ICMDT 2007

JSME-KSME Joint International Conference on Manufacturing, Machine Design and Tribology

Assumptions in Engineering Analysis of Wind Turbine Gearboxes and its Effects on the Design

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Abstract

Wind turbine companies have a vital interest to perform engineering analysis on the gearboxes themselves. Furthermore, they are in the positions to tune the analysis fine until it does better model the real behaviour of the gearboxes using the comparisons between theortical analysis and field experiences. To allow for the comparisons, a standardised and well defined analytical procedures are required. These analytical investigations require several critical resources, i.e., time, know-how and tools. While tools are available on the marked, time and know-how are sparse. Especially the detailed know-how about how to perform such analysis are hard to find as there are not many engineers with the required background for hire.

In this paper, it is investigated which parameters in the analysis process carry higher relevance than others using sensitivity analysis on a gearbox system level. It will be illustrated which are the critical issues that should be known for a state of the art of static strength and fatigue analaysis of a wind turbine gearbox.The identification of relevant parameters will allow the wind turbine designer (who is purchasing gearboxes) to further detail his specifications for the gearboxes.

In the same manner, different currently used standards will be compared. It will be shown how the selection of a particular standard may influence the strength and life rating of a gearbox, hence influencing the overall design and cost. This selection typically leads to rather dramatic changes in the results and has to be executed with great care. A comparison of the variation of the results with the variation of input parameters will then identify those parameters most relevant in the design and analysis of wind turbine gearboxes.

Key words:

Wind Turbine Gearbox, Strength Analysis, Fatigue Analysis, Sensitivity Analysis, Design Parameters

1. Introduction

Wind turbine manufacturers do have a vital interest in carrying out gearbox calculations on their own from the multiple reasons: Influence on the design, Plausibility considerations for the calculations achieved, Quality control made by less experienced suppliers, Comparison of various gear boxes from different providers, Calculation for different load conditions.

Furthermore, the wind turbine manufacturer is in a unique position, that of being able to study gearbox behaviour using field experience with test bench results and calculations. This comparison allows the adjustment of the theoretical calculations to the hands-on experience and the increasing of the validity of future calculations. These calculations consume resources: time, knowledge and tools. The allocation of the resource time is quite difficult because, for the time being, the market of qualified calculation engineers is rather exhausted. Tools are available; there is a wide range of commercial solutions being offered. The critical parameter resides in the know-how to be able to calculate and also standardize to the last detail. This know-how must be cultured and with-honoured in accordance.

Using sensitivity analysis it is investigated, how changing the calculation' starting parameters can influence the results (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 plant regulations and calculation standards or helps him/her checking specifications on missing but important data. The engineer is conscious that his/her 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/she must concentrate on the assumptions which could bring a really clear improvement in the quality of the calculation. This task is further complicated by the fact that not all assumptions can be appropriately evaluated. Thus,

The 1.5 MW gearbox shown in Fig. 1 will be used in the calculations, and is based upon suppliers' data after a slight modification. The gearbox is consist of single stage planetary gear, and double stage spur gears (LSS - slow stage, HSS - fast stage).

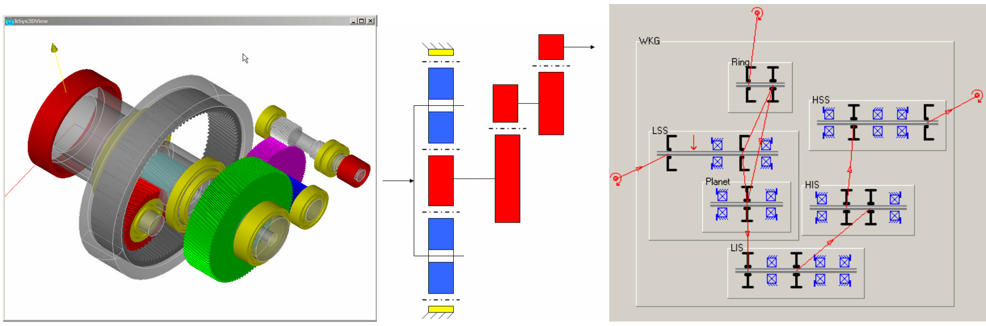

Fig. 1 Analysis model of a 1, 5 MW gearbox and its power flow

2. Methodology and Influencing Values

2.1 Different load spectra with the same nominal torque

The gearbox will be calculated for five different load spectra shown in Fig. 2 (different caracteristics) but for an equal nominal torque of about 800 kNm. The rotor speed for all

