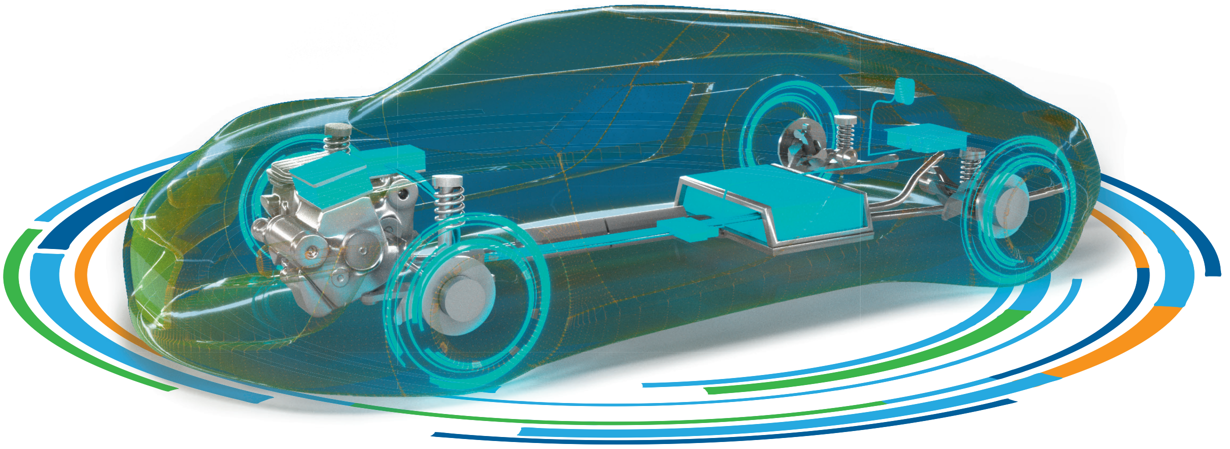

INTERNATIONAL SYMPOSIUM ON DEVELOPMENT METHODOLOGY

Gear Noise Reduction for an e-Axle

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1 Introduction

With upcoming electromobility applications, an e-Axle is needed for Battery Electric Vehicles, as well as for Fuel Cell Vehicles. Also, for Plug in Hybrids the silent drive in electric mode is perceived as a positive feature for e-drives.

The challenge for the transmission, which comes along with this feature, is that gear noise is not masked any more by combustion noise. The noise can be mitigated by proper design of the gearbox and consideration for the gear teeth meshing.

Typically, the gear wheels have corrections in order reduce transmission error. The wide range in e-motor torque leads to the problem, that the flexible system has different deformations. The tooth corrections are ideal for one torque only. The challenge is to find corrections which are stable over the entire operating range. Stable corrections lead to larger allowable tolerances, which reduces costs.

Figure 1 shows such an e-axle design. Different noise sources are present in such an electric axle. Major contributions come from gear mesh excitations and from electromagnetic excitation acting directly on the stator. The gear wheel excitations are typically transferred via the shafts and the bearings to the housing, and lead to an excitation of the housing. The transfer path is modelled in detail with the Multi – Body – Dynamic (MBS) approach. Additional excitation from the bearings can be considered as well. For high frequency NVH problems, the MBS model deals with flexible bodies for the housing as well as for gear shafts and mounting system [1]. Typically, these bodies are represented by condensed FEM models. [2]

Undesired noise amplifications typically have two sources. A high excitation or a high amplification due to modes of the housing structure. In e-axles, it is difficult to avoid the excitation of local modes of the housing because of the whine noise characteristic. It is a tonal noise where the excitation frequencies are proportional to the speed of the axle. The frequency range for mechanical excitation is from 0 to 10 kHz and possibly higher. Therefore, it is of high importance to avoid undesired amplifications, as gear housing designs cannot avoid local excitation modes at such frequencies.

Besides a proper gear layout (macro geometry), the micro geometry of the gear mesh can help to reduce the force fluctuations in the gearing. The force fluctuations are the cause of the excitation. Very often the transmission error is used as a measure for these force fluctuations as it is commonly used in gear noise assessment by measurements.

Noise is typically assessed in measurement with microphones for the sound pressure level and with accelerometers for the structure borne noise. Due to the local modes of the housing, the positions of the accelerometers are of high importance. Simulation can help to define good positions for

