Optimized acoustic design of inequidistant gearings

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Introduction

In state-of-the-art gears the teeth of a gearwheel are of exactly the same size and are positioned exactly with the same pitch. This leads to a periodic excitation of vibrations during the meshing process and, therefore, to tonal noise. Tonal noise is considered more annoying and more prominent than broadband noise. The research group SAM at TU Darmstadt developed an inequidistant gearing aiming at minimizing the overall sound pressure level, the tonality, and the annoyance of gear noise. This new kind of gearing is characterized by uneven tooth widths and uneven tooth positions, leading to an uneven mesh and, therefore, to a less tonal noise. However, the potential to render every tooth of a gearwheel individually in its thickness and its position leads to a challenge in finding the optimal acoustic design. In this work a method is presented to evaluate the noise of inequidistant gears using the results of a psychoacoustic listening test performed in previous works [1]. This evaluation criterion is used to perform an optimization process that comprises a combination of Monte Carlo and neighborhood search algorithms. The optimization method is applied to an automotive electric drive train.

Inequidistant gears

The idea of inequidistant gears is derived from uneven blade positions of fan blowers and uneven tire patterns of vehicle tires. These noise reduction methods are used to reduce tonal noise caused by regular excitation mechanisms. Tonal noise is also a major challenge in gear noise since the perfectly regular design of the gears leads to a periodic vibration excitation and, therefore, to tonal noise. The approach of uneven excitation is applied to spur gears and helical gears in order to fulfill two objectives: the reduction of tonal noise (gear whine) and the reduction of the overall noise level caused by the meshing of the gear wheels. Figure 1 shows an example of a set of inequidistant spur gears. Inequidistant gears are characterized by uneven tooth thicknesses and positions, leading to an uneven meshing of the gears. However, when rendering a gear wheel inequidistant, a meshing pinion must match perfectly in order to avoid any transmission error. A thicker tooth of the gear wheel must always mesh with a wider gap of the pinion and vice versa. The pinion's geometry is derived from the gear wheel's geometry. Hence, in order to fully define the geometry of a set of inequidistant gears, only the gear wheel's uneven tooth thicknesses and tooth positions need to be defined, in addition to the parameters necessary to define the geometry of a regular set of gears.

Figure 1: set of inequidistant spur gears with uneven tooth thicknesses and uneven tooth positions

When designing a set of inequidistant gears, only a certain number of teeth is rendered uneven, called an *inequidistant pattern*. This pattern is repeated several times along the circumference of the gear wheel and the pinion. Figure 2 shows the principle of the pattern-wise design of inequidistant gears. In this example the gear wheel has 25 teeth, whereas the pinion has 15 teeth. The inequidistant pattern consists of 5 teeth. Hence, the pattern is repeated five times along the gear wheel's circumference and three times along the pinion's circumference. Due to the pattern-wise design, inequidistant gears come with the requirement of a common divider in the number of teeth that equals the number of teeth that form the inequidistant pattern.

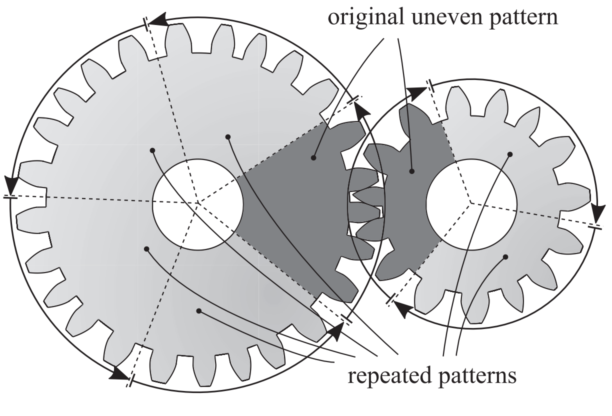

Figure 2: an inequidistant pattern of five teeth is repeated five times at the gear wheel and three times at the pinion

The main excitation mechanism in gear noise is the variable gear mesh stiffness [2, 3]. For regular gears the number of tooth pairs in the mesh alternates periodically. Due to the uneven design of inequidistant gears, their gear mesh stiffness is uneven in terms of shape and amplitude, see Figure 3. The uneven tooth thicknesses cause different stiffnesses of the teeth and, therefore, lead to uneven amplitudes of the gear mesh stiffness. The uneven tooth positions cause the tooth pairs to mesh early or delayed. Hence, the temporal behavior is rendered uneven as well.

