

Influence of Topography Deviations on the Psychoacoustic Evaluation of Ground Bevel Gears

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In the design process of transmissions, one major criterion is the resulting noise emission of the powertrain due to gear excitation. Within the past years, much investigation has shown that the noise emission can be attributed to quasi-static transmission error. Therefore, the transmission error can be used for a tooth contact analysis in the design process, as well as a characteristic value for quality assurance by experimental inspections.

The noise behavior of bevel gears can be subdivided in accordance to the two main manufacturing chains — face hobbing and lapping—in comparison to face milling and grinding. Apart from the different lead function of both bevel gear types, their acoustic behavior also differs. While the frequency spectrum of the ground gear set shows, in general, low background noise levels with pronounced gear mesh amplitude and their higher harmonics, the noise spectrum of a lapped gear set looks the opposite. The background noise level is higher and the tooth mesh amplitude is less dominant, as well as the higher harmonics.

The difference can be explained by the resulting microgeometry deviation after the hard-fine machining. The grinding process has high reproducibility and high pitch accuracy.

Introduction

A major criterion in the quality assessment of bevel gears is the noise behavior in the application. In the gear design, the noise behavior is addressed via the Ease-Off design that represents a compromise between a low transmission error level and a sufficient load-carrying capacity. Numerous past investigations proved that quasi-static transmission error is the appropriate evaluation parameter when it comes to noise behavior. The transmission error is the excitation source of the dynamic system (Fig. 1). Therefore, in the design process of noise-critical applications, optimization requires a low transmission error fluctuation or low amplitudes for the transmission error frequency and higher harmonics. The final Ease-Off geometry for an optimal performance over the load range poses a challenge in the bevel gear design. Dynamic influences and hearing-related noise evaluations are not considered, to date. Furthermore, micro deviations, e.g. — for a lapped gear set, or long-wave deviations as pitch or run-out deviations — lead to significant differences compared to nominal transmission error (Refs. 7, 9 and 15).

Noise behavior of bevel gears. Unique to bevel gears when compared with spur or helical gears is that the noise behavior is already influenced by the soft-machining process and the fol-

lowing process. The process chain for the discontinuous face milling processes results in a circular arc as lead function for the teeth. In this way heat treatment distortions can be eliminated in the grinding process. The ground gear set shows in general a high reproducibility of the nominal topography, with low pitch errors. Bevel gears manufactured in the continuous hobbing process provide an epicycloidal lead function and are lapped for the reduction of heat treatment distortions. The material removal rate in the lapping process is lower compared to grinding and, therefore, lapped gear sets are strongly influenced by the quality of the soft-cut gear geometry. In general, the topography deviation and the pitch deviations are higher for lapped gears (Refs. 15 and 19).

Furthermore, the closed-loop manufacturing allows narrow deviation fields. In contrast, the lapping process is subject to higher geometry fluctuations due to a lower material removal rate and, therefore, the strong dependency of the geometry after heat treatment.

The difference in the transmission error frequency distribution has a huge impact on dynamic noise behavior. Apart from the evaluation of the differential acceleration as a counterpart to the quasi-static transmission error, the consideration of weightings regarding human noise perception is common in other acoustic disciplines. Therefore, it is the aim to evaluate in the impact of an individual topography deviation on the dynamic noise behavior. In the first step, the tooth contact analysis *ZaKo3D* is evaluated comparing the measured and simulated loaded transmission error. Afterwards, a microgeometry scatter is applied on a ground gear set and the impact on the transmission error is shown. As observed for lapped bevel gear set, the background noise increases and the gear mesh amplitudes decrease. Finally, in a simulation study two variants are compared regarding the difference in psychoacoustic parameters as loudness, sharpness, roughness and tonality.

In Figure 2 the structure-borne spectra for a lapped and ground gear set is shown (Ref. 18). The process-related differences in the topography and the pitch error characteristic after the hard finishing lead to a tendency of higher tonality for ground gears, with pronounced noise shares by the tooth mesh frequency and their higher harmonics, and a low background noise level. The noise spectrum of a lapped gear set, in contrast, shows a lower tonality and higher background noise level. The differences in the noise spectra of the lapped and ground gear

lowing process. The process chain for the discontinuous face milling processes results in a circular arc as lead function for the teeth. In this way heat treatment distortions can be eliminated in the grinding process. The ground gear set shows in general a high reproducibility of the nominal topography, with low pitch errors. Bevel gears manufactured in the continuous hobbing process provide an epicycloidal lead function and are lapped for the reduction of heat treatment distortions. The material removal rate in the lapping process is lower compared to grinding and, therefore, lapped gear sets are strongly influenced by the quality of the soft-cut gear geometry. In general, the topography deviation and the pitch deviations are higher for lapped gears (Refs. 15 and 19).

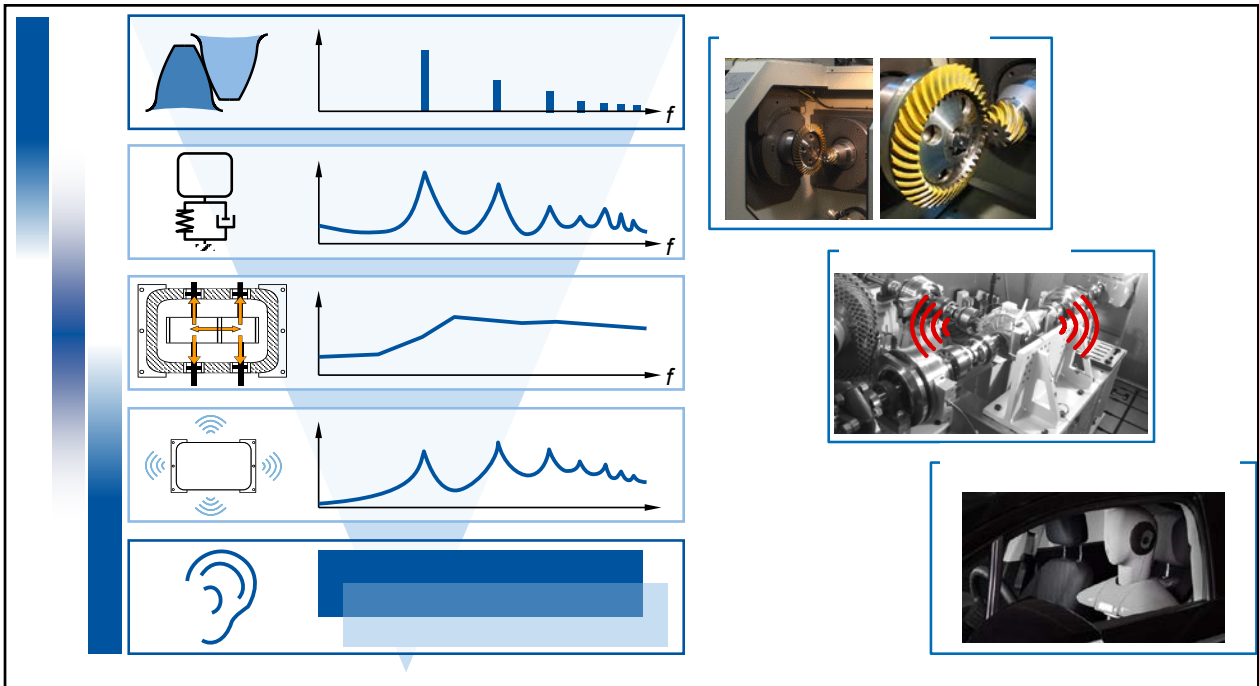


Figure 1 Source-path-receiver concept of bevel gear transmissions.