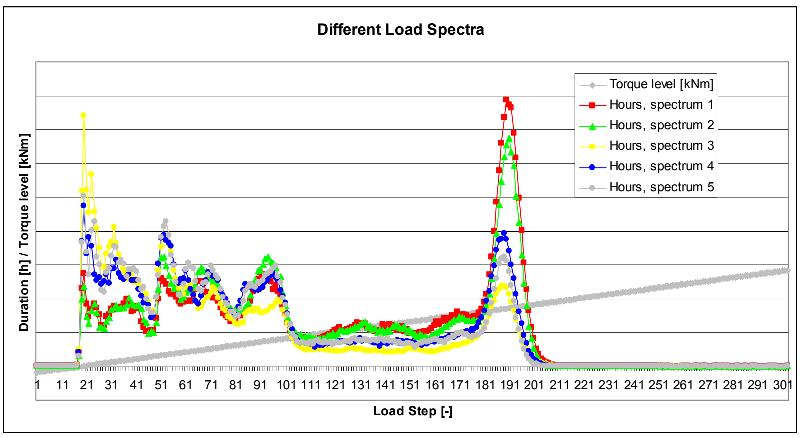

Fig. 2 Five load spectra with the same nominal torque

stages stays constant at 16 rpm. The influence on the resulting root and flank safety has to be investigated.

2.2 Ring gear calculation

One of the known weaknesses of the latest edition of the ISO 6336: 1966 (also the DIN 3990) is the calculation of the tooth-root stress for ring gears. This is now calculated in a completely different way in the new edition of ISO 6336:2006, 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, critical 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 graphical method proposed by Obsieger and improved by Kissling, 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.

Table 1 Three methodologies for ring gear calculation

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 graphical method	Tooth profile based on simulation

2.3 Uniform load factor $K_{H\beta}$

A summary of *K* factor values to be used, according to several guidelines and standards is given in [4]. Especially interesting is the comparison between the two following cases:

- For each load spectrum step, a *K* factor will be separately calculated / modified.
- The same *K* factor will be kept as a fixed value for each load spectrum step.

In the fifth case ($K_{H\beta}5$) of the calculation of gear safeties for six different assumptions, $K_{H\beta}$ will be separately calculated for each individual load step according to ISO 6336, Method B. Table 2 shows the values for the fast stage (HSS) considered. In the sixth case ($K_{H\beta}6$), the calculation is carried out in comparison to a fixed $K_{H\beta}$ value, determined in the verification for a nominal load.

Table 2 Six $K_{H\beta}$ cases

Case	Uniform load factor value	Remarks
$K_{H\beta}1$	1.05 for all spectrum steps	
$K_{H\beta}2$	1.15 for all spectrum steps	
$K_{H\beta}3$	1.25 for all spectrum steps	
$K_{H\beta}4$	1.35 for all spectrum steps	
$K_{H\beta}5$	Variable for each load step	According to ISO 6336-B
$K_{H\beta}6$	1.44 for all spectrum steps	According to ISO 6336-B, at nominal load

2.4 Load distribution factor K_γ

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

Table 3 Five different K_γ cases

Case	Load distribution factor value	Remarks
$K_\gamma 1$	1.00 for all spectrum steps	According to GL Guidelines
$K_\gamma 2$	1.10 for all spectrum steps	According to EC 61 400
$K_\gamma 3$	1.20 for all spectrum steps	According to AGMA 61 23 (1.23)
$K_\gamma 4$	1.25 for all spectrum steps	In comparison with $K_\gamma 5$
$K_\gamma 5$	Variable between 1.25 and 1.00	According to Fig. 3

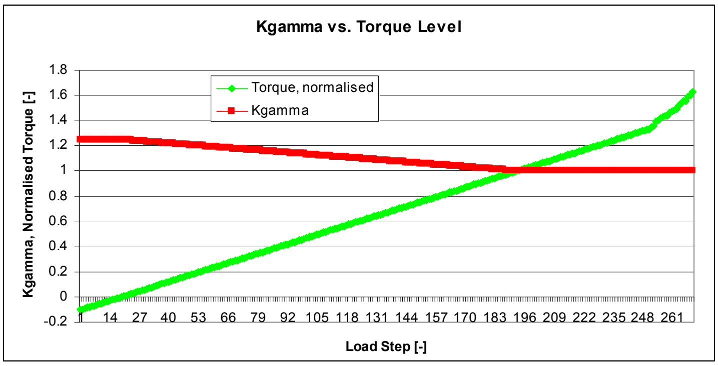

Fig. 3 K_γ variation with the load spectrum

2.5 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 (HSS) will be studied in the following cases. Again interesting is the comparison between cases K_v5 and K_v6, 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.

