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Gear Design for Wind Turbine Gearboxes to Avoid Tonal Noise According to ISO/IEC 61400-11

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Present wind turbine gearbox design usually includes one or two planetary gear stages and at least one high speed helical gear stage, which play an important role regarding noise and vibration behaviour. Next to the overall noise of the gearbox and the structure-born noise on the gearbox housing also tonal noise is becoming a much more important issue in recent years. Since tonal noise is problematic due to the human perception as “uncomfortable”, avoidance is important. Conventional theories regarding low noise gear design are not developed in view of tonal noise. This leads to the question: How to deal with tonal noise in the design stage and which gear parameters can be used for an optimisation regarding good tonal noise behaviour?

Within a research project measurements have been performed on different gearboxes using different gear designs. These measurements have been evaluated according to ISO/IEC 61400-11 and the results have been analysed in view of the influence of different gear parameters. It was also investigated if it is possible to rank gearboxes in wind turbines according to their tonal noise behaviour as observed on the test rig.

The paper will give an introduction into the definition of tonal noise according to ISO/IEC 61400-11 and give insight in measurement results from test rigs and from gearboxes in the field, where noise behaviour is also evaluated according to ISO/IEC 61400-11. Furthermore, the paper will show and discuss the link between measurement results and different gear parameters, which are affecting tonal noise behaviour. In addition simulation results will be presented, showing how tonal noise can be estimated within the design stage using state-of-the-art calculation software.

The paper will give recommendations regarding a gear design process that is considering tonal noise in the design stage and will compare an, regarding tonal noise, improved gear set with an older one.

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Gear Design for Wind Turbine Gearboxes to Avoid Tonal Noise According to ISO/IEC 61400-11

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Introduction

Typical wind turbine gearbox (WTG) designs include one or two planetary gear stages and at least one high speed helical gear stage (HSS). These high speed gear stages play an important role in overall gearbox noise and vibration behavior. Noise behavior is characterized by overall sound pressure level as well as tonal noise behavior. The latter aspect is gaining importance in the wind turbine industry. Hence requirements are being specified in international standards [2] [5] [6], which are still under development.

Figure 1 illustrates how the gear mesh can generally be seen as the main source of noise and vibrations. Shafts, bearings and housing are the transfer path to the environment, resulting in airborne and structure-borne noise.

Conventional theories regarding low noise gear design focus mainly on reduction of overall sound pressure. This approach, which has been used over the last years, leads to a gearbox design with lower overall sound pressure level (including

reduction of gear mesh excitation), but pure tones may become more problematic due to a reduction of background noise levels, e.g. by improved gear quality. Figure 2 shows a comparison of overall sound pressure level and tonal noise for a gearbox design where tonal noise is masked due to high sound pressure level (Figure 2a) and a gearbox design where tonal noise is more visible due to a low overall sound pressure level (Figure 2b). Although overall noise and gear mesh excitation levels have been reduced in the latter gearbox design (Figure 2b), pure tones are clearly visible in the frequency spectra.

The term “tonal noise” is used for a sound which includes a specific frequency component with such high amplitude that it is perceived as “uncomfortable” for human hearing. Current evaluation methods are based on standard 1/3rd octave band analysis or procedures according to [2] and [6] specifying that no pure tone is allowed and depending on the level of the pure tone a penalty is applied on the total sound power of the gearbox [1].