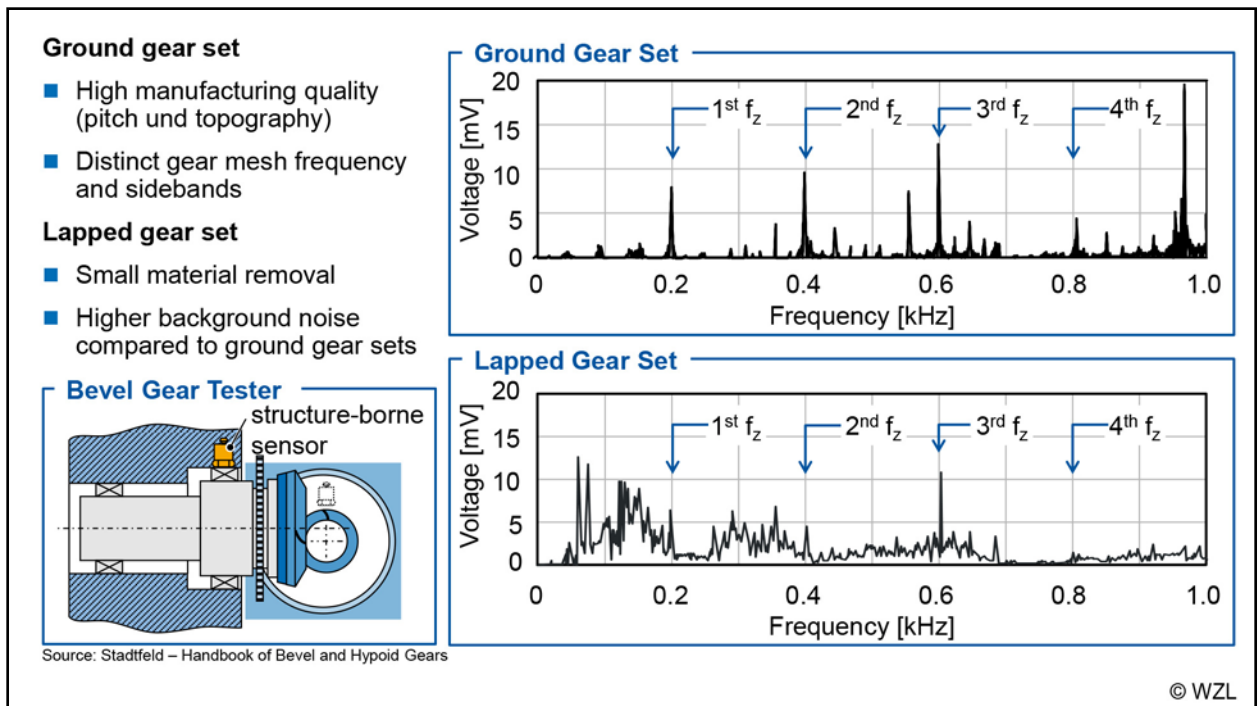


Figure 2 Structure-borne noise spectra of ground and lapped bevel gears.

set leads to the fact that lapped gear set are perceived to be quieter or more pleasant (Refs. 3, 16, and 18).

Aim. The aim of the report is the investigation of the influence of the topography deviations of ground bevel gears on noise behavior; therefore, three gear set variants are analyzed here. Quasi-static transmission error analysis and dynamic simulation in accordance to Gacka are performed (Refs. 4 and 12). The simulated, differential rotary acceleration is evaluated by means of psychoacoustic parameters.

The gear set macro geometry originates from an automobile application. The reference gear set is ground with high pitch

quality and low topography deviations. The other two ground variants are developed in similarity to lapped gear set characteristics. They provide a low, individual deviation from tooth to tooth and high pitch quality as well. By reproduction of the topography deviation from tooth to tooth it is targeted to imitate the low tonality of the lapped gear set.

The gear set variants are simulated with the FE-based tooth contact analysis (TCA) *ZaKo3D*. Therefore, it is the first subordinate target in the validation of the FE-based TCA under load for bevel gears. Loaded transmission error measurements are performed with a new bevel gear fixture and compared to sim-

ulations in *ZaKo3D*. In the following a procedure is shown to imitate the topography deviations of a lapped gear set by means of overlaying a small microgeometry deviation from tooth to tooth. In connection with the manufacturing of the gear set variants, single flank tests are performed. The load-free results are validated via simulations considering the topographies of all teeth. Finally, the dynamic behavior of the gear set is simulated and the results are evaluated considering human hearing with psychoacoustic metrics.

Measurement and Simulation of the Loaded Transmission Error

The transmission error is a central design parameter for the optimization of the acoustic excitation behavior. For the development of software programs for gear design, a validation of test rig results in comparison to simulations is essential. In (Ref. 5) a bevel gear fixture for the investigation of the acoustic behavior is presented. In the following, results of loaded transmission error (LTE) measurements and loaded contact pattern are shown. Finally, the test rig results are simulated with *ZaKo3D*.

Commissioning of bevel gear fixture. In past years a significant number of gear set fixtures were developed. Plewnia developed a bevel gear fixture for the investigation of the excitation behavior of bevel and hypoid gears (Ref. 17). The test fixture consists of two bearing blocks for the pinion and ring gear shaft that can be positioned via rail guidance on the base plate. In order to acquire the gear mesh excitation close to the tooth contact, the measurement systems for the measurement of the transmission error and the rotary acceleration are integrated in these bearing blocks. The hypoid offset is realized by ground adapter plates below the bearing block of the pinion. Reproducibility of the positioning of the gear set via bearing blocks was a disadvantage for the narrow tolerances of automotive applications. Hohle used the same test concept for a test fixture for spur and helical gears. Due to the smaller sensitivity of

the involute for spur and helical gears, the disadvantage regarding the positioning is not given for this application.

Carl analyzes the noise behavior of spur and helical gears. The test set-up is developed as a frame base structure with bolted front plates with integrated bearing seats. The frame structure provides good accessibility for contact patterns and a close connection of the measurement systems. In order to measure the dynamic excitation, rotary accelerometers with a telemetric system were developed that provide a compact design (Ref. 7).

Figure 3 shows the constructional design of the bevel gear fixture for the investigation of the noise characteristic of bevel and hypoid gears derived from the results of (Refs. 6, 7 and 17). As well, a frame structure instead of separated bearing blocks is realized. In this way the gear set can be positioned by housing elements. The pinion and ring gear installation position are adjusted by paired ground shim plates (Fig. 3-1, 2). The hypoid offset and the crossing angle are given by the pinion front plate (Fig. 3-3). In order to avoid influences from the differential gear set, the differential and differential basket are substituted by the ring gear shaft. The measurement of the transmission error is realized by angle encoders. The dynamic rotary excitation is measured by the measurement systems from (Refs. 5 and 7).

The commissioning of the gear fixture is performed with a hypoid axle with a ratio of $z_1/z_2 = 11/46$ and a hypoid offset of 35 mm. The test fixture is tested on the universal gear test rig II at WZL and flanged on the machines via CV shafts. One criterion of the commissioning tests is the reproducibility of the transmission error measurements. A measurement cycle consists of constant torque levels at 60 rpm from 25 to 1,200 Nm in drive mode, and from 25 to 600 Nm in coast mode. The measurement cycles are repeated three times, measuring always a complete tooth hunt. The ground gear set shows relevant amplitudes of excitation for the first four gear mesh orders. In Figure 4 the minimum, average and maximum graphs over load for the amplitudes of the first four gear mesh orders are depicted.

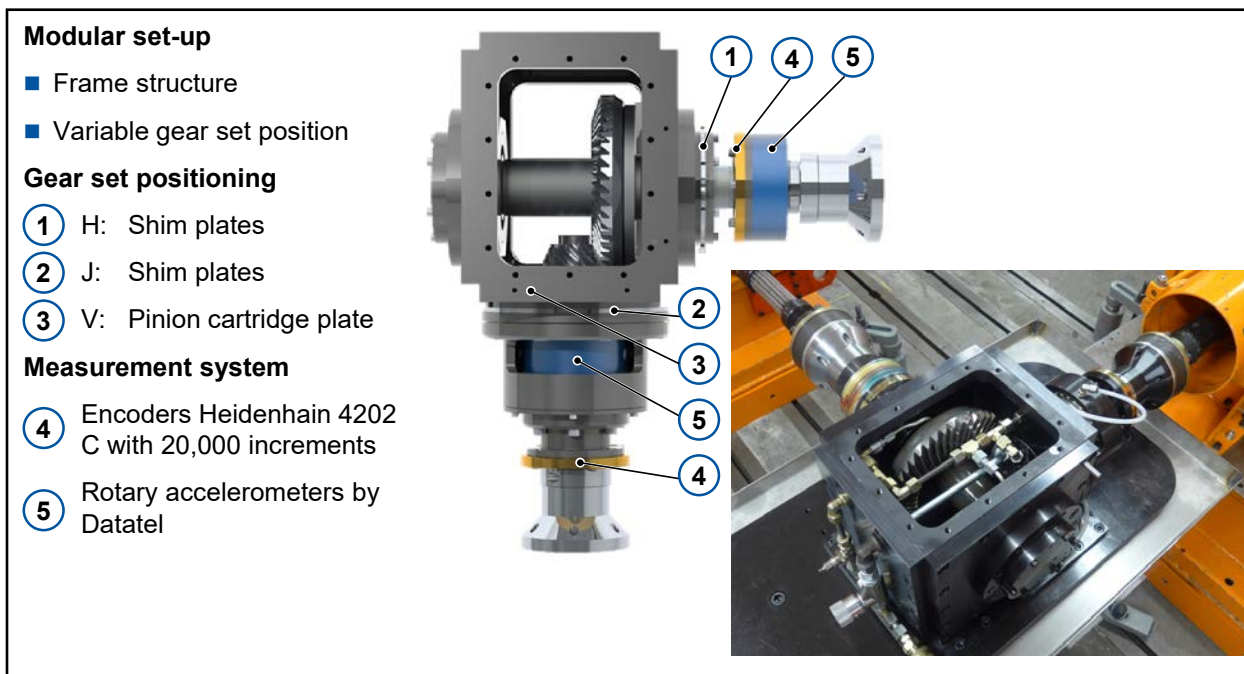


Figure 3 Bevel gear fixture for investigation of excitation behavior of hypoid gears.

