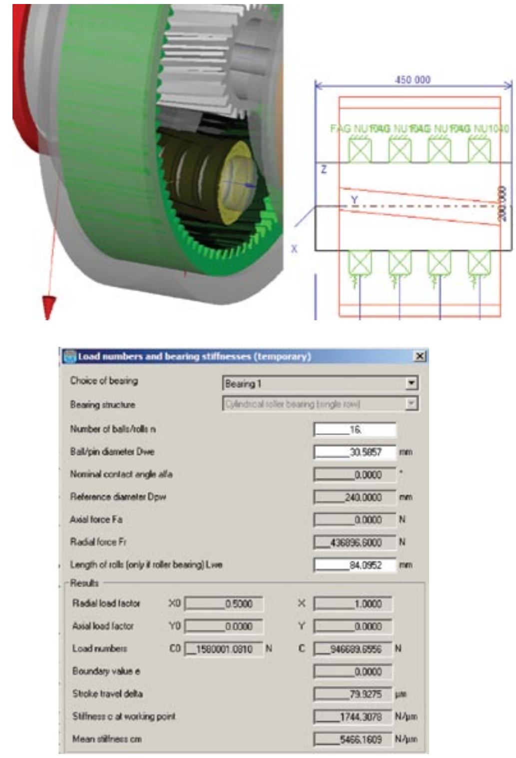

FIG. 18 TOP: PLANET SUPPORT FIRST STAGE. CENTER: SUPPORT WITH FOUR BEARINGS WITH STIFFNESS. BOTTOM: BEARING STIFFNESS CALCULATION IN KISSSOFT.

Planetengetriebe in Windenergieanlagen mit flexibler Planetenlagerung, ATK03
4) H. Dinner, Gleichberechtigter Dialog als Erfolgsrezept, Verzahnungsberechnung für Windenergieanlagengetriebe, Antriebstechnik 5/2006 – [Participating Dialog as a Success Solution, Gear Calculation for WTIs]
5) F.D.Krull, T. Siegenbruck, Windenergieanlagen fordern hohe Leistungsdichten, Ermittlung der Breitenlastverteilung in Planetengetrieben, Antriebstechnik 9/2004
6) H. Dinner, Integrierte Auslegung, Optimierung, Nachrechnung und Zeichnungserstellung von Verzahnungen für Windkraftgetriebe, Antriebsstränge in Windenergieanlagen, Haus der Technik, März 07

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