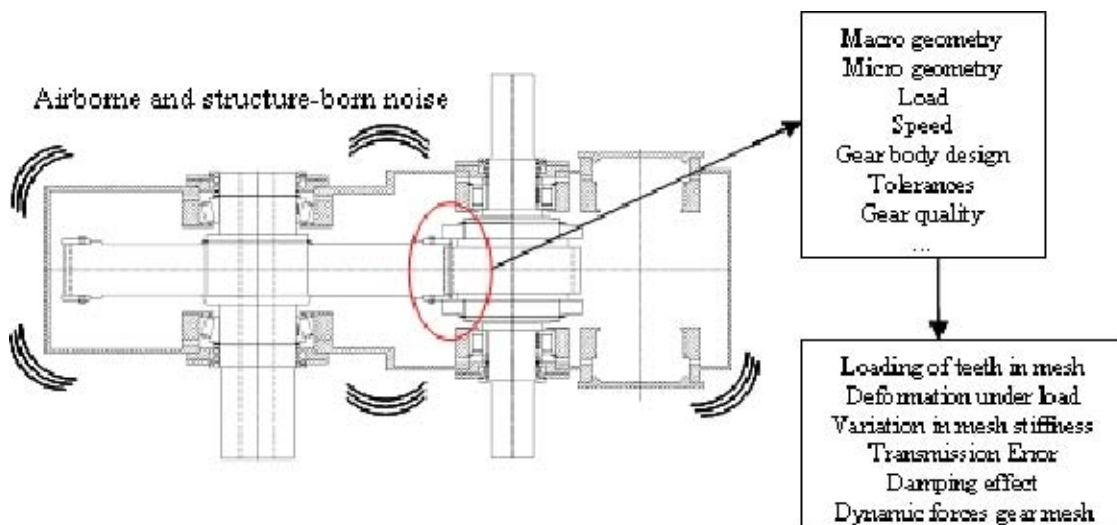


Figure 1. Model of noise origin coming from the gear mesh of a 13.2 MW HSS gear set with 770 mm center distance

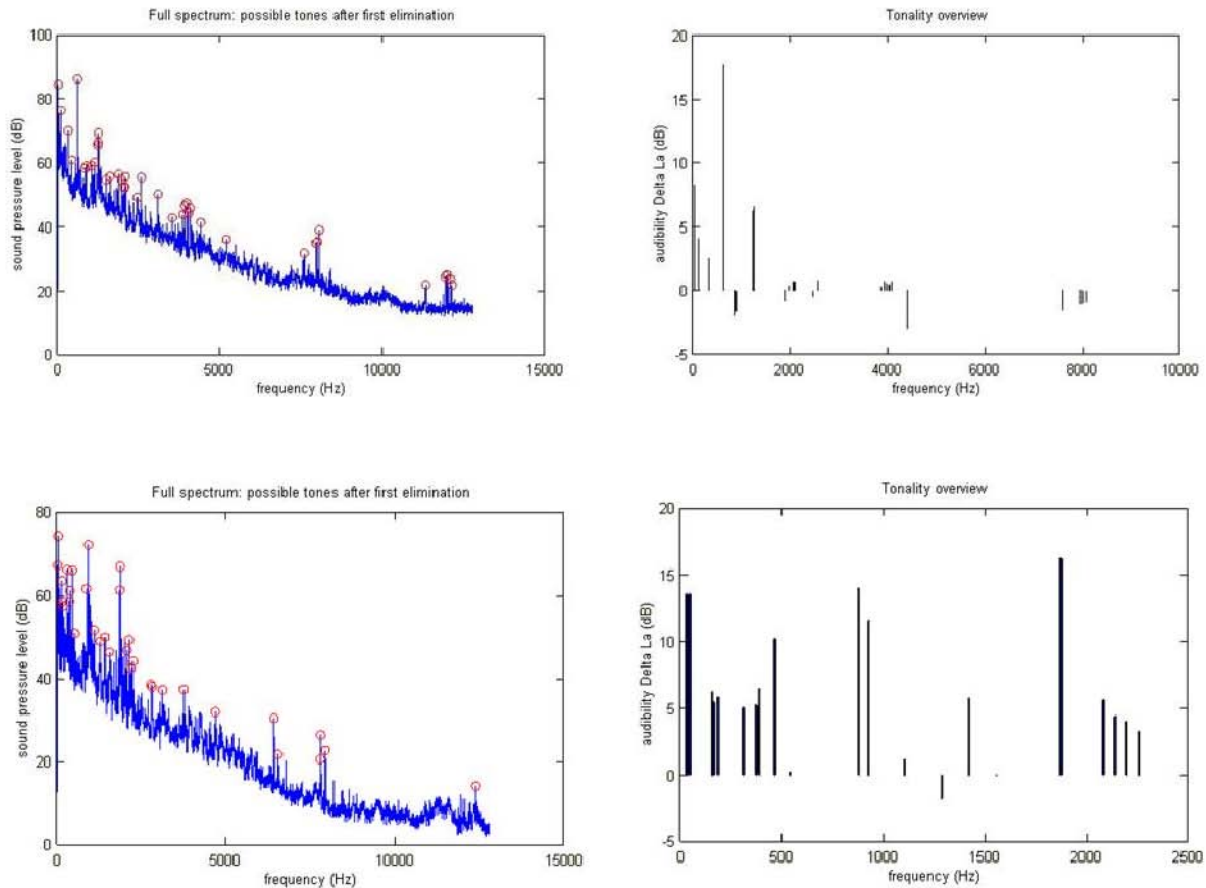


Figure 2. Tonal noise compared to sound pressure level a) gearbox with high sound pressure level where tonal noise is masked b) gearbox with low sound pressure level where tonal noise is less masked