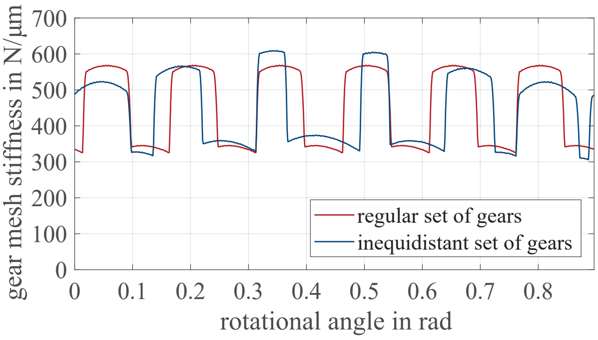

Figure 3: gear mesh stiffness of a regular set of gears and of an inequidistant set of gears

Figure 4 shows an example of the sound pressure spectra excited by a regular set of gears and an inequidistant set of gears. The regular set of gears excites the sound pressure at few distinct frequencies in the spectrum, leading to a tonal noise character. The inequidistant set of gears, however, spreads the sound pressure over many more frequencies. Each of the peaks has a lower (or equal) amplitude than the corresponding peak excited by the regular set of gears. Furthermore, the total sound pressure level (SPL) is decreased from 90 dB (Z) for the regular set of gears to 89 dB (Z) for the inequidistant set of gears. In summary, inequidistant gears are capable of spreading the sound pressure in the spectrum, reducing the amplitude of prominent peaks, and decreasing the SPL. However, the distribution of the sound pressure in the spectrum depends on the specific uneven design of the inequidistant gears. Every change in the teeth's thicknesses and positions will affect the excited gear noise. Hence, finding the optimal design of an inequidistant set of gears is a challenge that might be approached by calculating the excited gear noise, evaluate the noise and iteratively optimize the design.

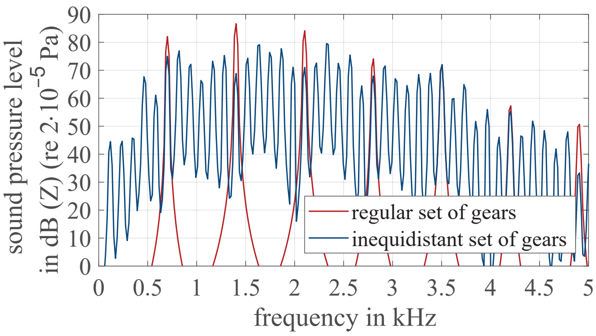

Figure 4: exemplary sound spectra excited by a regular set of gears and an inequidistant set of gears

Calculation of the gear noise

The method to calculate the gear noise of a set of gears is shown in Figure 5. In the static step, the input parameter, such as the number of teeth z, the modulus m_n, the load M, as well as the parameters to define the individual tooth thicknesses Δs and positions Δp are defined. The geometry of the set of gears is calculated. In the elastostatics step, the gear mesh stiffness – the main excitation mechanism – is calculated using the approach of Weber and Banaschek [4]. The dynamic tooth forces are calculated by modeling the set of spur gears as a single mass oscillator, as described in [5], for example. These dynamic tooth forces excite the gear box. The machine acoustic properties of the gear box are determined in the machine acoustic step. As described by Kollmann [6], for example, these properties might be modeled as a transfer function between the force excitation and the radiated sound power. Finally, the sound power is converted to the sound pressure at a point located 1.5 m from the gear box in order to allow for psychoacoustic evaluations.