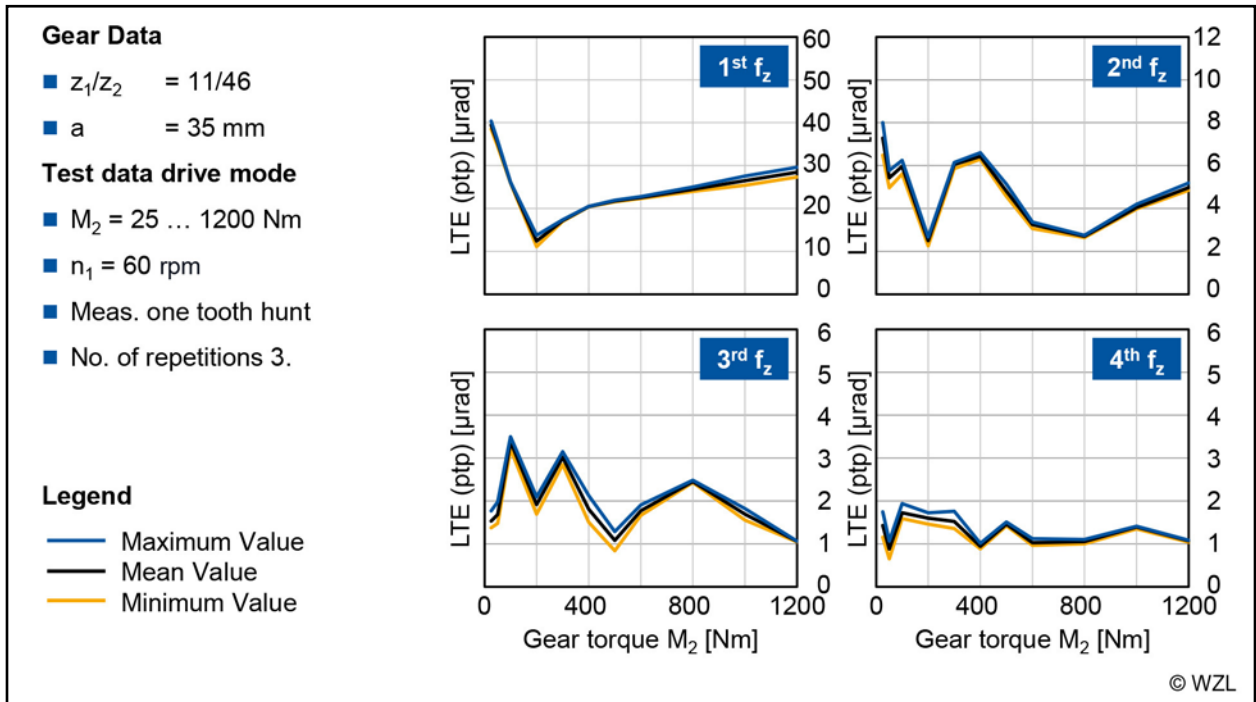


Figure 4 Repeatability of measured loaded transmission error (LTE) graphs.

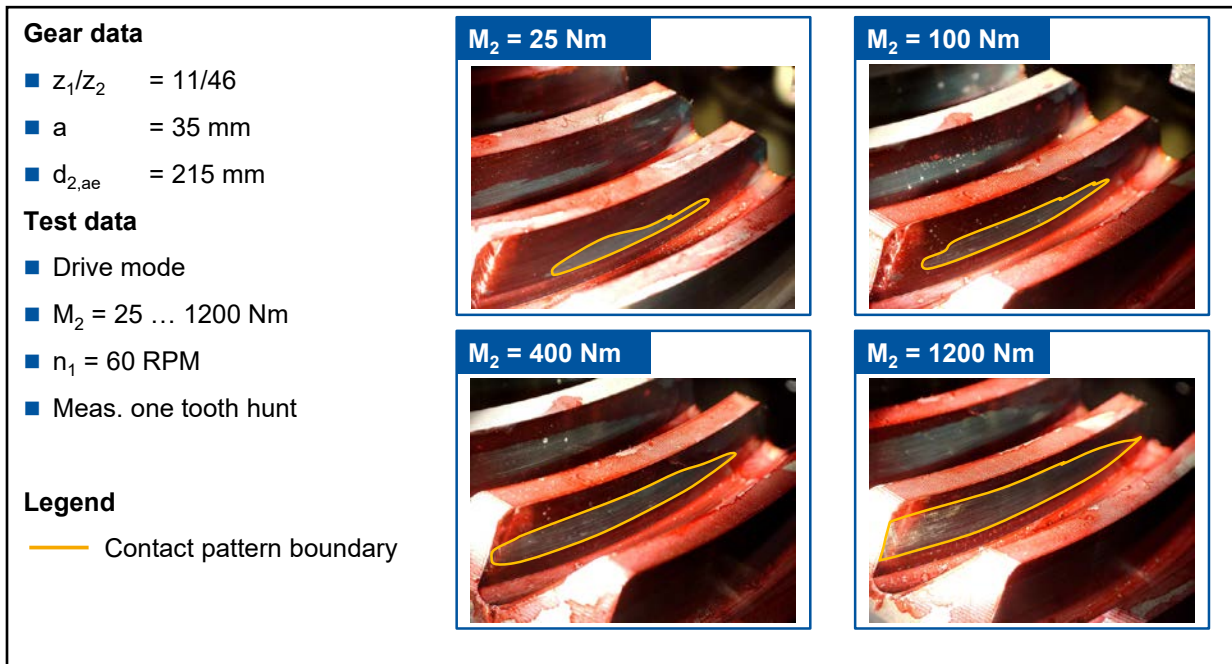


Figure 5 Measured contact pattern for various load levels.

Qualitatively, the graphs of the orders show a high reproducibility. The local extrema are repeatedly measured for the considered gear mesh orders. The maximum deviations of the first and second gear mesh amplitude are smaller than $2.7 \mu\text{rad}$; the maximum differences for the third and fourth gear mesh amplitude are below $0.7 \mu\text{rad}$. The measured differences in the test rig investigations are attributed to the temperature behavior of the test fixture, which influences the bearing stiffness and therewith the displacement behavior of the gear set.

Comparison of measured and simulated loaded transmission error. The measured transmission error graphs are simulated in the FE-based tooth contact analysis *ZaKo3D*. For characteriza-

tion of the installation position of the gear set, the contact pattern for different load levels is considered. Furthermore, the contact pattern enlargement—as well as the contact pattern displacement—is considered for the load-caused displacement. The contact pattern from the test rig is depicted in Figure 5. The quasi-load-free contact pattern at 25 Nm shows a centered contact area in lead direction with a tendency to the tooth root area. With increasing load the contact area increases and reaches the heel of the flank at a gear torque of 400 Nm. At maximum torque of 1,200 Nm the contact pattern is enlarged over the edge at the heel. The position is especially unfavorable for high torques; nevertheless, the commissioning tests enable conclu-

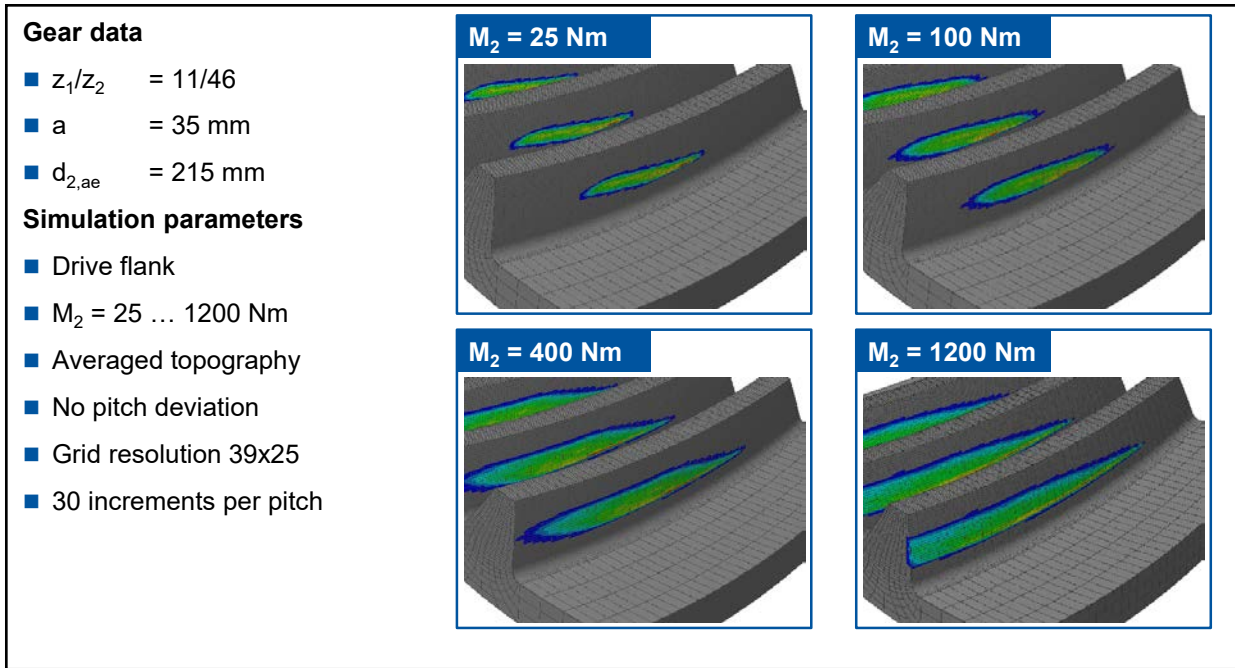


Figure 6 Simulated contact pattern for various load levels.

