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A Fully Integrated Simulation Approach of Drive Trains towards Tonality Free Wind Turbines

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Abstract

Wind turbine manufacturers are now facing stricter noise regulations as turbines are being placed closer to urban areas. While rotor blade aero-acoustic noise is the dominant noise component in the total sound pressure level of a wind turbine, mechanical noise originating from gears and generators are gaining importance as well, especially when producing tonal noise. To minimize the tonal noise originating from the gears it is crucial to compare different drivetrain concepts and gear designs early in the design phase and consider the NVH performance as part of the design selection criteria. Unfortunately, the software available on the market allows the prediction of the acoustic performance of a drivetrain design when all details are already known, but these software's are unable to do this in the early concept design phase when making design choices have the biggest impact. Therefore, an integrated simulation platform has been developed to predict wind turbine tonality, considering factors like gear mesh excitation and aero-acoustic masking, which is already applicable in the design concept phase when lots of design details are not known yet. This platform, linked to optimization software, allows ZF to proactively develop strategies to mitigate Noise, Vibration, and Harshness risks, resulting in cost-effective and harmonized solutions across their product platforms.

Keywords

energy, wind turbines, drivetrain, gears, NVH, FEA, vibration, acoustics, ROM

1 Introduction

As onshore wind turbines are installed closer to urbanized areas, wind turbine Original Equipment Manufacturers (OEMs) are being faced with increasingly stringent noise regulations. On top of the compliance with acoustic regulations, gearbox and drivetrain suppliers are being faced with many other challenges. These include torque density increase measures, drastically increased power and torque levels of onshore wind turbines and shorter design cycles. Moreover, gearbox and powertrain designs need to fit within so-called product platforms, see Figure 1.

Aero-acoustic noise originating from the rotor blades is the dominant source of total wind turbine noise. Although mechanical noise is not determining the total wind turbine noise, it could still result in non-conformity to local noise regulations when it contains audible tonal components. Noise sources which produce mechanical noise are among others gears, the generator, etc. These noise sources excite at discrete frequencies and induce vibrations which are propagated throughout the drivetrain and radiated by the exterior of the wind turbine. The mechanical noise is only to be found critical when it exceeds the masking level (in essence the rotor aero-acoustic noise) and thereby producing tonal noise. Wind turbine acoustic noise is evaluated according to the international standard IEC 61400-11 [1].

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Major design levers with respect to wind turbine acoustic noise emissions are being decided upon in the concept phase, where the impact on cost savings is large but where there is uncertainty about some design parameters. Thus, it is key to be able to compare different drivetrain concepts, gear designs and other design parameters related to the interaction with the system level in a quick and easy way: the design target is to “vibro-acoustically” tailor the gearbox to the customer wind turbine.

This requires a holistic simulation approach, involving all aspects of the tonality prediction being:

- gear mesh excitation,
- vibro-acoustic transfer paths from the gear mesh contacts throughout the wind turbine to the IEC microphone(s),
- aero-acoustic masking originating from the wind turbine rotor.

Therefore, an integrated simulation platform has been developed which combines these different aspects involved in predicting wind turbine tonality. This simulation platform is linked to a library with fully parameterized conceptual drivetrain multi-body models, see Figure 2. This allows the evaluation of NVH risks of potential drivetrain configurations as well as different gear macro- and µ-geometry options.

The simulation platform is also linked to optimization software enabling structural optimizations of drivetrain components with respect to stiffness, mass, and inertia into their feasible design ranges. Structural optimizations are being performed using either of two approaches:

- parameter sensitivity studies,
- design-of-experiment studies followed by surrogate modeling of the data.

This enables ZF to develop Noise, Vibration and Harshness (NVH) risk mitigation strategies pro-actively in an early phase of the design leading to less expensive and better integrated solutions which can be implemented throughout their product platforms.