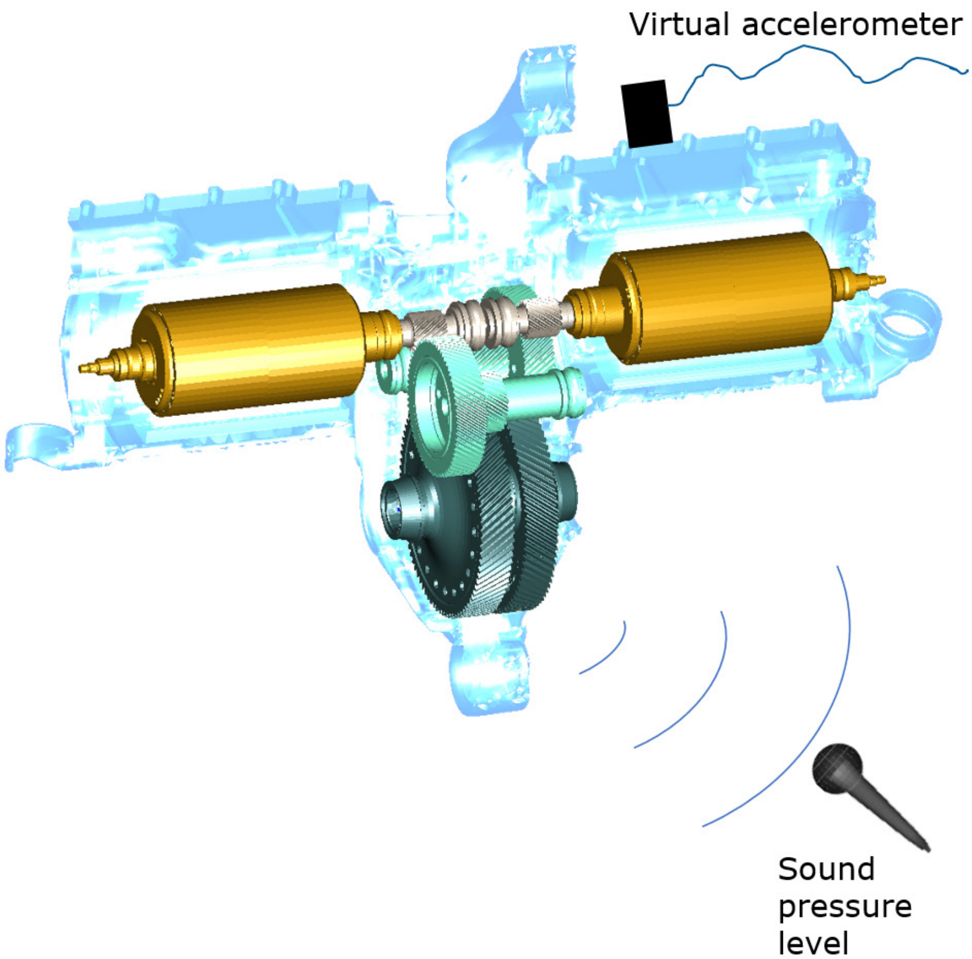


Figure 1: Typical e- Axle and the noise emission principle

accelerometers to cover the most important vibration profiles. In simulation these are also known as virtual accelerometers.

It is possible to use calculated sound pressure levels out of simulations for noise assessment, but it is not very efficient due to the large amount of calculation data and long simulation times. Also, the integral surface accelerations of the radiating housing could be used, what avoids already simulation time for air borne noise calculation, but this still leads to a large amount of data.

Within this work, we use the virtual accelerometers which can be placed properly with a few simulations and recalculation to the entire surface acceleration levels. The goal is to find regions with the highest response amplifications to place the virtual accelerometer. This is by the way also valid for placing the real accelerometer later in the development process.

The gear noise is characterized by specific orders and therefore most of the gearings can by analyzed independently from other gear stages. In the current work, simulation times were shorted by focusing on the NHV relevant surface accelerations in the 38th and 76th order of the gear stage under investigation. Design Space Exploration approaches [3] with proper KPI calculations were also used to get - with a comparable low number of simulations runs - a design decision for the gear micro geometry, which is at the same time robust against costly manufacturing tolerances.

2 Development environment & development methodology

2.1 State of the art

In the past, the gearbox simulation was performed for some of the design variants only. A systematic variation of the influencing parameters was avoided as the time effort

a) to simulate

b) to check the large amount of output data

was so high, that rough engineering judgement was used instead to get a first decision. In case of NVH problems in a later stage of the product development cycle, additional components can be used to mask noise either by applying vibration absorbers or by noise insulation. A better and more cost-effective design of the remaining gear transmission error over the whole Speed / Torque area in combination with the gear housing was typically shifted towards the next generation design.

2.2 Methodology and Tool interaction proposal

The proven Methodology of high-quality physical simulation is combined with a model-based, robust optimization process for reducing the whine noise:

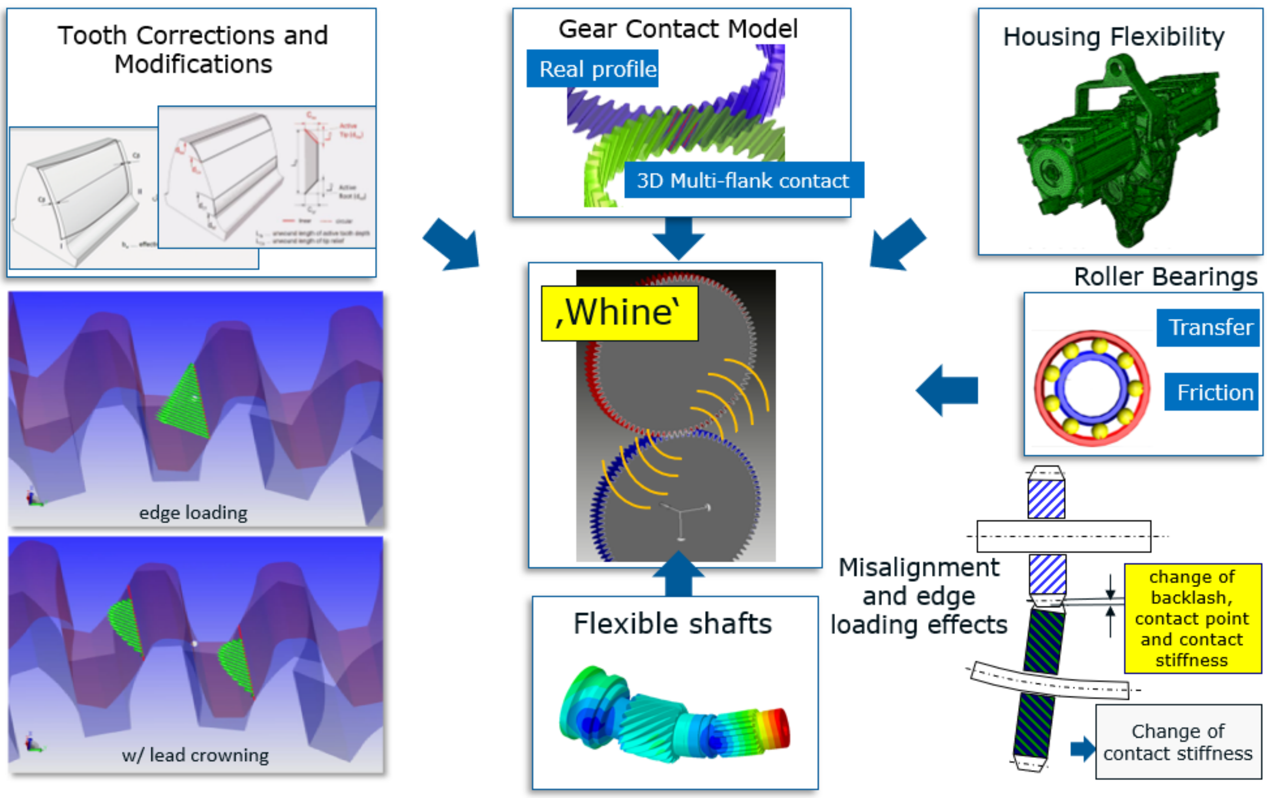


Figure 2: Physical modelling of e- Axle noise generation and emission

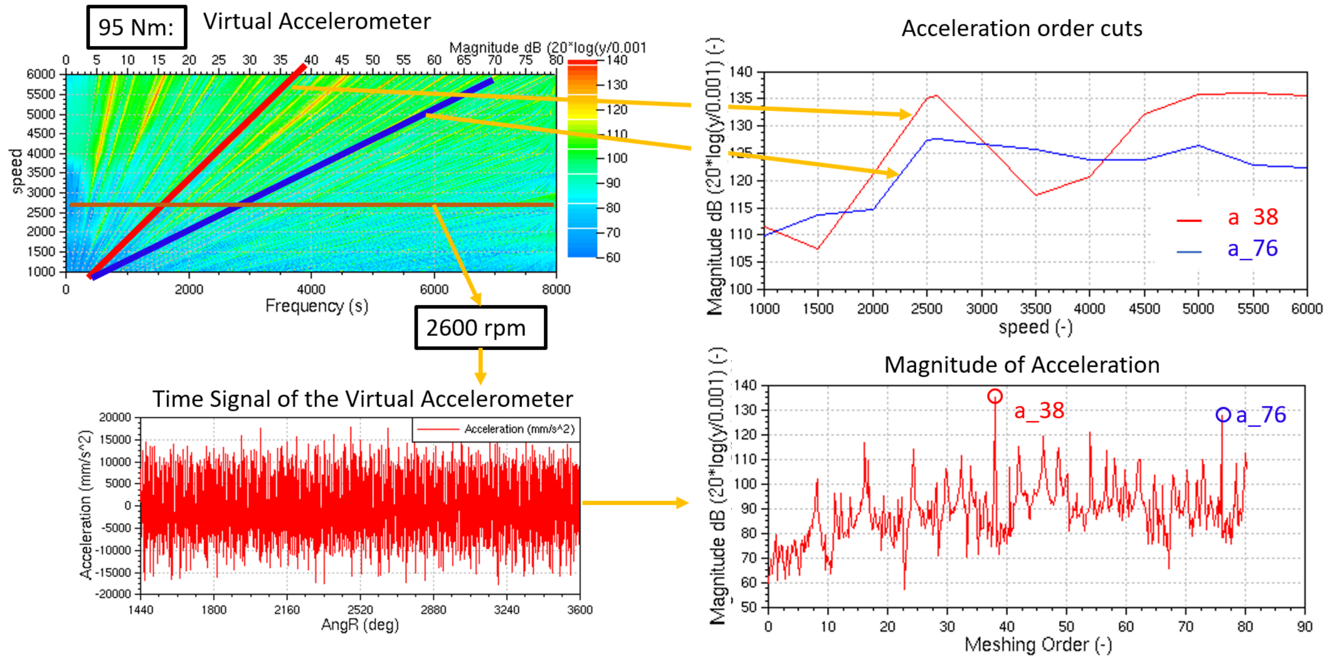


Figure 3: Surface vibrations as function of time, Campbell and Order Cuts for first KPI generation ("a_38" and "a_76") e.g. at 2600 1/min, 95 Nm

Figure 2 shows the physical modelling of the gear whine within the MBS-tool. A central feature for this is a highly detailed gear contact simulation [4] which is able to simulate the time resolved gear contact geometry. The gear contact is influenced by deflections of the shafts which can come from shaft bending or general misalignment due to deflection of the housing under load. As the gear flank profiles are brought into contact, gear profile modifications can also be considered. The force fluctuations from the gear mesh are transferred via the shafts, bearings, splines etc. to the bearings supporting the system in the housing. The forces excite the housing which lead to deflections of the housing, which influences the gear contact, as well as leading to vibrations of the entire housing. These vibrations can have a larger effect, when occurring close to the mounts of the e-axle. In addition, local surface vibrations influence the radiated noise.

This study focuses on 2 commonly used gear corrections:

- *Lead Crowning*
- *Profile Tip Relief*

Nevertheless, the described workflow can utilize more parameters as well.

In order to assess the unwanted whine, the surface acceleration of the housing in a certain hot spot area is calculated. Figure 3 shows the time trace and the order analysis of this response for a Speed / Torque combination (2600 1/min, 95 Nm) where the 38th order excitation meets an Eigenmode of the housing.