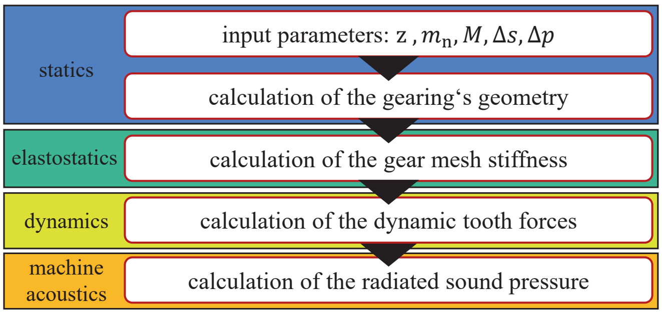

Figure 5: method to calculate the gear noise of a set of gears

Once all the necessary parameters and transfer functions of the calculation procedure are determined, the sound pressure emitted by a gear box can be calculated for various designs of gears.

Noise evaluation

The results of a previously performed online listening test show that the noise of inequidistant gears is perceived less annoying than the sound of comparable regular gears [1]. Three important findings were made regarding the annoyance of gear noise. Gear noise is perceived the more annoying

1. the less the sound pressure is spread in the spectrum,
2. the more prominent single peaks are, and
3. the higher the SPL is.

However, for optimization algorithms, a single value criterion is needed. One approach could be to derive a mathematical criterion for each of the three findings and combine them into one by weighting each criterion. This approach is considered too complex and, thus, may lead to inexplicable results. A simple and comprehensible approach is to use the maximum peak amplitude in the sound pressure spectrum as a criterion to evaluate the annoyance of the gear noise. If the maximum peak in the sound pressure spectrum has a low value

1. this indicates that the sound pressure is spread over many frequencies,
2. the highest peak is less prominent, and
3. this indicates a low SPL.

However, the maximum peak in the sound pressure cannot directly quantify how the sound pressure is spread in the spectrum or the SPL itself, but it can give an indication of these characteristics. Hence, the maximum peak in the sound pressure level is chosen as a suitable criterion to evaluate the annoyance of the gear noise.

Acoustic optimization approach

Performing an acoustic optimization of a set of inequidistant gears is challenging since every tooth of an inequidistant pattern may be rendered uneven in its thickness and its position. In this work, a local search algorithm is used to find the optimal inequidistant design. However, previous works show that many local optima exist for the noise excited by inequidistant gears [7]. Therefore, a suitable starting point for the local search algorithm is necessary. In a first optimization step, a random-based Monte Carlo optimization algorithm is used to determine a starting point for the local search performed in the second optimization step.

Application of the methodology to an electric drive train

The calculation procedure and the optimization approach are applied to a state-of-the-art electric drive train, as shown in Figure 6. The concept of the drive train is taken from the research project Speed2E [8]. It comprises two electric engines with a rotational speed of up to 30000 rpm. The first stage of the gear box part I originally was designed as a low-noise gearing, see the green set of gears in Figure 7. Even though it already is a low-noise design, this set of gears is still to be virtually replaced by an inequidistant set of spur gears in order to reduce the gear noise even more.

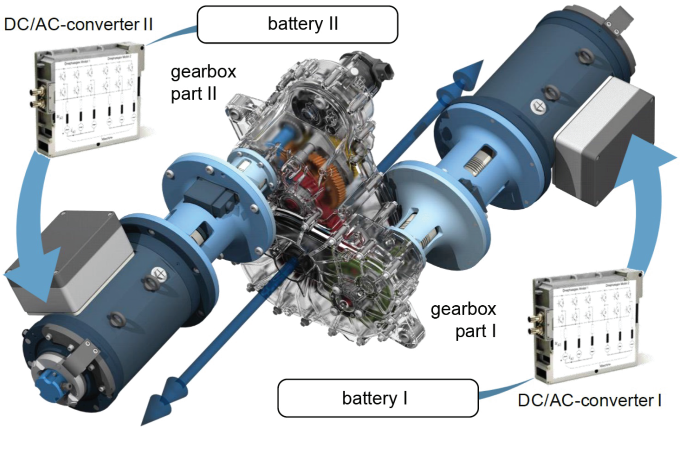

Figure 6: state-of-the-art drive train taken from the research project Speed2E (based on [8])