Tonal noise problems should be prevented in gearboxes from the design stage. However, experience on how to avoid tonal noise in wind turbine drive trains is limited and only few guidelines are available. The available information is more related to automotive [9], helicopter [7] or train applications [8], or focuses merely on damping e.g. “How to eliminate tonal noise by active damping” [7].

Currently used parameters for low noise gear design include:

- low transmission error (TE) [9] [10] [18] [20]
- large total contact ratio, preferably an integer number [13]
- overlap ratio preferably an integer number [10] [12] [15]

Furthermore, international standards (such as [4]) specify minimum safety factors for tooth flank and tooth root loading, as well as the design load for tip relief – one of the main flank modifications, related to noise and vibration behavior. These specifications lead to a basic gear design optimized for nominal load. However, since noise plays typically a more important role in the partial load range of a wind turbine, the rules for basic gear design might not result in a design optimized for noise. However, standard parameters for low noise gear design are often in conflict with WTG design restrictions, such as bearing load capacity, overall dimensions and weight.

Above considerations lead to the question: How to optimize gear design with respect to tonal noise behavior? In order to define a best practice for low

tonal noise gear design a research project was set up, which is described in the next paragraphs.

Setup of the research project

As mentioned earlier, high speed (gear) stages (HSS) are an essential part of current WTG's. Therefore the research project as described below focuses on HSS with the aim to give an answer to the following questions:

1. Which gear parameters influence tonal noise?
 - a. What are the differences in the gear design of the investigated gearboxes?
 - b. Is there a direct link between measurement results and differences in gear design?
2. Is it possible to establish a design method, which balances load carrying capacity and noise behavior, preferably by using state-of-the-art software to analyze the gear mesh under various load conditions?
 - a. Definition of simulation model and available software.
 - b. Definition of output parameters and correlation with tonal noise measurements.
3. Which measurement methods can be used to evaluate gear sets on tonal noise behavior? When applied in test rig conditions, are the results also representative for field conditions?

To answer these questions, a measurement campaign has been set up on relevant gear sets, using standardized methods for tonal noise determination. Gear design parameters were analyzed in combination with a dynamic simulation of the gear mesh. Ultimate goal was to find a correlation between design and simulation parameters on the one hand and measurement results on the other, both for test rig and field conditions, see Table 1.

The gear stages used in the research project are designed according to the requirements of ISO/IEC 61400-4 for WTG with a nominal power between 1.5 MW and 2.5 MW.

Table 1. General gear set parameters

	Symbol	Unit	Unit range
Helix angle	β	°	11 - 17.5
Normal module	m_n	mm	7 - 8
Nominal speed	n	rpm	1450 - 1900
Quality according DIN 3962	Q	- -	< 6
Specific line load	F_t/b	N/mm	700 ($\pm 5\%$)

All gear teeth have flank modifications consisting of: lead correction, tip relief, lead crowning and profile crowning.

Determination of tonal noise according to ISO/IEC 61400-11

Tonal noise of a wind turbine is identified by means of field measurements at integer wind speeds in the range of 6 - 10 m/s, at 10 m height. Tones have to be determined on the basis of a narrow band analysis, as described in [2] and [3]. See Figure 3. The procedure includes the following steps:

Step 1: Determination of the critical bandwidth around the center frequency of the critical band f_c .

$$25 + 75 \left(1 + 1.4 \left[\frac{f_c}{1000} \right]^2 \right)^{0.69} \quad (1)$$

Step 2: Identification of possible tones and determination of the tone level, where the sound pressure level $L_{pt,j,k}$ is determined by energy summing all spectral lines as identified within the critical band. A possible correction for the Hanning window is necessary if two or more tones are involved leading to a division of the energy by the factor 1.5.