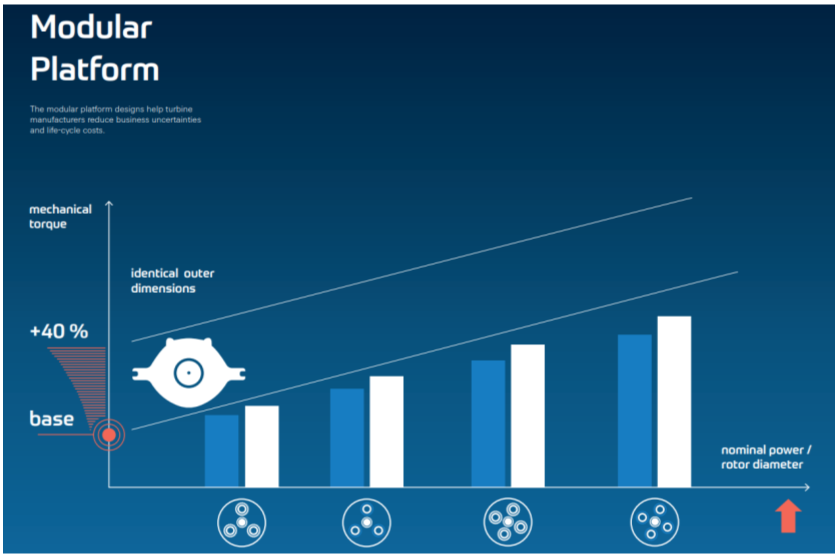

Figure 1. Modular gearbox platforms.

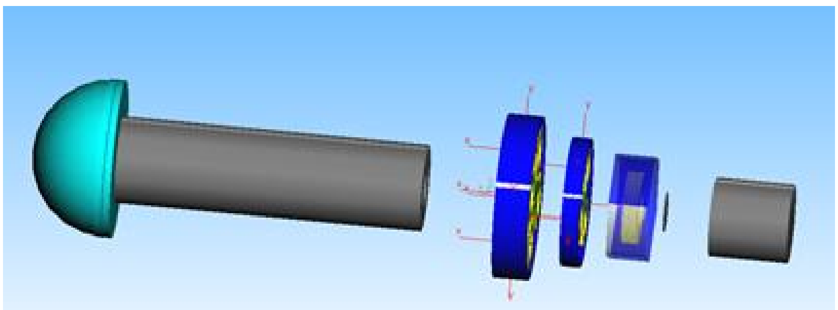

Figure 2. Fully parameterized conceptual drivetrain model.

2 Wind turbine acoustics

The total wind turbine noise is dominated by aero-acoustic rotor noise, where the largest contribution is originating from the down-going blade at 70% - 90% radius. Different sources contributing to aero-acoustic rotor noise are (displayed in Figure 3.a):

- turbulent-boundary layer trailing-edge noise,
- tip vortex formation noise,
- flow separation noise,
- trailing-edge-bluntness vortex shedding noise,
- turbulent inflow noise caused by scattering of turbulent wind fluctuations by the rotor blades.

The turbulent-boundary layer trailing-edge noise, see Figure 4, is the dominant noise source and is broadband by nature.

The total Sound Pressure Level due to turbulent-boundary layer trailing-edge noise can be described by the following equation [2]:

$$SPL_{1/3} \approx 10.log\left(U^5 \frac{\delta.L}{r^2}\right), \tag{1}$$

where SPL1/3 refers to the 1/3 octave Sound Pressure Level, U to the free-stream velocity, δ to the boundary-layer thickness, L to the aerofoil span and r to the distance of the source to the receiver. Therefore, one can conclude that the wind turbine aero-acoustic masking energy (M.E.) is dependent on:

- rotor speed (Ω)
- rotor size (D), and
- tower height (H).