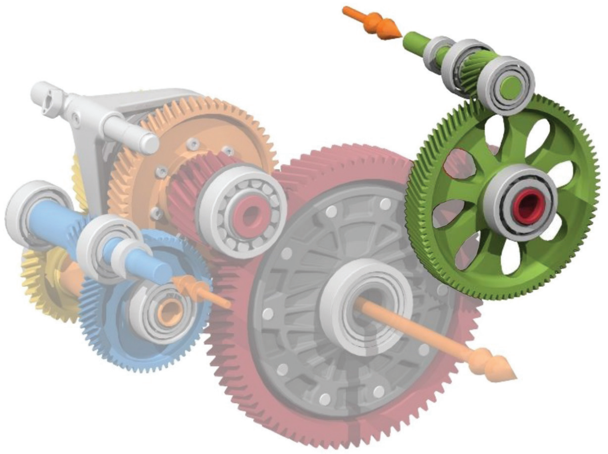

Figure 7: gears in the electric drive train; first stage of gear box part I highlighted in green (based on [8])

The machine acoustic properties of the gear box housing are determined by performing a numerical harmonic analysis. The gear box housing is excited at the bearing seats and the normal surface velocities are determined. A monopole is used as a simplified radiation model.

Optimization results

The acoustic performance of the acoustically optimized set of inequidistant gears is compared to that of the regular low noise set of gears. Figure 8 shows the geometries of the regular set of gears and the acoustically optimized inequidistant set of gears. The acoustically optimized inequidistant set of gears is uneven in its tooth thicknesses and positions. Some of the pinion's tooth spaces are very narrow (red box in Figure 8), leading to thin teeth at the gear wheel. In order to limit the decrease in load capacity due to very thin teeth, a maximum deviation in tooth thickness of ±20 % was set as a limit value for the optimization. Figure 9 shows the Campbell diagram of the sound pressure level excited by the regular set of gears for rotational speeds from 0 to 30000 rpm and a load of $M = 50\,\mathrm{N\,m}$. The regular set of gears excites distinct and prominent tones. The resonance of the set of gears is at about 11 kHz, indicated by the increased amplitudes at this frequency. Figure 10 shows the Campbell diagram of the sound pressure level excited by the acoustically optimized set of inequidistant gears for the same rotational speed and load. Compared to the regular set of gears the acoustically optimized set

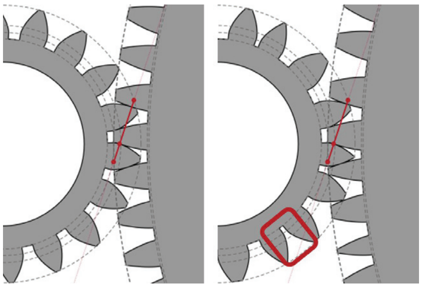

Figure 8: geometries of the regular set of gears and the acoustically optimized inequidistant set of gears

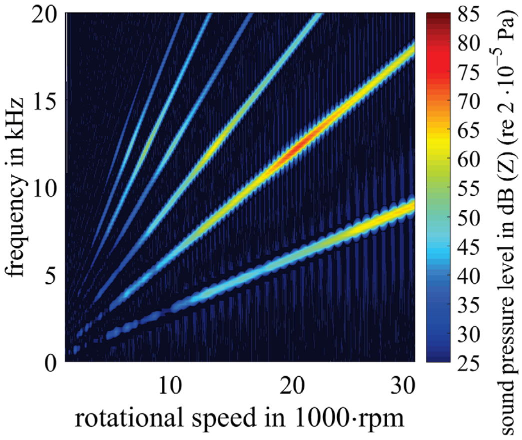

Figure 9: Campbell diagram of the sound pressure level excited by the regular set of gears for rotational speeds from 0 to 30000 rpm and a load of $M = 50\,\mathrm{N\,m}$