Table 4 Calculated K_v cases for the fast stage (HSS)

Case	Dynamic factor value	Remarks
K_v1	1.00 for all spectrum steps	
K_v2	1.05 for all spectrum steps	
K_v3	1.10 for all spectrum steps	
K_v4	1.15 for all spectrum steps	
K_v5	1.20 for all spectrum steps	
K_v6	Individually calculated for each step (according to ISO 6336)	

2.6 Gear damage distribution

It should display which load spectrum steps contribute to the total damage. Should it be determined that certain steps do not provoke damage, they could, for instance, be neglected in a test bench essay.

3. Results

3.1 Different load spectra with the same nominal torque

A change in the spectrum constitution results in a change of the gearing safety of approximately 5% (see Figs. 4 ~ 6). This is lower than expected but, once again, it depends upon the selection of the spectrum to be used. However, the changes in the bearing service lives are dramatic. This shows how questionable the bearing service life calculation is and that the results are another indication that it would also be advisable to calculate a bearing safety factor based upon a stress. This is possible, for instance, with the calculation of an acting surface pressure as in ISO 81400.

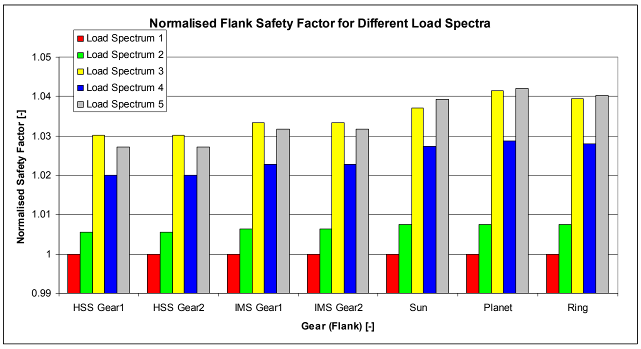

Fig. 4 Normalised flank safeties for various load spectra

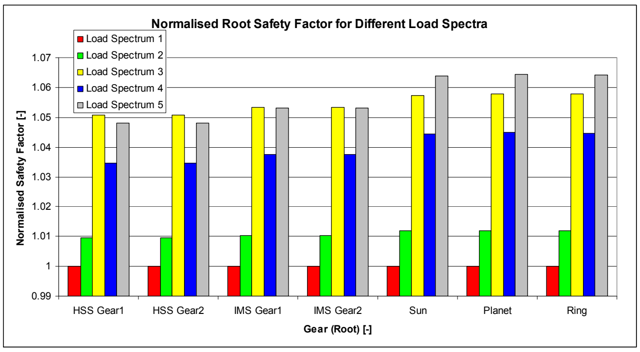

Fig. 5 Normalised root safeties for various load spectra with same nominal torque

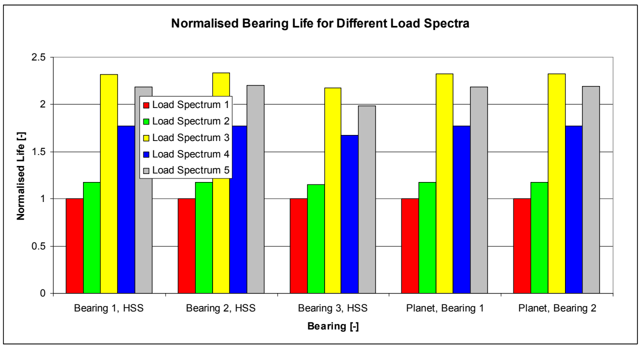

Fig. 6 Normalised bearing service life for various load spectra with same nominal torque

3.2 Ring gear calculation

It appears a modest ring gear root strength deviation for the three calculation methodologies which naturally has dramatic effects upon the calculated service life. This deviation strongly depends upon the tooth height, the pressure angle and the addendum modification. In the calculations show here slightly higher pressure angle and tooth height were used.