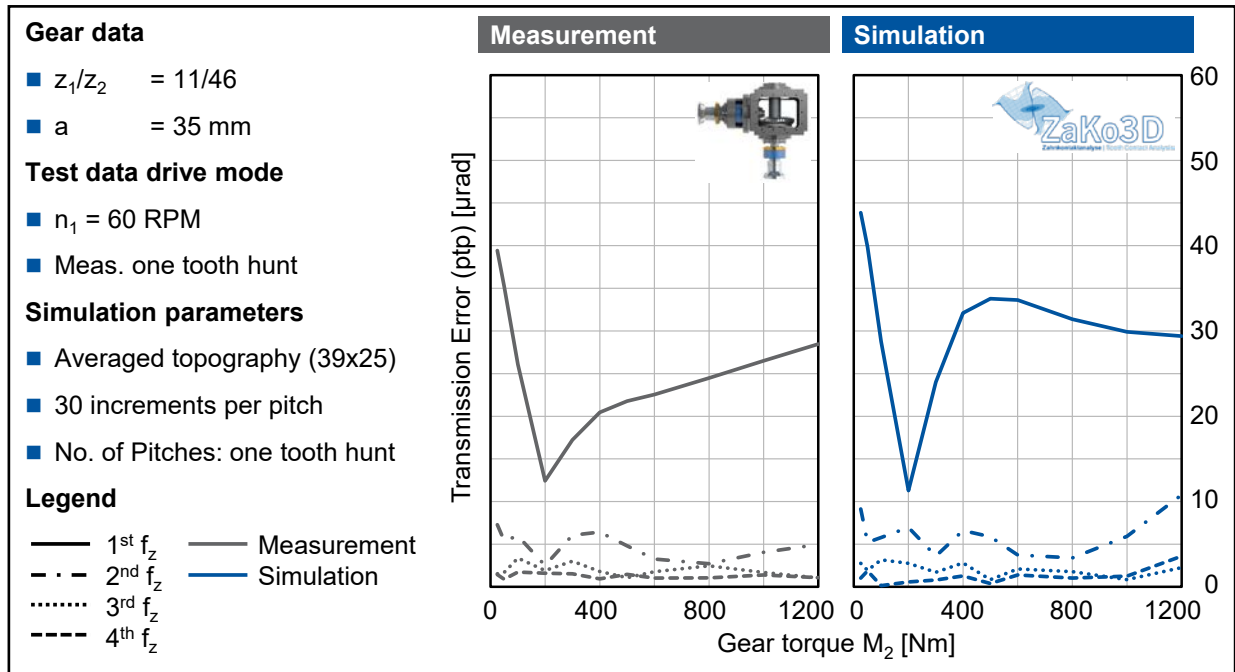


Figure 7 Comparison of measured and simulated transmission error in drive mode.

sions about the simulation capability of *ZaKo3D*.

Following is the contact pattern from the tests imitated in the simulations in *ZaKo3D*. Input data of the tooth contact analysis are the averaged topographies from four teeth of pinion and gear. In Figure 6 the simulated contact pattern is depicted. In general, the simulated contact pattern position in lead and profile direction on the flank show good agreement with the tests. The color scale of the pressure distribution allows identification of the center of the contact pattern — located in the root area. At 400 Nm the contact pattern also reaches the edge of the flank at the heel. At 1,200 Nm it is prolonged clearly over the edge. In toe direction the contact pattern is tapered to the root but, in comparison to the test rig, the distance is too great.

Furthermore, the contact pattern in profile direction at higher loads shows a smaller width in the simulation compared to the tests, meaning that the distance of the contact pattern edge to the tip of the flank at maximum torque is bigger. The difference between measurement and simulation can be explained by the fact that only the averaged topography of the simulation is considered. Furthermore, as the documented contact pattern from the measurement is an accumulation of various pitches, the simulated contact pattern in this case should always be of a smaller extent. But the deviation in profile direction could also be caused by a lower crowning compared to the flank used for the simulation.

After comparison of the gear set position the graphs of the

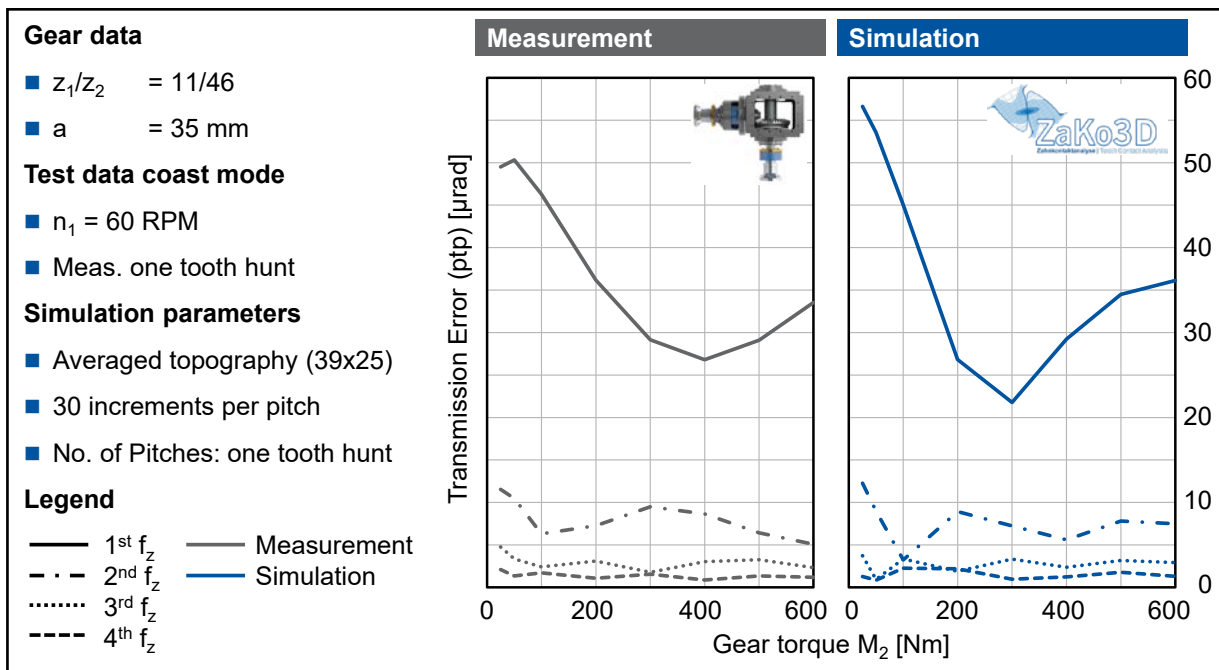


Figure 8 Comparison of measured and simulated transmission error in coast mode.

first four gear mesh orders for drive mode (Fig. 7) and coast mode (Fig. 8) are presented. For the drive flank the amplitude drops from 40–45 μrad at 25 Nm to a minimum of approx. 13 μrad . This characteristic is shown in the simulation and in the measurement. For higher torques the amplitudes rise higher in the simulation. While the simulation shows a local maximum at 500 Nm of 34 μrad with a following digressive drop to 30 μrad at 1,200 Nm, the amplitude in the measurement increases non-stop up to 28.5 μrad . The difference in the transmission error amplitudes is in agreement with the difference in the simulated contact pattern size. The higher contact pattern size of the measurement leads to a higher contact ratio and, therefore, lower transmission error.

The change of the simulated transmission error amplitude for the second to the fourth gear mesh order shows qualitatively and quantitatively a good agreement to the measurement. Especially for the measured and simulated second gear mesh order local minima at the torque levels of 50, 200 and 800 Nm are shown.