Wind turbine tonal noise is originating from mechanical noise. Typical sources are gears, generator, cooling fans, etc. This mechanical noise can be either structure-borne or airborne typically being dependent on the frequency range. In the case of structure-borne noise originating from gears, the induced vibrations are amplified and transferred over the drivetrain towards tower, rotor or nacelle and then radiated towards the wind turbine surroundings. The tonal content is related to the gear mesh base frequencies and their higher harmonics but are not determining the wind turbine overall Sound Pressure Level. A Tone Level (T.L.) is only to be found critical when it protrudes from the (aero-)acoustic masking M.E. of the wind turbine, see Figure 3.b. Therefore, the Tonal Audibility (T.A.) can be described by ([1],[3]):

$$T.A. = T.L. - M.E. \tag{2}$$

$$T.A. = (T.E.).FRF_{drivetrain}.FRF_{wind\ turbine} - M.E. \tag{3}$$

The gear excitation is typically characterized by Transmission Error or T.E. which is defined as the difference between the driving and ideal position of the driven gear, where the latter position refers to a situation determined by the gear ratio and a perfectly conjugate mesh action without error deflections [4]:

$$T.E. = \theta_2 - \frac{z_1}{z_2}.\theta_1, \tag{4}$$

where θ_1 is the angular position of the driving gear, θ_2 is the angular position of the driven gear, z_1 is the number of teeth of the driving gear and z_2 is the number of teeth of the driven gear. T.E. is depending on torque (T), gear misalignment, machining errors, assembly errors and the frequency (f).

The transfer path between the gear meshes and the IEC position can be characterized by means of the combined transfer functions of the drivetrain $FRF_{drivetrain}$ and the wind turbine $FRF_{wind\ turbine}$ which are frequency and load dependent. Gears excite along the gear mesh orders: these orders depend on gear tooth counts and the rotational speed of the drivetrain. Therefore, Gear Tooth Frequencies (GTF) can be described as [5]:

$$GTF_{stage\ k} = q.\Omega.\prod_{p=1}^{k} i_{stage,p}.z_{RNG,k}, \tag{5}$$

where q refers to the q^{th} harmonic, Ω is the rotor speed, k refers to the k^{th} gear stage, $i_{stage,p}$ is the gear stage ratio of stage p and $z_{RNG,k}$ is the tooth count of ring gear stage k. As the aero-acoustic masking energy is dependent on wind turbine rotor speed, it is consequently also related to the gear mesh orders.

Wind turbine acoustic noise is evaluated according to the international standard IEC 61400-11 [1], where the acoustic measurement takes place at a downwind position with a distance relative to the size of the wind turbine, see Figure 5.

The reference distance R_0 according to IEC at which the acoustic measurements shall be taken for horizontal axis wind turbines is described by:

$$R_0 = H + \frac{D}{2}, \tag{6}$$

where H is the tower height and D is the rotor diameter. This results in a direct relation of SPL measured at the IEC position to the wind turbine Sound Power Level (PWL) and wind turbine dimensions [6]:

$$SPL = PWL - 11dB - 20.log(R_0). \tag{7}$$

From these equations one can conclude that e.g., increasing rotor sizes lead to higher masking energy M.E., but this needs to be evaluated for an IEC microphone located at a further distance, based on tower height and rotor size, which subsequently reduces the M.E. at the microphone.

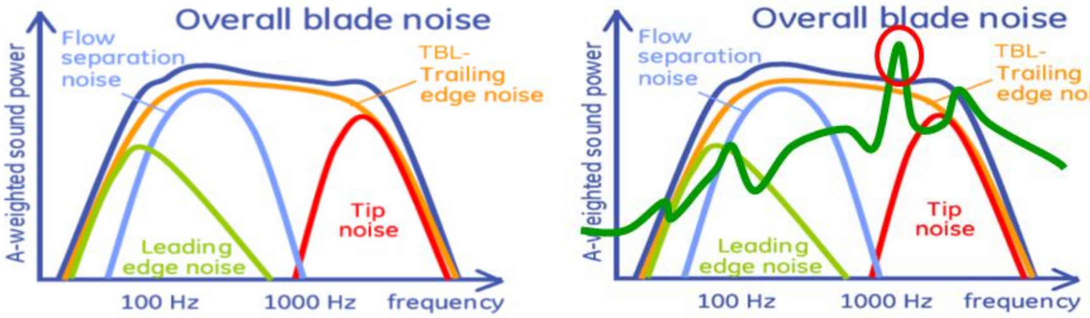

Figure 3.a Contributions to overall wind turbine rotor noise [7], b. Illustration of tonality.