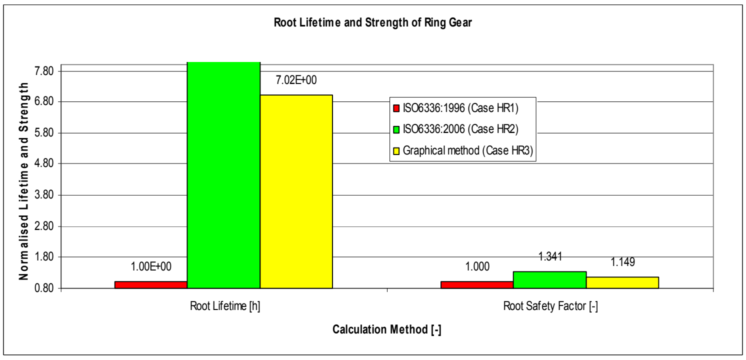

Fig. 7 Resulting ring wheel safeties for various calculation methodologies

3.3 Uniform load factor $K_{H\beta}$

Naturally, the root and the flank safeties vary with the $K_{H\beta}$ value in a linear/exponential way. However, it is interesting that the difference between calculations with a constant $K_{H\beta}$ for all spectrum steps (Case $K_{H\beta}6$ with a fix $K_{H\beta}$ considered at nominal load) gives identical results as when $K_{H\beta}$ varies. That is, the choice of procedure (both are contained in the guidelines) does not show any influence worth mentioning. This is confirmed by a similar study in [1].

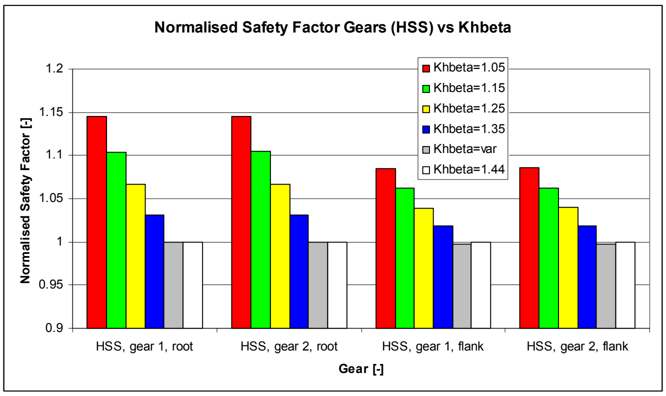

Fig. 8 Root- and flank safeties as a function of $K_{H\beta}$: fast stage (HSS).

3.4 Load distribution factor K_γ

Also here, it is especially interesting the comparison of the last two cases, fixed K_γ against variable K_γ. Once again, there is no difference worth mentioning, i. e., there is no need to consider / set K_γ separately for all spectrum steps.

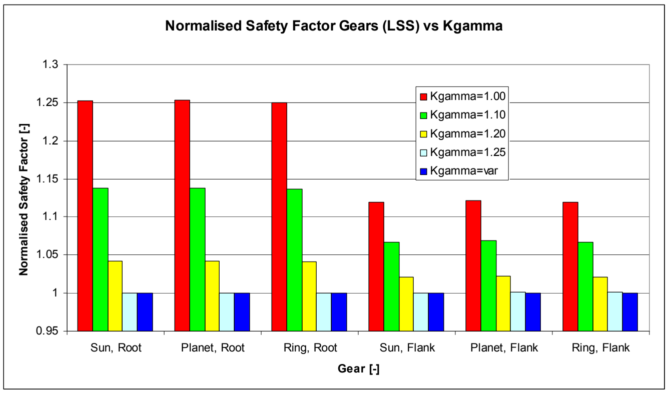

Fig. 9 Root- and flank safeties in the planet set.

3.5 Dynamic factor K_v

The same tendency is also valid for the dynamic factor K_v. Whether separately calculated for all spectra or, set as a constant value at nominal load, there are negligible differences in the resulting safeties.