The comparison of the first four gear mesh amplitudes for torque levels up to 600 Nm in coast mode is depicted (Fig. 8). The change of amplitude over torque shows a minimum at a medium torque in the simulation and measurement. In the measurement the transmission error drops digressively from 50 to 27 μrad at 400 Nm, and rises afterwards up to approx. 33 μrad . In the tooth contact analysis, the transmission error starts at 56 μrad and falls down to 22 μrad at 300 Nm, and rises to 36 μrad at 600 Nm. The gradation matches qualitatively for the higher harmonics.

The differences in the validation results for the gear set are explained with effects of the topography of all teeth and pitch deviations. In spite of a grinding finishing process, slight deviations in the topography and pitch quality are given that might impact transmission error. Also, the gear set position was not determined via CMM measurements and, therefore, no correc-

tion to the nominal position with a better contact pattern was done. In future investigations, the installation position needs to be measured and the load-caused deflections for various bearing preloads for the torque range must be determined.

Summarizing the validation of the FE-based tooth contact analysis *ZaKo3D*, it can be stated that the simulation provides high agreement with the measurement with the new bevel gear fixture; the load-caused contact ratio increase is displayed accordingly. The gradation and change of the amplitudes of the higher harmonics of the gear mesh order over torque match qualitatively.

Reduction of Tooth Mesh Amplitudes by Individual Topography

The results of the measured and the simulated transmission error show an excitation of the gear set up to the fourth gear mesh order. This is in accordance to the formulated characteristic of ground bevel gear sets. Landvogt investigates the impact of the hard finishing process of bevel gears on the transmission error. Landvogt concludes that the higher individual deviations of the topography of a lapped gear set may lead to a higher background noise level in the transmission error frequency spectrum. Derived from that fact he stated that a small topography changes from tooth to tooth may lead as well to a transmission error reduction (Ref. 16).

The load-free results of the tooth contact analysis *ZaKo3D* were already validated in (Refs. 2–3). Herein it can be seen that the amplitudes in the frequency spectrum for a lapped and structural lapped gear set are clearly higher compared to the simulation. Only if the topography of all teeth and the pitch error are considered do the amplitudes decrease and match the measurement.

Procedure for the reduction of tooth mesh amplitudes. Figure 9 shows the derived procedure for the reduction of the tooth mesh amplitudes of the transmission error spectrum via

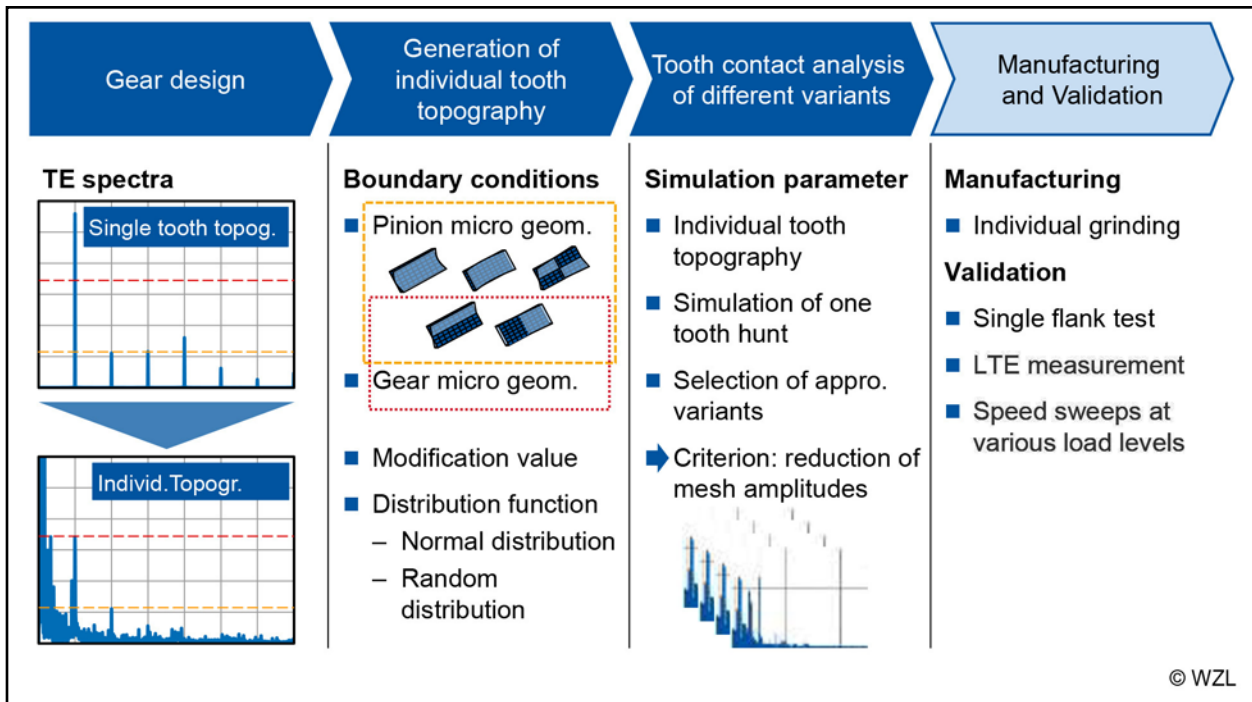


Figure 9 Procedure for reduction of tonality of ground bevel gears via individual tooth topography.

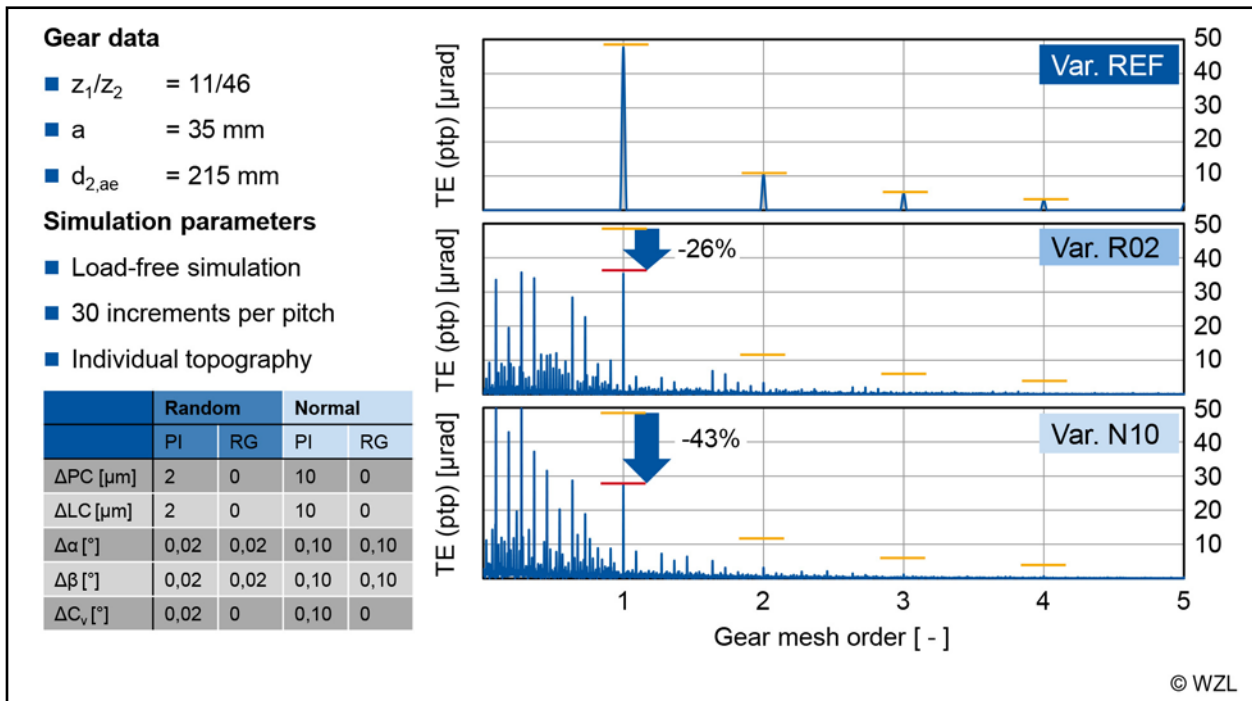


Figure 10 Simulation of transmission error spectra with individual tooth topography.

increase of background noise level. The starting point is the ideal Ease-Off topography of the ground gear set. The simulation of the ideal micro geometry possesses excitation for the gear mesh order and its higher harmonics. Via application of an individual topography deviation from tooth to tooth, the noise spectrum of the ground gear set is approximated to the noise spectrum of a lapped gear set.