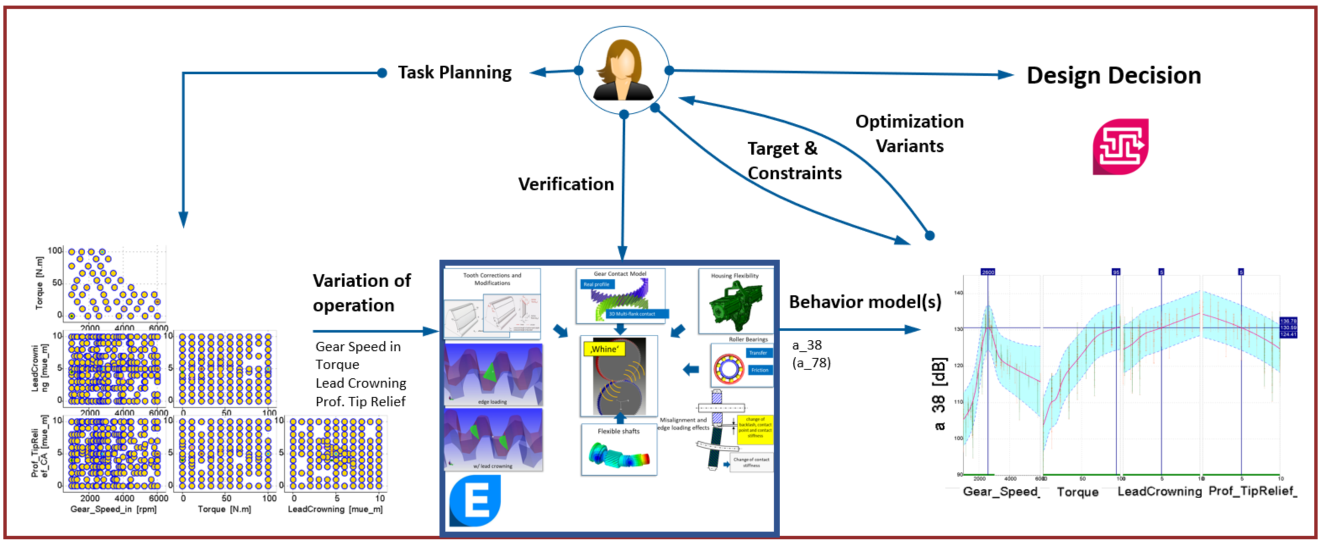


Figure 4: Combination of the AVL Tools CAMEO™ and EXCITE™ to support the Design Space Exploration and finally the Design Decision

One can imagine, that the data load to be analyzed would become enormous, when

1. *having a look towards the whole Speed / Torque range*

   and

2. *starting to vary design parameters in order to decide on a proper definition of those.*

Consequently, a first data reduction to relevant KPIs is performed for each Speed / Torque combination and each design variant by taking the 38th and 76th orders of the acceleration signals.

Therefore, the "Design Space Exploration" is supported as shown in Figure 4.

By defining a design of experiments in the optimization tool (AVL CAMEO™) for the input parameters needed by the MBS-Tool (AVL EXCITE™) one can get a systematic overview, of the trends of the relevant responses (e.g.: the "a_38") behave as function of Speed / Torque, Lead Crowning and Profile Tip Relief. This could be reached using an S-optimal design of Experiments with 360 simulation runs [3]. The models gained for a_38 and a_76 are called "first models" below in the text.

One has to start with the first step, which is the task planning, which will lead one step beyond such "first behavior modelling" in order to reach a robust design decision:

3 Extended KPI calculations to address the challenge of robust optimization

In our case one could summarize this robust optimization task as follows:

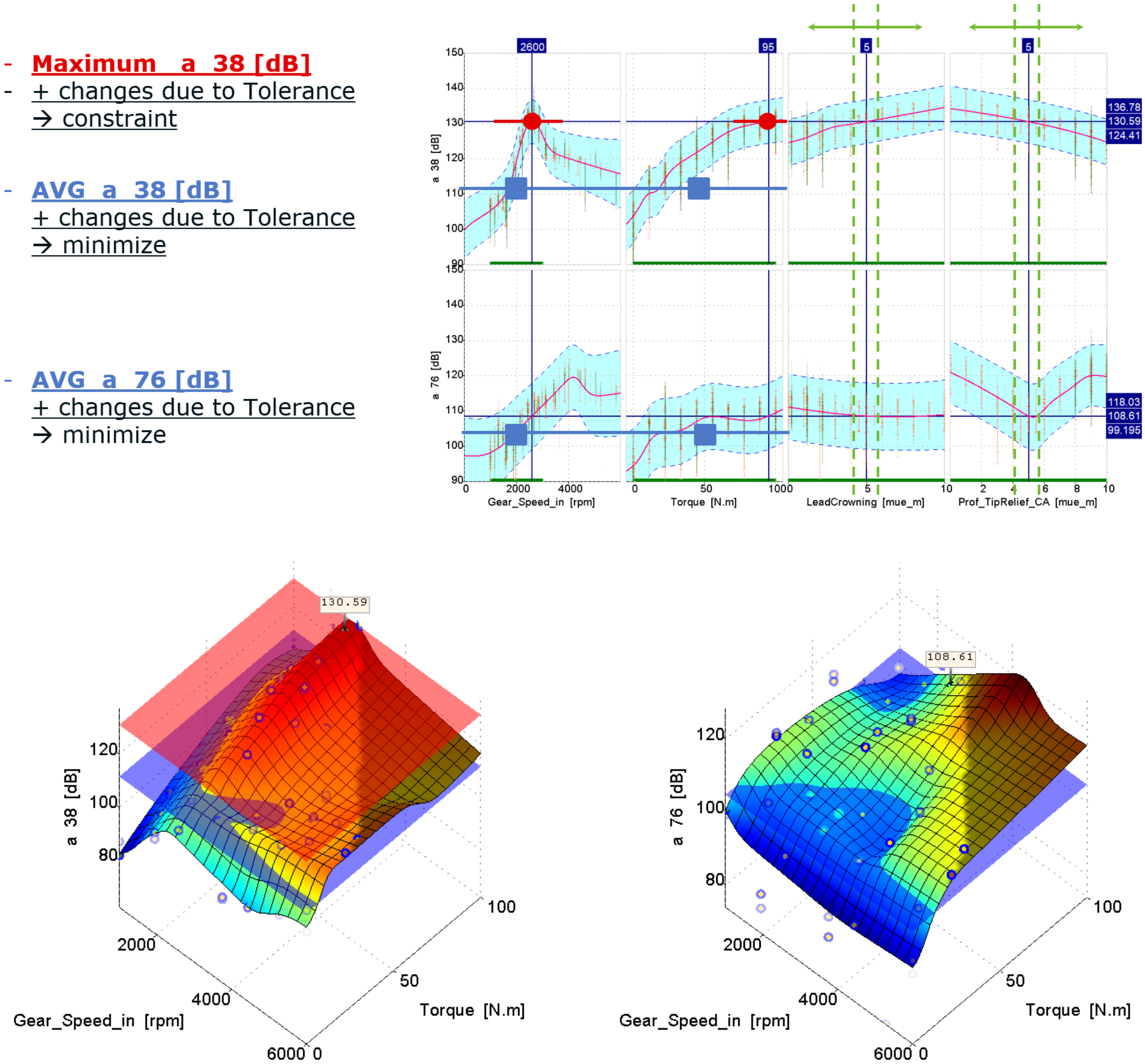


Figure 5: 38th and 76th order of the relevant surface acceleration as a function of the e-drive operation condition (input speed and torque)

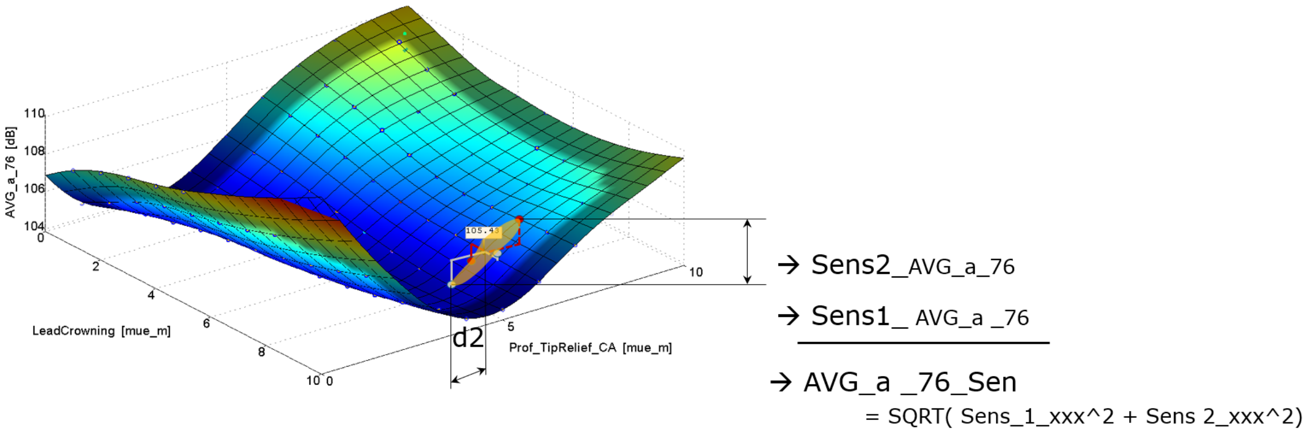


Figure 6: "AVG_a_76" as influenced by Lead Crowning and Profile Tip Relief and its tolerances, giving the sensitivity KPI "AVG_a_76_Sen"

1. ***Target:*** *Minimize air borne noise, while maximizing the tolerances for the design parameters, which should not lead to strong changes in the noise behavior over the Speed / Torque range.*
2. ***Variation parameters:***
   a. *Lead Crowning [0.0 to 10.0] micro m,*
   b. *Tolerance for Lead Crowning [0.1 to 1.5] micro m*
   c. *Prof_Tip Relief [0.0 to 10.0] micro m,*
   d. *Tolerance for Prof_Tip Relief [0.1 to 1.5] micro m*
3. ***Observe:*** *KPIs over the whole Speed / Torque range in order to judge "air borne noise".*

In order to fulfil this optimization task, there is still the open point of how to characterize the relevant KPIs for the "air borne noise" over the whole Speed / Torque range?

We have already decided to use the a_38 and a_76 as KPI for each Speed / Torque point to judge the air borne noise. However according to the task definition, we should not look into selected Speed / Torque points but somehow consider all Speed / Torque combinations. Further we should ensure that the changes for the finally selected design are not too large, when it comes to production with its immanent tolerances.

So, to define this next KPI level, Figure 5 shows the ratings as maximum and average of noise level over the speed load ranges: In the upper part of the Figure, the dependency of "a_38" and "a_76" as a function of all the 4 varied input parameters is shown. In the lower part, one can see the model cuts as 3D plots showing the function of Speed and Torque