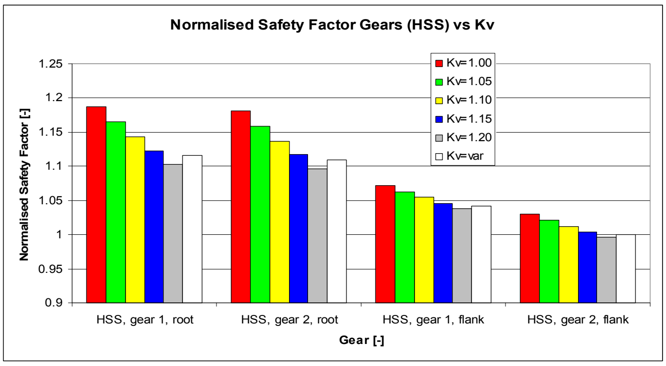

Fig. 10 Root and flank safety factors: fast stage (HSS)

3.6 Gear damage distribution

From the gearing damage calculation for all gears, the individual damage is D_i ($\Sigma_i(D_i)=1$), separate for tooth root and flank. It is evident that for the highly stressed gears (thus, not the ring gear), the damages originate in very few spectrum steps, representing about 10% of the total. The load spectra being not very informative in many areas, there is need for action here. In such cases the gearing manufacturer must insist on the need for a load spectra study to be carried out. The objective must be grouping together the non damage-relevant areas and study in detail the load levels in the relevant areas (typically around nominal loads).

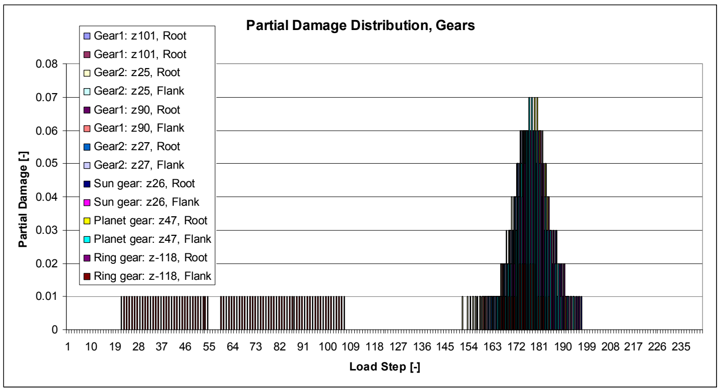

Fig. 11 Damage distribution for all gears, root and flank: all load stages

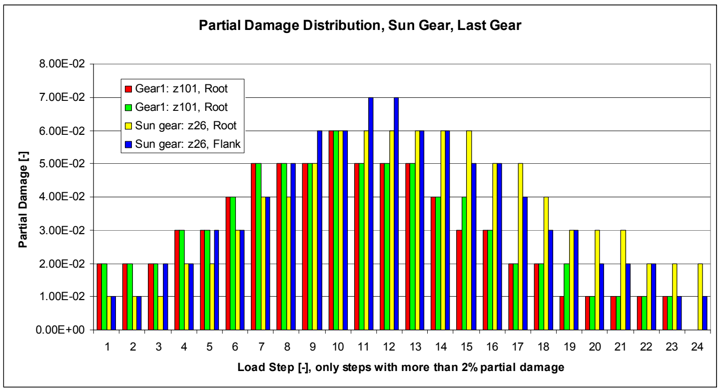

Fig. 12 Detail from the previous figure: only sun and last gear wheel are displayed

4. Summary

The use of a parameterized model allows a very quick verification of a transmission with a spectrum of about 250 load steps for safety factors, service life as well as damages under various assumptions. With it, a tool and a methodology are at the engineer's disposal allowing him/her to examine, quickly and with very little complexity, how a given design will react to changes in the calculation input parameters.

The investigations show that it is irrelevant whether one works for the spectra with variable or constant (calculated at nominal load) *K* factors. With the easier methodology, it is also possible to work maintaining the *K* factors constant. Interesting is, with what clarity one can find out in what spectrum area the damage occurs. This information is of the utmost interest for the person who has to establish the load assumptions or test them.

It has also been shown that the spectrum form has an influence upon the calculation that cannot be neglected. There is the need to calculate a transmission not only for a spectrum but also for various spectra. For instance, these can result from a changed tower height, blade length or from different installation places.

The question of at what cost can the assumptions identified as relevant be improved is the most difficult question to answer. The question here goes especially to the equipment manufacturers (in view of the availability of various load assumptions), the gearbox manufacturers (for instance, in view of material characteristics, gearing quality and other influencing values originated in the production), component suppliers (for instance, bearings) and also certifiers (for instance, who specify the choice of *K* factors in their guidelines).

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