The boundary conditions for the design of the individual topography are determined by the grinding process. An additional dressing of the grinding tool during the grinding process needs to be excluded regarding feasibility of the manufacturing

process; i.e. all microgeometry changes are realized by the kinematics of the machine tool. For the design of the pinion topography scatter, all modifications are allowed while for the ring gear topography, only spiral and profile angle deviations are permitted. For the investigated gear set the pinion has 11 teeth and the ring gear 46 teeth. Therefore, an additional restriction for the ring gear is defined that means that only 10 different topographies are distributed over the circumference. The topography scatter is varied in terms of the distribution function of the deviations and the deviation absolute value. The absolute values for the crowning deviations are varied in steps of 1 µm and the

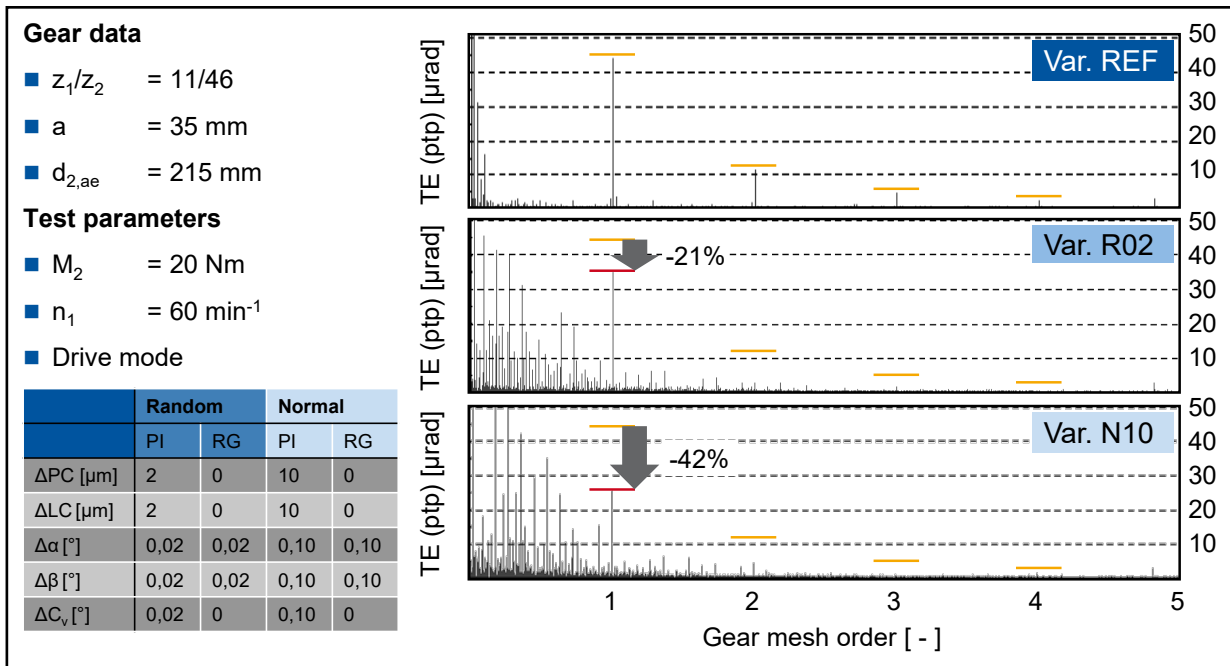


Figure 11 Measurement of transmission error spectra with individual tooth topography.

angle deviations in steps of 0.01° . The considered distributions of the deviations are the normal distribution and the random distribution. Systematic distributions are excluded due to their impact on the sideband characteristic, as it is expected for sinusoidal pitch deviations.

Afterwards, tooth contact analysis for a complete tooth hunt, which means 506 pitches, are performed. In order to limit the calculating time, only load-free simulations are done. The evaluating criteria for selection of the variants manufactured afterwards are reduction of the gear mesh orders of the load-free transmission error.

Finally, the selected variants are manufactured and the procedure is validated via test rig investigations. In the first step of the validation, single flank tests of the gear set are evaluated. In addition, loaded transmission error measurements, as well as speed ramps under loaded conditions, are carried out. The results in the following chapter are limited to the validation of the load-free transmission error spectra.

Simulation and measurement of load-free transmission error. Three variants are chosen for manufacture—the reference variant REF and two variants (R02 and N10) with a topography scatter. The absolute values of the crowning and angle deviations are listed (Fig. 10, bottom left). The variant R02 is characterized by a random distribution of an absolute value of $2 \mu\text{m}$, respectively 0.02° . For variant N10 a normal distribution of the topography scatter with absolute values of $10 \mu\text{m}$ for the crowning and 0.10° for angle deviations is realized. The averaged topography of the variants R02 and N10 is almost equal to the REF variant. The transmission error spectra for the three variants are depicted (Fig. 10, right). Additionally, the amplitude values for the first four gear mesh orders are marked in all three diagrams. Via application of the topography scatter, the amplitude of the first gear mesh order is reduced by 26% for the R02 variant and by 43% for variant N10. But in parallel, the amplitudes of the frequency range up to the first gear mesh order rose. Many of these amplitude peaks are in the range of the first

tooth mesh amplitude, and for the variant N10 the amplitudes are even exceeding the first gear mesh amplitude. The amplitudes for the higher harmonics instead are almost eliminated. The characteristic of the transmission error spectra of R02 and N10 are similar to the one shown for the lapped gear set (Fig. 2).

The simulated variants are tested in the following on a bevel gear tester Klingelnberg T50.

For manufacturing, individual topographies are given and translated into corrected machine kinematics. The focus is given to the drive flank of pinion and ring gear. The pitch quality of the pinion is 3 for the single pitch deviation and quality 2 for the cumulated pitch deviation. The achieved quality for the ring gear is quality 3 for both pitch deviations.

Figure 11 shows the transmission error spectra of the measurement of 506 pitches for the various gear set variants. In contrast to the simulated transmission error spectrum of the variant REF, the results from the bevel gear tester provide as well the rotation order and higher harmonics. The higher frequency range is in agreement with the simulation and is only dominated by the gear mesh order and their harmonics.

The measurement of the variants with the topography scatter shows very good correlation for the reduction of the first gear mesh amplitude. The amplitude of the R02 variant is reduced by 21% and for the N10 variant by 42%. Furthermore, the higher harmonics of the gear mesh order are almost eliminated. The characteristic of the increased amplitudes in the frequency range up to the first gear mesh order is also in accordance with the simulation.

Analysis of Dynamic Excitation Behavior and Psychoacoustic Evaluation

The results of the single flank tests are proving that by application of an individual topography scatter from tooth to tooth, it is possible to reduce the amplitudes of the gear mesh order and their higher harmonics. For the chosen variants, the focus is the first gear mesh order of the load-free transmission error.

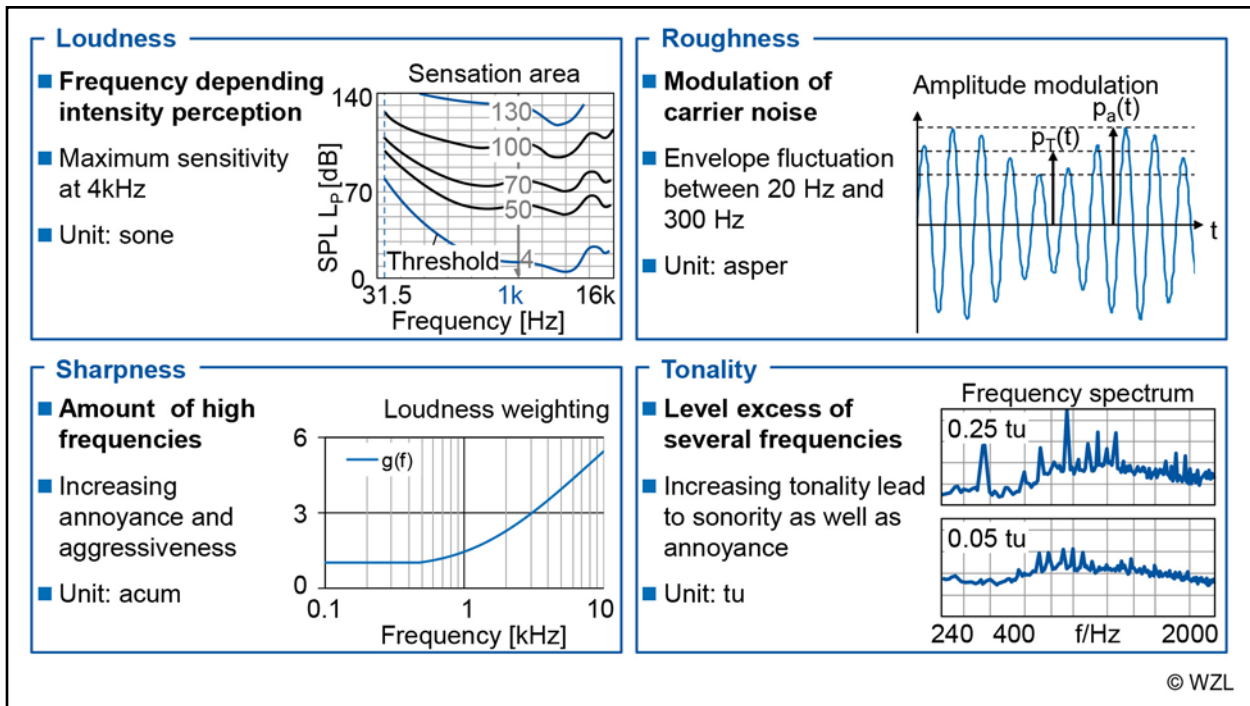


Figure 12 Psychoacoustic parameters.