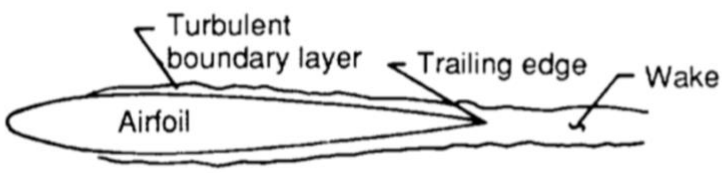

Figure 4. Turbulent-boundary layer trailing-edge noise (source [2]).

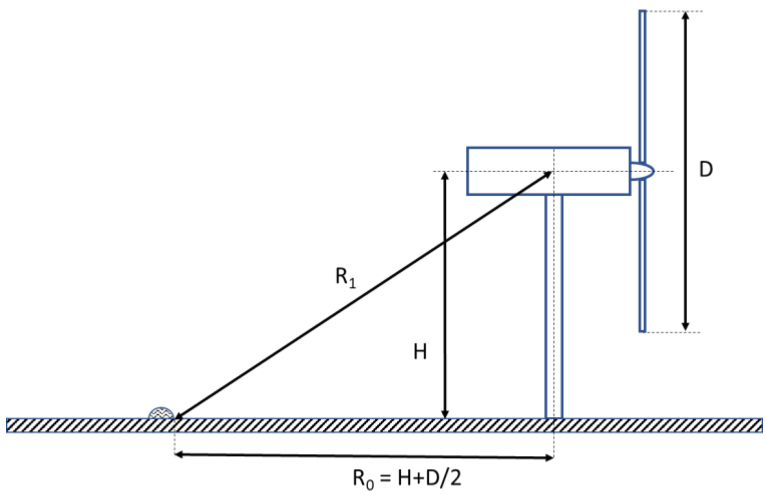

Figure 5. Definition of the IEC reference distance for horizontal axis wind turbines at which acoustic measurements shall be taken.

3 Integrated simulation platform

The core of the NOise and Vibration Analytics (NOVA) simulation platform is a Python package connected to a PostgreSQL database which combines the inputs from different CAE software packages to predict tonality, see Figure 6.

The NOVA simulation framework combines the influences of:

- Aero-acoustic masking,
- Gear excitation,
- Drivetrain transfer functions from gear mesh contacts to drivetrain interfaces,
- Wind turbine transfer functions from drivetrain interfaces to the IEC microphone,

and enables the quick and easy comparison of different wind turbines within the same platform or among different customers. The integration of optimization software enables the optimization of gear excitation and powertrain dynamics to achieve “vibro-acoustic” tailoring of powertrains to wind turbines.

The NOVA simulation framework makes use of a model library containing the gearbox and drivetrain architectures currently in use in the ZF product portfolio. These are:

- Combinations of planetary and helical gear stages,
- Integrated, partially integrated or non-integrated powertrains.

For the drivetrain simulation, low fidelity multi-body models are used which are fully parametrized. The models allow both simulations in frequency and time domain and are used to characterize the drivetrain transfer function “FRFdrivetrain”, see Figure 7.

For the wind turbine transfer path “FRFwind turbine” validated transfer functions are used to describe the relation between the dynamic drivetrain loads to a resulting SPL of the wind turbine at the IEC microphone location. If this is not provided by the wind turbine OEM, a generic model is used to calculate a most precise estimate.

As state-of-the-art wind turbines are variable speed machines, the tonality needs to be evaluated for several wind turbine operating modes over the entire speed range:

- Wind turbine nominal operating mode, i.e., normal operation,
- Wind turbine low-noise modes, with reduced power output which are used to limit the total wind turbine SPL, e.g., during nighttime.