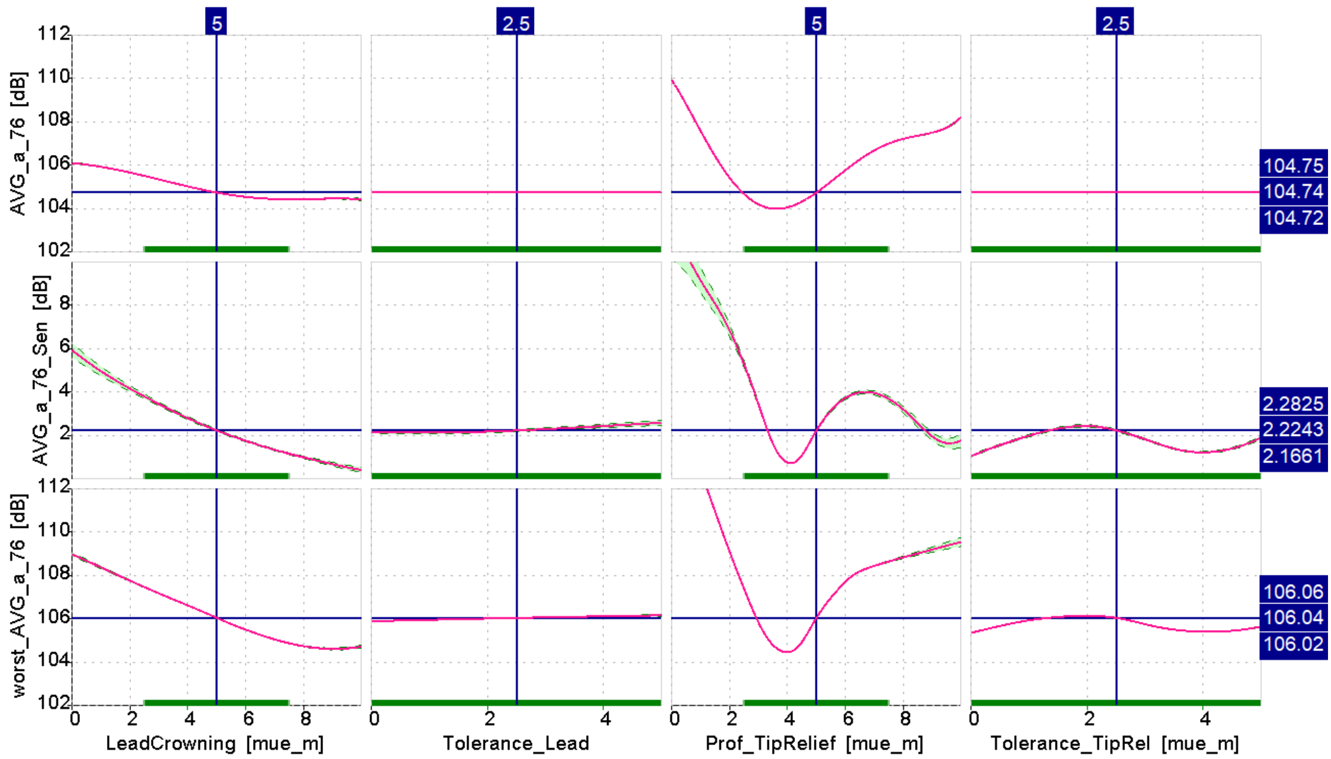


Figure 7: "AVG_a_76" and "worst_AVG_a_76" taking the tolerances into consideration

only. For this model cuts the Lead Crowning and the Profile Tip Relief is fixed to 5 micro meters each.

This is the kind of KPI needed, as these values are independent from Speed and Torque, when changing the Lead Crowning or the Profile Tip Relief. Also the micro teeth geometry will not - and cannot - be changed as a function of Speed / Torque.

As an optimization target it should be achieved the lowest possible value for both excitation orders on average and for the maximum values. However, this result should also be robust against manufacturing tolerances of the micro teeth geometry. So, the corresponding tolerances should be maximized at the same time.

In order to calculate "AVG_a_38", "Maximum_a_38" and "AVG_a_76", the "first models" from Figure 4 were resampled in AVL CAMEO™ with a resolution of 500 1/min in Speed, 10 Nm in Torque and 1 micrometer in Lead Crowning and Profile Tip Relief. As the resampling is a fast process, this was done in full factorial mode. The number of resulting points was 14641 - just for comparison to the 360 simulation runs as calculated by the in the S-Optimal Design requested to be simulated by the MBS tool. These results were grouped according to the Lead Crowning and Profile Tip Relief.

In each group the Speed / Torque range is far away from being a "full factorial square" (compare Figure 4 on the left side). Consequently, searching for maxima also in the extrapolated range would completely mislead the KPI target to be relevant. So, all extrapolated estimations for "a_38" and "a_76" were removed using the design space information available in the optimization tool. For the remaining points (11139) in each group, the desired average and maximum values were calculated. Figure 6 shows for example "AVG_a_76" as a function of Lead Crowning and Profile Tip Relief.

Using this model, one can allow two additional degrees of freedom - as requested in the task definition: The variation of the tolerances for the design parameters.

A simple robustness KPI "AVG_a_76_Sen" was calculated in the following steps:

The first sensitivity value was calculated out of the "AVG_a_76" deviation depending on how far the plus/minus tolerance deviates from the target value in each direction of two design parameters (e.g.: "Sens2_AVG_a_76" for the tolerance in "Prof Tip Relief" direction). As sketched further in Figure 6, in a second step an "AVG_a_76_Sen" is finally calculated as the Pythagorean sum of both part sensitivities.

This sensitivity KPI is further used to estimate the "worst_AVG_a_76" as visualized in Figure 7.

Figure 7 shows in the middle intersection model the dependency of the "AVG_a_76_Sen" on all the 4 parameters. As one can see in the first row, the model for the "AVG_a_76" is a function of Lead Crowning and Prof Tip Relieve only, and independent from the tolerances. So finally adding to that signal 50% of the sensitivity KPI ("AVG_a_76_Sen") gives the worst case acceleration in the 76th order on average over the whole Speed / Torque range, when applying the corresponding tolerance ("worst_AVG_a_76", lower row).

Consequently, the following KPIs will be used for the robust optimization in order to judge the degree of target fulfillment as defined in the task description.