Step 3: Determination of the masking noise level using equation 2:

$$L_{pn,j,k} = L_{pn,avg,j,k} + 10 \lg \left[\frac{\text{critical bandwidth}}{\text{effective noise bandwidth}} \right] \quad (2)$$

$L_{pn,avg,j,k}$ represents the background corrected energy average of the spectral lines, identified as "masking" within the critical band.

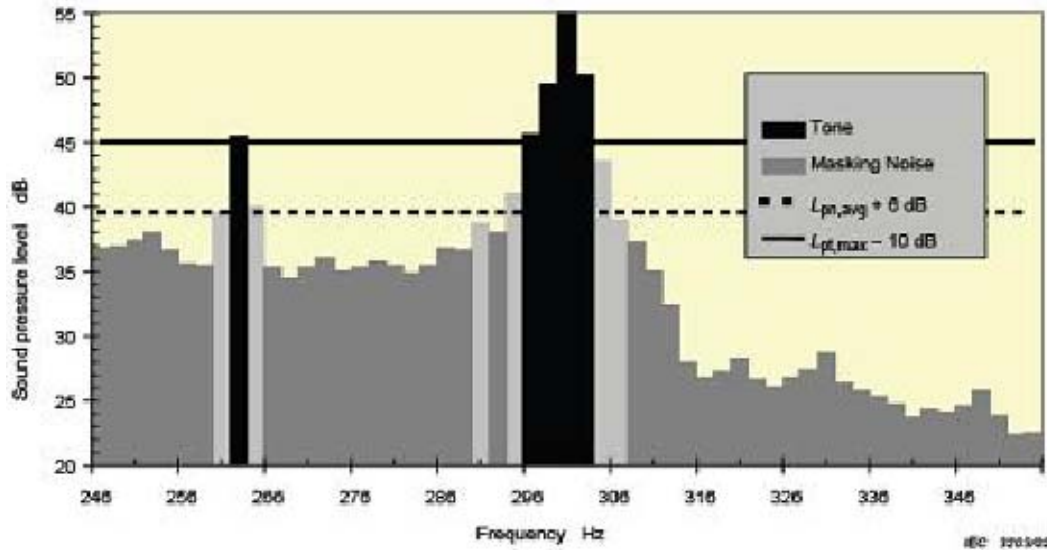


Figure 3. Classification of all spectral lines according to ISO/IEC61400-11 [2, page 34]

Step 4: Determination of tonal noise: The difference between the tone level and the masking noise in the corresponding critical band is defined with:

$$\Delta L_{tn,j,k} = L_{pt,j,k} - L_{pn,j,k} \quad (3)$$

In case no tone was identified $\Delta L_{tn,j,k}$ is defined by:

$$L_{tn,j,k} = -10 \lg \left[\frac{\text{critical bandwidth}}{\text{effective noise bandwidth}} \right] \quad (4)$$

Step 5: Averaging of $\Delta L_{tn,j,k}$ to one overall tonal noise ΔL_k for $k = 6, 10$.

Step 6: Compensation of the human ear response to tones of different frequencies for every value of ΔL_k using the frequency depending audibility criterion:

$$L_a = -2 - \lg \left[1 + \left(\frac{f}{502} \right)^{2.5} \right] \quad (5)$$

Step 7: Calculation of the tonal audibility using:

$$\Delta L_{a,k} = \Delta L_k - L_a \quad (6)$$

The criterion is defined with $\Delta L_{a,k} \geq -3.0$ dB. $\Delta L_{a,k} < -3.0$ dB is categorized as “No tonal noise present” at this specific tone.

Experimental methods

Measurement techniques

Experimental tonal noise values were derived from in-house measurements on a back-to-back test rig

(Figure 4). Sound is measured within a defined sweep window as shown in Figure 5 on the side of the HSS using one microphone type 4197 from Bruel and Kjaer.



Figure 4. Gearbox on back-to-back test rig

The necessary FFT analysis is done with a resolution of 2 Hz as defined in [2] and a measurement time of 20 sec is used, equal to a standard octave band measurement.

The evaluation of the measurement results is done according to the method as described earlier.

Since [2] is a standard developed for wind turbines and field measurements, small adaptations to the given measurement procedure were necessary to allow in-house evaluation.