The tonality level along different gear mesh orders for a specific wind turbine operating curve can then be evaluated in a dashboard, see Figure 8.

To enable tailoring a gearbox or drivetrain to a wind turbine, the NOVA workflow is extended to allow for a more detailed inclusion of the wind turbine dynamic behavior and acoustics into the prediction. This is done by solving a model of the tower and a model of the drivetrain completely independently and merging the results using a frequency based sub-structuring approach [9]. The main advantage is that the expensive structural tower and acoustic model do not need to be solved repetitively to analyze parameter changes in the drivetrain. In consequence, drivetrain parameter optimizations or the influence of wind loads acting on the blades can be investigated very rapidly. While the initial simulation (drivetrain + tower structural + acoustic model) finishes within days, the influence of a different wind load could be studied within minutes (solving tower and drivetrain model), the influence of drivetrain parameters even within seconds (solving only drivetrain model). This workflow allows for example to compare the transfer functions from the gear mesh to an average IEC microphone, as shown in Figure 9 for the example of a 120 m high tower with double-row roller bearings for the yaw bearing and closed yaw brakes.

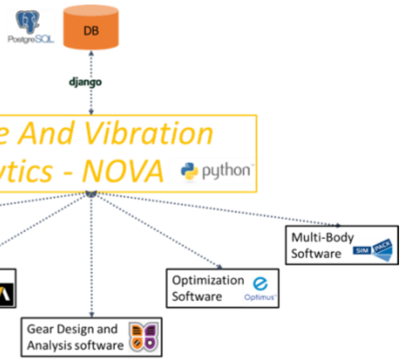

Figure 6. Noise And Vibration Analytics framework.

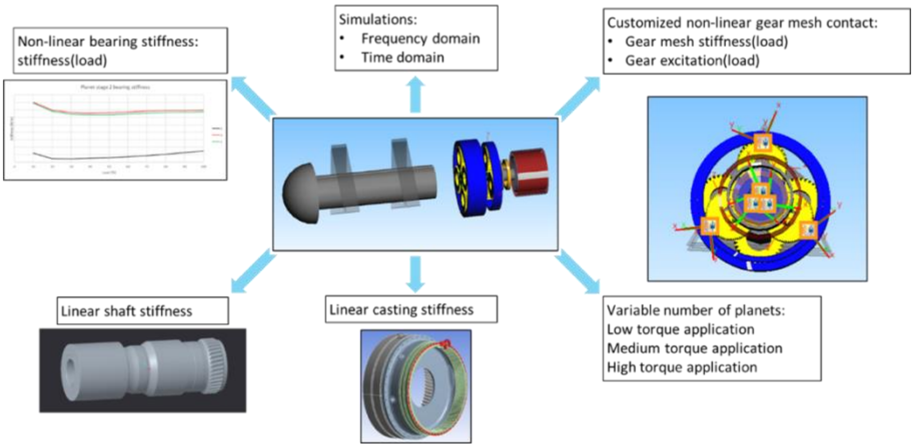

Figure 7. Fully parameterized wind turbine drivetrain model.

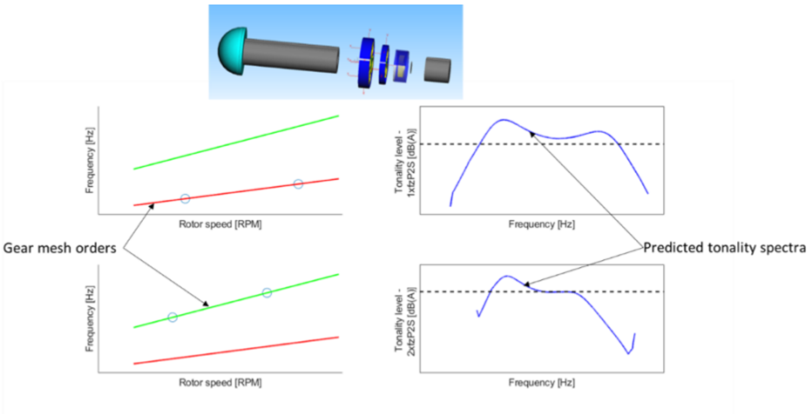

Figure 8. Tonality dashboard.