"Minimize air borne noise, while maximizing the tolerances for the Variation parameters, which should not lead to strong changes in the noise behavior":

- *worst_AVG_a_38 [dB]* → *minimize*
- *worst_AVG_a_76 [dB]* → *minimize*
- *worst_Max_a_38 [dB]* → *constraint below a low value*

4 Optimization

For the optimization itself, a multi objective optimization is used. In order to stay within 2 target functions and also to maximize the costly tolerances, there was further defined a

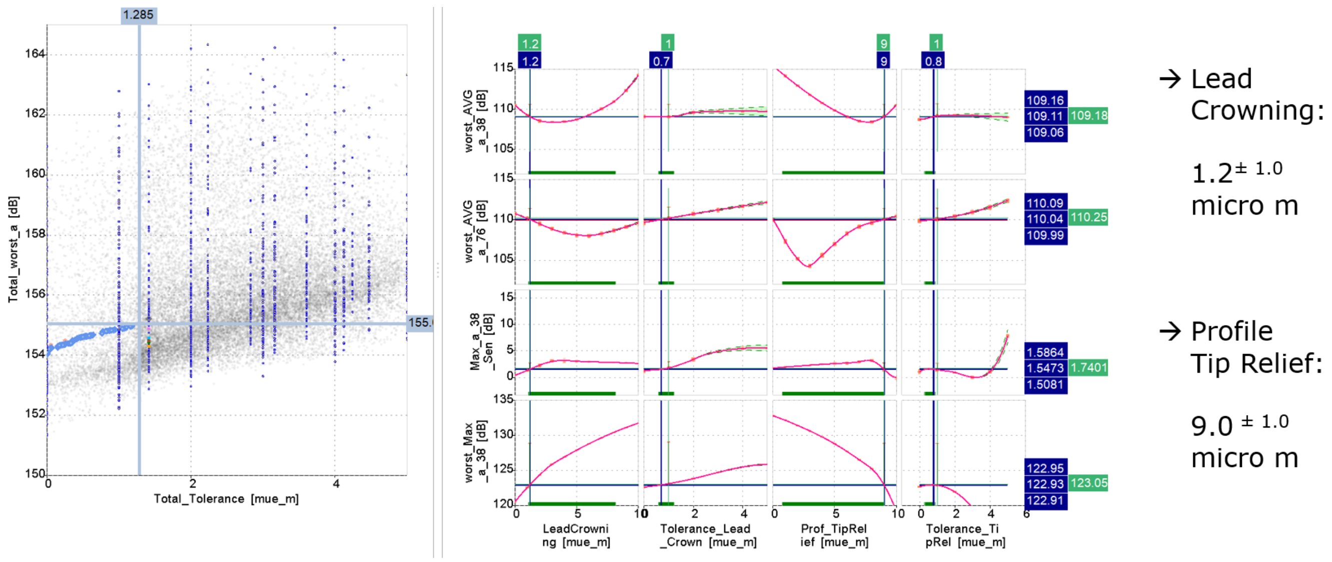


Figure 8: Minimizing total worst acceleration and maximizing tolerances

- "Total_worst_a" = Sqrt (worst_AVG_a_38 ^{2} + worst_AVG_a_76 ^{2})

  and

- "Total_Tolerance" = Sqrt (Tolerance_Lead_Crowning ^{2} + Tolerance_Tip Relief ^{2})

While the "Total_worst_a" shall be minimized the "Total Tolerance" shall be maximized and the "worst_Max_a_38" shall be kept constrained below 123 dB.

Also "worst_AVG_a_38" and "worst_AVG_a_76" shall both be kept constrained below 110 dB.

Figure 8 shows the corresponding trade-off and the intersection plots at the finally selected high tolerance and thus the robust design selection:

The trade-off position with the highest tolerances at acceptable low "Total_worst_a", came out as

→ Lead Crowning: 1.2 $^{\pm 0.7}$ micro_m

and

→ Profile Tip Relief: 9.0 $^{\pm 0.8}$ micro_m

A manual extension towards an increase of the tolerances to +/- 1 micro meter for both, showed also acceptable results (in green), so with these values the final verification runs in MBS tool are triggered.

5 Verification

In order to judge the final selection and to get the data for calculating the corresponding KPIs, 250 simulation runs were needed as shown in Figure 9.

The reason for this relatively high number of points is given by the 50 operating points used for the "Speed / Torque" range also for the S-optimal start design of 360 points, which all the model based evaluation and optimization is based on.

One can see in the upper right projection of the original design towards the Lead Crowning / Profile Tip Relief area, that the suggested position has moved from the center towards the green points checking the sensitivity towards production tolerances (compare Figure 6)

→ Lead Crowning: 1.2 $^{\pm 1.0}$ micro m

and

→ Profile Tip Relief: 9.0 $^{\pm 1.0}$ micro m

In the lower right corner one can see, that all the 250 verification results for the "first level KPIs like "a_38" and "a_76" are within the predicted tolerances. This can be seen as the green simulation results are inside the blue shaded area. The magenta line in the middle represents the main trend models for "a_38" and "a_76" respectively.