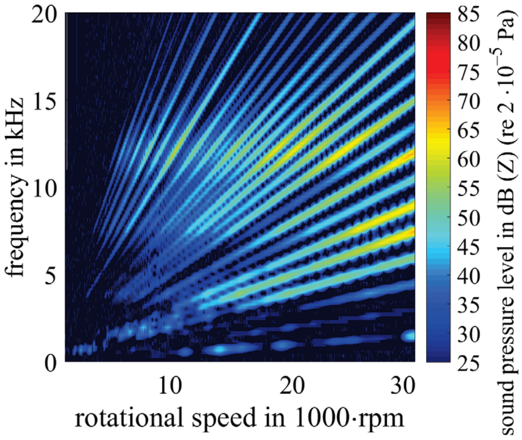

Figure 10: Campbell diagram of the sound pressure level excited by the acoustically optimized set of inequidistant gears for rotational speeds from 0 to 30000 rpm and a load of $M = 50\,\mathrm{N\,m}$

of inequidistant gears spreads the excited sound pressure over many more frequencies, with a lower sound pressure each. Figure 11 shows the SPL for both sets of gears. For many rotational speeds the SPL excited by the optimized set of inequidistant gears is lower than that of the regular set of gears. For the rotational speed of 20000 rpm, the SPL is decreased from 73 dB (Z) to 66 dB (Z), for example. However, there are also some rotational speeds with an increased SPL. The calculation results confirm that acoustically optimized inequidistant gears are capable of reducing the gear noise even compared to state-of-the-art low noise gears.

Summary and conclusions

In this paper a method to calculate the sound pressure levels excited by gears is presented. The method comprises static, elastostatic, dynamic, and machine acoustic modeling approaches and can be used to calculate both the sound pressure generated by regular gears and by inequidistant gears. A criterion is presented to evaluate the annoyance of gear noise, based on the amplitude of the maximum peak in the sound pressure spectrum. The approaches are used to perform an acoustic optimization of a state-of-the-art electric drive train. The results show that acoustically optimized inequidistant gears are capable of reducing the noise of gears in terms of the annoyance (evaluated by the criterion introduced) and the total sound pressure level.

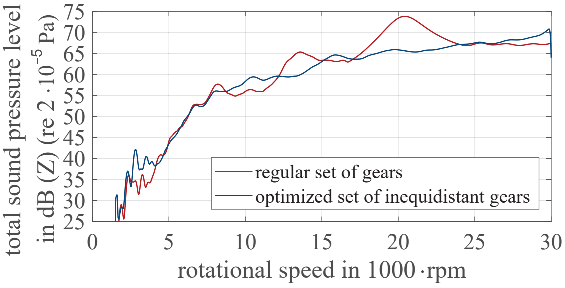

Figure 11: SPL excited by the regular set of gears and by the acoustically optimized set of inequidistant gears for rotational speeds from 0 to 30000 rpm and a load of $M = 50\,\mathrm{N\,m}$

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References

[1] P. Neubauer, J. Bös, and T. Melz: Online listening test to evaluate the gear mesh noise of inequidistant gearings, in: proceedings of the DAGA 2018 – 44. Jahrestagung für Akustik, 03/19–22/2018, Munich, Germany, pp. 1065–1068, 2018.

[2] P. Zeller (ed.): Handbuch Fahrzeugakustik (Handbook of Vehicle Acoustics), Vieweg+Teubner Verlag, Wiesbaden, 2012.

[3] M. Pflüger, F. Brandl, U. Bernhard, and K. Feitzelmayer: Fahrzeugakustik (Vehicle Acoustics), Springer-Verlag, Vienna, New York, 2010.

[4] C. Weber and K. Banaschek: Formänderung und Profilrücknahme bei gerad- und schrägverzahnten Rädern (Shape adjustments and profile modifications of spur gears and helical gears), Schriftenreihe Antriebstechnik, 1953 (11), pp. 1–17, 1953.

[5] S. Radev: Einfluss von Flankenkorrekturen auf das Anregungsverhalten gerad- und schrägverzahnter Stirnradpaarungen (Influence of flank modifications on the excitation behavior of spur gears and helical gears), PhD dissertation, TU München, Germany, 2007.

[6] F. G. Kollmann: Maschinenakustik (Machine acoustics), Springer-Verlag, Berlin, Heidelberg, 2000.

[7] H. Diefenbach: Development of a method to optimize gear noise emitted by an inequidistant gearing, Master Thesis, TU Darmstadt, Germany, 2018.

[8] K. Stahl and P. Fischer: Schlussverbundbericht Speed2E (Final report Speed2E), Munich, Germany, URL: `https://doi.org/10.2314/GBV:1017888809` (accessed 02/26/2019), 2017.