However, the significant reduction of 20% and 40%, respectively, of the first gear mesh order is offset by an increase in the lower frequency range. Thus the question that must be asked is how the increase in background noise in the lower frequency range impacts the noise impression. Therefore, a dynamic simulation as described (Ref. 4) is performed with a subsequent psychoacoustic evaluation.

Psychoacoustic parameters. The hearing range of the human auditory system is in the frequency range from 16 Hz to 16 kHz. Within this frequency range the volume perception is not constant. Therefore, the objective parameters (A physical quantity, property, or condition that is measured) sound pressure and frequency are in contrast to the subjective human noise evaluation. The construction of the human ear influences significantly the noise perception.

The term “psychoacoustics” summarizes parameters that are considering the human auditory system. A spread approach to consider human hearing in noise measurement is the A-evaluation that is used in the airborne noise measurement. Thereby, curves with the same loudness impression (isophones) of the auditory sensation area diagram are imitated. That means that low frequencies cannot be perceived as well as middle-to-high frequencies with the same sound pressure level (Ref. 14).

Initial test rig investigations at the WZL using psychoacoustic noise evaluation were conducted (Ref. 1). Subsequently, Carl conducted comprehensive investigations of the noise behavior of cylindrical gears with various modifications, finishing processes and pitch deviations in order to prove its potential for the evaluation of gear noises (Ref. 7); some main psychoacoustic parameters are summarized (Fig. 12). The loudness evaluates to what extent human hearing perceives sound (Ref. 8). For the statement of the annoyance level of a noise signal, other parameters are defined. The sharpness considers the amount of high frequencies in the total noise signal and describes with increasing sharpness an

annoyance or aggressiveness of a noise signal; in Reference 10 a weighting function for higher frequencies is defined.

The roughness describes the influence of patterns by modulation of the noise signal. Noises with a signal envelope fluctuation of 20 to 300 Hz are considered rough. If a tone signal is modulated above 300 Hz, it can be determined as main and side bands of the carrier signal (Ref. 11). Noises are described as tonal if single frequencies or narrow bands of the frequency spectrum are dominant, in comparison to the background level (Ref. 9). Noise signals with a high background noise level have a low tonality in comparison to tone-like noises.

Psychoacoustic evaluation of ground bevel gear variants. The force coupling element that is developed at WZL is used in order to connect the torsional vibration model of the test rig (Fig. 3) and, thereby, allows investigation of the individual topography scatter to the dynamic excitation behavior. The parameterization of the test rig (Ref. 4) is not validated yet and, therefore, the psychoacoustic parameters are normalized. Nevertheless, the simulation can be used for the investigation of the effect of the topography scatter to the different variants. Stiffness matrices are calculated in the FE-based tooth contact analysis *ZaKo3D* for the variants REF and R02 considering the load-caused deflection of the test rig and the individual topography.

In the calculation model—as for the later test rig trial—at the driving end of the drivetrain the speed is controlled and a constant torque is applied at the output shaft. In the following, speed ramps from 0 to 5,000 rpm, with an increase of 100 rpm/s at an output torque of 425 Nm, are discussed. The order spectra of the differential rotary acceleration up to the 110th rotation order of the pinion are shown (Fig. 13).

The order spectrum of the reference variant shows, as expected, the tonal characteristic of the transmission error spectrum; excitation is noticeable up to the 10th gear mesh order. Furthermore, two dominant Eigen frequencies can be identified with a hyper-

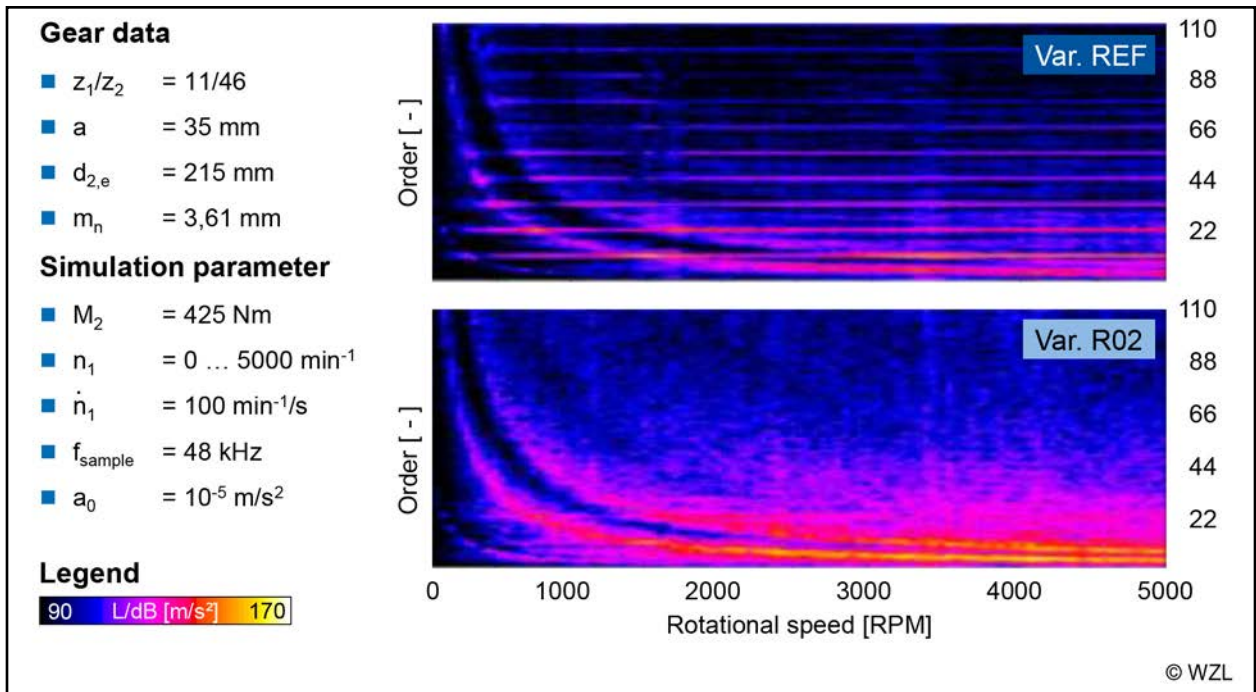


Figure 13 Order spectra of differential acceleration.