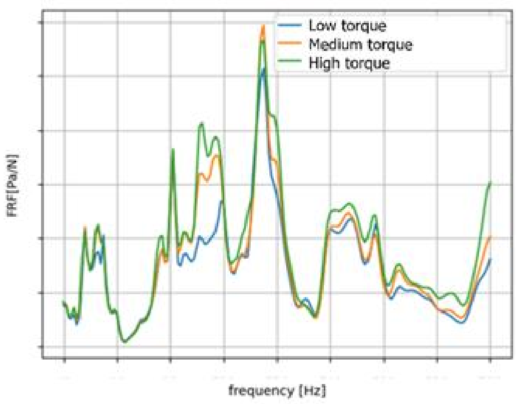

Figure 9. Non-liner transfer function from gear mesh to IEC microphone in function of torque load.

4 Drivetrain optimization

Drivetrain NVH performance is optimized with respect to gear excitation and structural dynamics.

Gear excitation can be optimized by proper selection in between different gear macro-designs and the optimization of the gear tooth µ-geometry. Usually gear excitation is optimized such that a dominant drivetrain resonance and a minimum gear excitation occur at the same wind turbine operating condition.

Structural dynamics are optimized to:

- Reduce the amplification of dominant drivetrain modes,
- Shift dominant drivetrain modes outside the wind turbine operating range,
- Shift dominant drivetrain modes to coincide with minimum gear excitation.

Structural optimizations are being performed using either of two approaches:

- parameter sensitivity studies,
- design-of-experiment studies followed by surrogate modelling of the data.

With parameters sensitivity studies component stiffnesses, masses and inertias are varied in between the lower and the upper limit of the design space. This is done for one parameter at the time and indicates what the optimal component stiffness, mass and inertia will be. This approach is applied here to a sun shaft where the most flexible and the stiffest version are shown within the available Volume Of Control, see Figure 10.

A sensitivity study on the sun shaft stiffness and its influence on tonal audibility is shown in Figure 11.

In this example, the sensitivity analysis clearly indicates the trends of how predicted tones are shifting both in frequency and amplitude by changing the sun shaft torsional stiffness. For this case, making the sun shaft design stiffer leads to reduced wind turbine Tonal Audibility levels.

The second approach makes use of design-of-experiments (DOE's) followed by surrogate modelling of the data and multi-parameter optimization. The approach is visualized in Figure 12.

When applying this to the gearbox design one can compare two different optimization approaches in order to achieve a gearbox design enabling a tonality free wind turbine, i.e. get the dynamic drivetrain loads originating from stage 2 and stage 3 below the masking noise balanced target line. The masking noise balanced target line is indicated as a skewed blue dotted line. The original spectrum as well as the optimized spectra are shown in Figure 13.

From this comparison one can observe that both optimization methodologies reach the tonality free target and a slightly better performance of the Differential Evolution Algorithm (DEA) compared to the Sequential Quadratic Programming (SQP) algorithm.