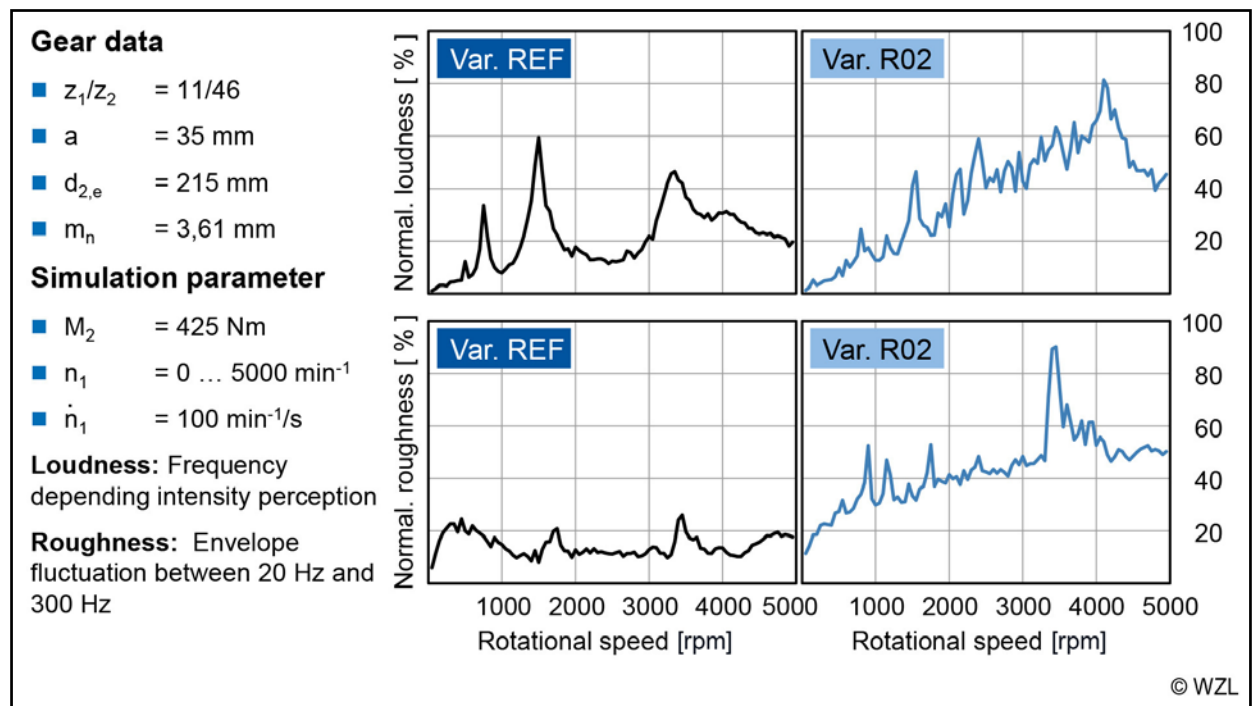


Figure 14 Influence of individual topography deviations on loudness and roughness of differential acceleration.

bolic form in the order spectra. The variant R02 provides completely different noise spectra. Due to the high background noise of the transmission error up to the first gear mesh order, a high excitation of the complete lower order range is given, as the Eigen frequencies are always excited by background noise. The higher harmonics cannot be identified in the spectrum.

The comparison of the order spectra already shows a higher tonal excitation of the reference variant—in contrast to the variant R02 with the topography scatter. In the following, the impact of the different spectra to psychoacoustic parameters is shown.

Loudness is depicted (Fig. 14, top); the loudness for the ref-

erence variants provides clear and dominant peaks for the reference variants when the gear mesh excitation of the first and second gear mesh order is meeting the Eigen frequencies of the system. Especially at $n_1 = 750, 1,500$ and $3,300 \text{ rpm}$ peak values of the loudness are determined. The loudness graph for the R02 variant instead shows almost a linear relationship between loudness and input speed. The peaks of the Eigen frequencies are almost at the same level of the reference variant, but less dominant. Above $1,750 \text{ rpm}$, the loudness of the R02 variant is constantly higher than the loudness from the reference variant.

The roughness for the reference variant is constantly at a

lower level over the complete speed range and thus provides a lower modulation fluctuation in the signal. The R02 variant instead shows as well a linear behavior for the roughness.

The sharpness evaluates the loudness in higher frequency range (Fig. 15). The graphs for the loudness and the sharpness are almost the same. Small differences are only shown above 3,000 rpm.

The psychoacoustic evaluation of the tonality again reveals clear differences between the reference variant REF and random variant R02. The values range of the tonality of the REF variant is, on average, three times higher compared to the R02 variant. The tonality is the only psychoacoustic parameter that is constantly lower for the variant with the topography scatter. However, the loudness and sharpness parameters provide an almost linear graph over the speed while the reference variant provides peak values. In further investigations based on test rig trials, it is planned to measure the dynamic excitation behavior of the manufactured variants and evaluate it by means of differential rotary acceleration, structure-borne noise and airborne noise. Furthermore, a scientific method is needed in order to find the best compromise between a reduction of gear mesh excitation and an increase of background noise.

Summary and Outlook

With this report a procedure is demonstrated for altering the noise behavior of ground bevel gears by applying an individual topography deviation. As such, the major aspect is the reduction of the amplitudes of tooth mesh harmonics and a rise of background noise. Finally, the impact of the manipulated transmission error spectrum on the psychoacoustic parameters is shown.

In a first step, validation of the FE-based TCA *ZaKo3D* for bevel gears is shown. The results prove the capability to display the load-caused contact ratio increase. In particular, the resulting loaded transmission error graphs of the first four tooth mesh orders for drive and coast flank show good agree-

ment between measurement and simulation. Differences in the amplitude values are explained with gear set position deviations caused by the assembly and load-induced displacements, as well as an incomplete itemization of the gear set.

Next a procedure is presented for the reduction of the tonality of the transmission error spectrum. This approach is derived by results discussed (Refs. 10 and 14), as well as analyses done by Landvogt (Ref. 16). With application of a minor topography deviation, from tooth to tooth the regularity of transmission error graphs is disturbed. Two different distributions of topography deviations are simulated and manufactured, as well as a conventional ground reference variant. The validation of the procedure is first done by means of a comparison of the load-free transmission error spectra from TCA and results from a bevel gear tester.

Finally, the gear set variants are virtually tested at operational-relevant parameters with a dynamic simulation (Ref. 4). In the order spectrum of the speed sweep it can be seen that the reference gear set is exciting up to the tenth gear mesh order. In comparison, the variant with fluctuating topography deviation does not provide a dominant excitation by the tooth mesh, but has a higher background noise level. This effect can be even better understood by evaluating the psychoacoustic parameters. On one hand, the loudness shows clear peak values for the reference variant when the gear mesh excitation meets the Eigen frequencies. For the variant with the topography fluctuation there is an almost linear relationship between loudness and rotary speed. Conversely, the higher background noise level of the variation with topography fluctuation has a huge impact on tonality and is three times lower in average compared to the reference variant.

In future research projects the results of the dynamic simulation must be validated by means of test rig trials measuring the differential acceleration, as well as the structure-borne and airborne noise. Furthermore, a method for the reduction of ampli-

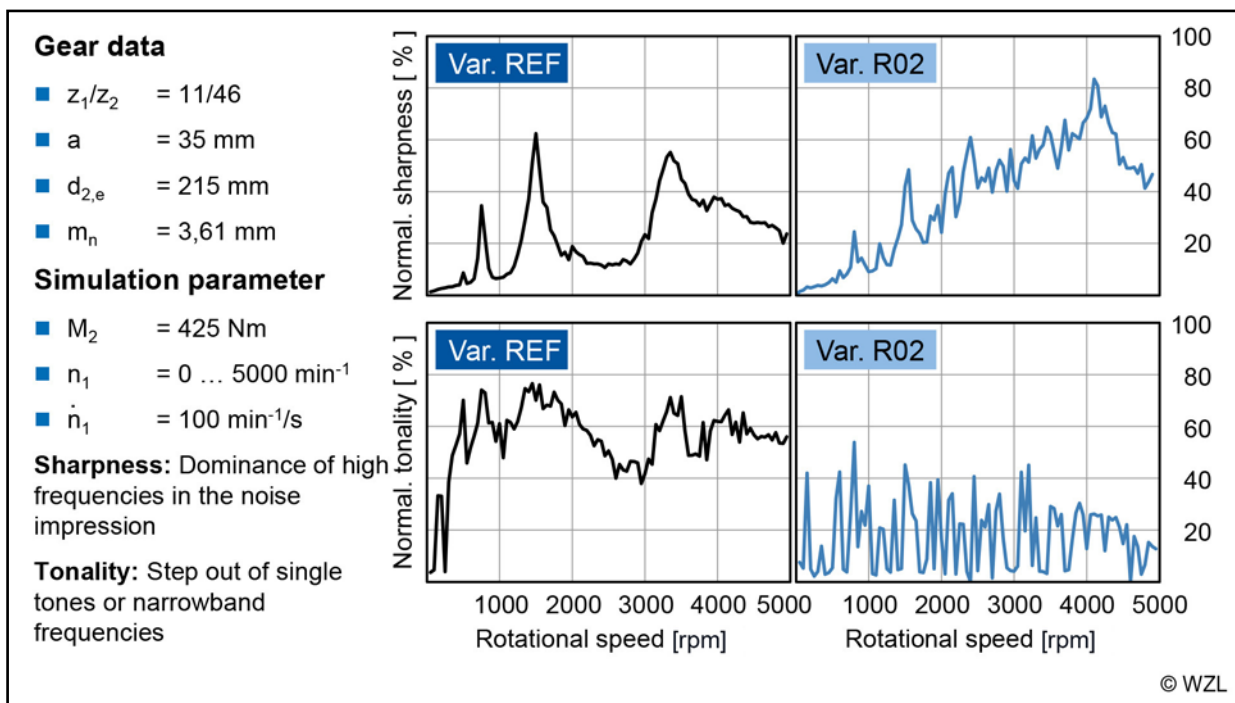



Figure 15 Influence of individual topography deviations on sharpness and tonality of differential acceleration.

tudes of the tooth mesh frequency and the higher harmonics, together with an increase of the background noise level, must be developed. The initial results presented in this report prove that the dominance of gear mesh excitation can be reduced by narrow disturbance of the regularity of the transmission error. 

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