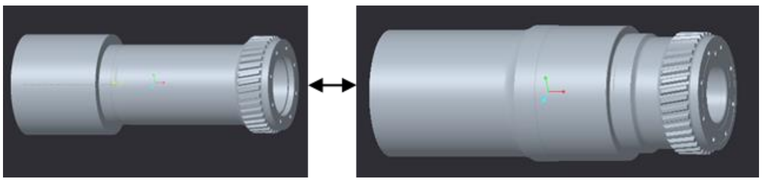

Figure 10. Most flexible (left picture) and stiffest (right picture) sun shaft design possible within volumetric constraints

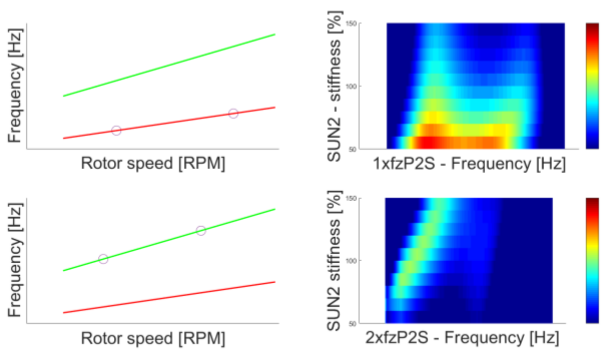

Figure 11. Predicted tonality spectra in function of sun shaft stiffness.

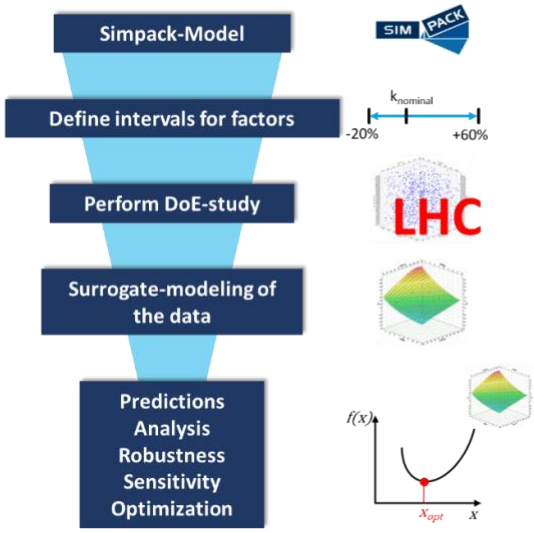

Figure 12. DOE workflow [9].

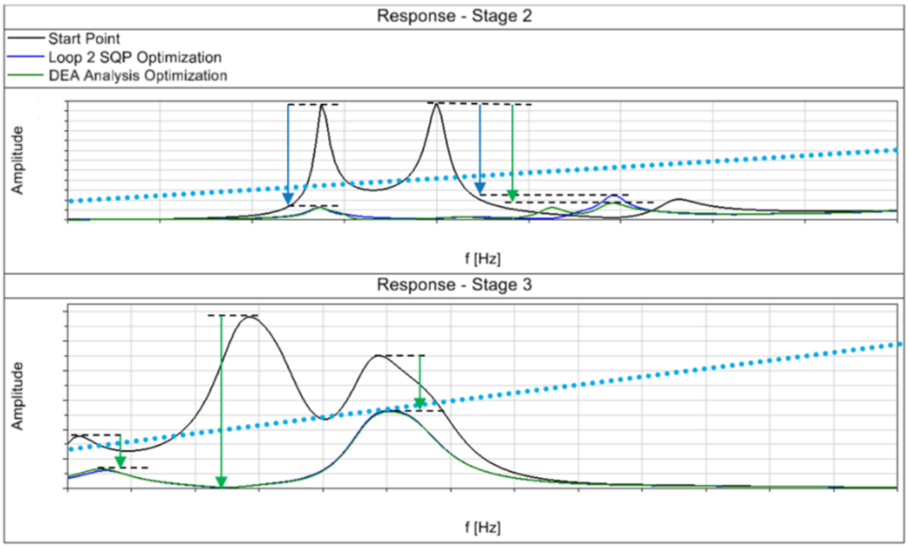

Figure 13. Comparison of optimization approaches to achieve a gearbox design enabling a tonality free wind turbine.

5 Conclusions

Using a holistic simulation approach, involving all aspects of the tonality prediction, enables ZF to develop optimized gearbox and drivetrain designs with respect to NVH. This can be done pro-actively in an early phase of the design leading to less expensive and better integrated solutions which can be implemented throughout their product platforms.

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