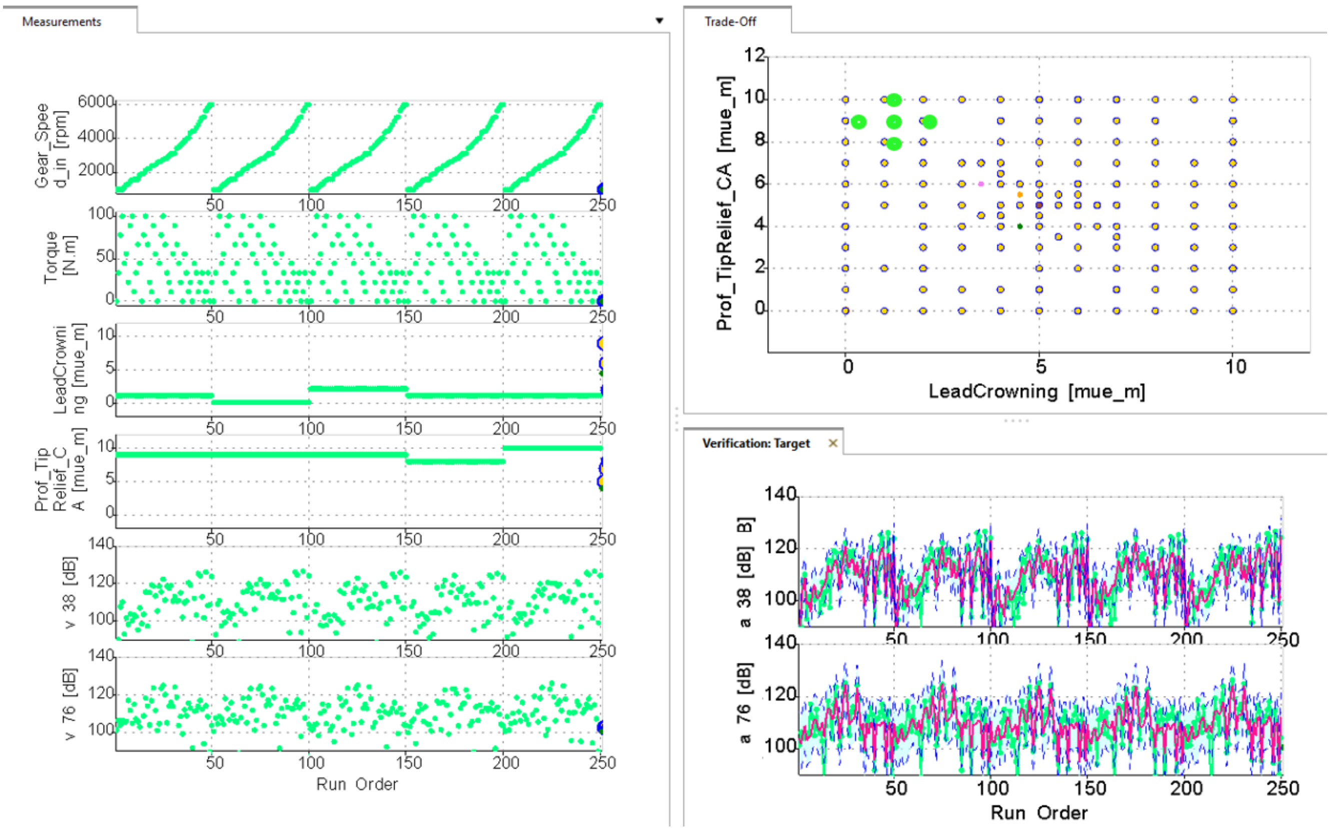


Figure 9: Verification runs prove the base model validity within its prediction intervals

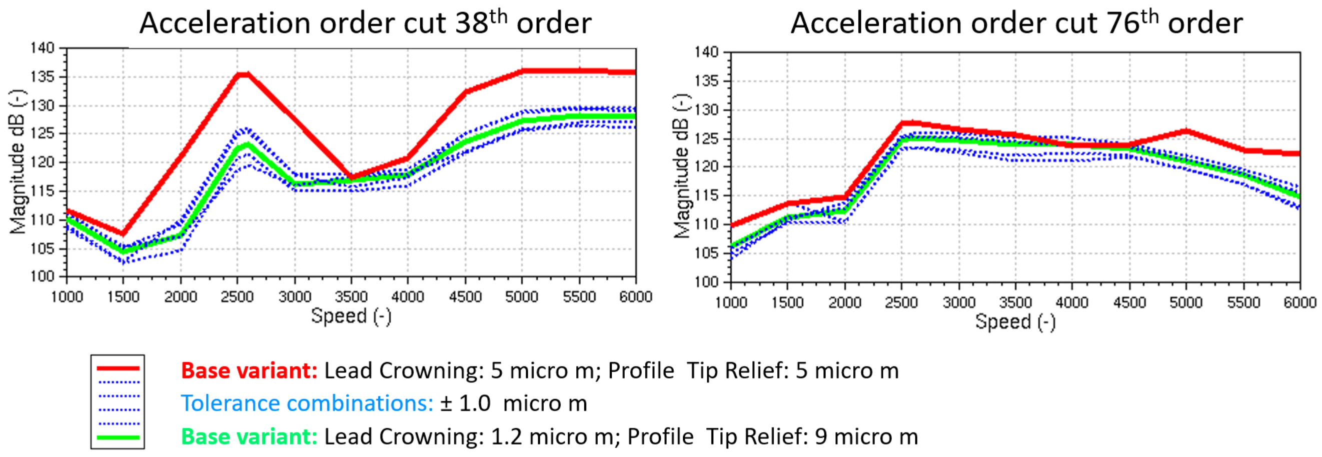


Figure 10: Comparison of the initial acceleration along the order cuts for a torque of 95 Nm and the final optimized result including the effects of worst-case tolerances.

6 Results

Comparing now the „first KPIs" – the acceleration amplitudes in the 38th and 76th orders, one can see the strong improvement and harmonization reached in Figure 10.

While the "a_76" was partly compromised a clear improvement was reached in the 38th order ("a_38"), which was dominating before. Further the target of low influence of the production tolerances towards the NVH behavior can be seen.

When suggesting this result to the manufacturing division, there was a request to further increase the tolerances, especially the one for Lead Crowning. As shown in Figure 11, the consequences to be expected can clearly be shown: Mainly the "Max_a_38_Sen" increased from ca. 1.5 to ca. 5.2 dB and consequently led to an increase in the "worst_Max_a_38".

Overall this tool can now form the base for a robust design decision, estimating the consequences and enabling an open discussion with different stakeholders in a company.

7 Summary

By combining high fidelity physical models (MBS-Simulation) with empirical trend models resulting from Design Space Exploration and robust optimization, the following targets could be achieved:

- *The challenge of understanding the main effects of micro teeth geometry parameters within the whole Speed / Torque range could be achieved with a relatively low number of simulation runs: 360*
- *With simple KPI calculations the effects on production tolerances could be judged.*
- *A robust design decision considering trade-offs with the highest allowable tolerances supports sustainable, cost effective production to provide the best combination of attributes for e-drive customers.*

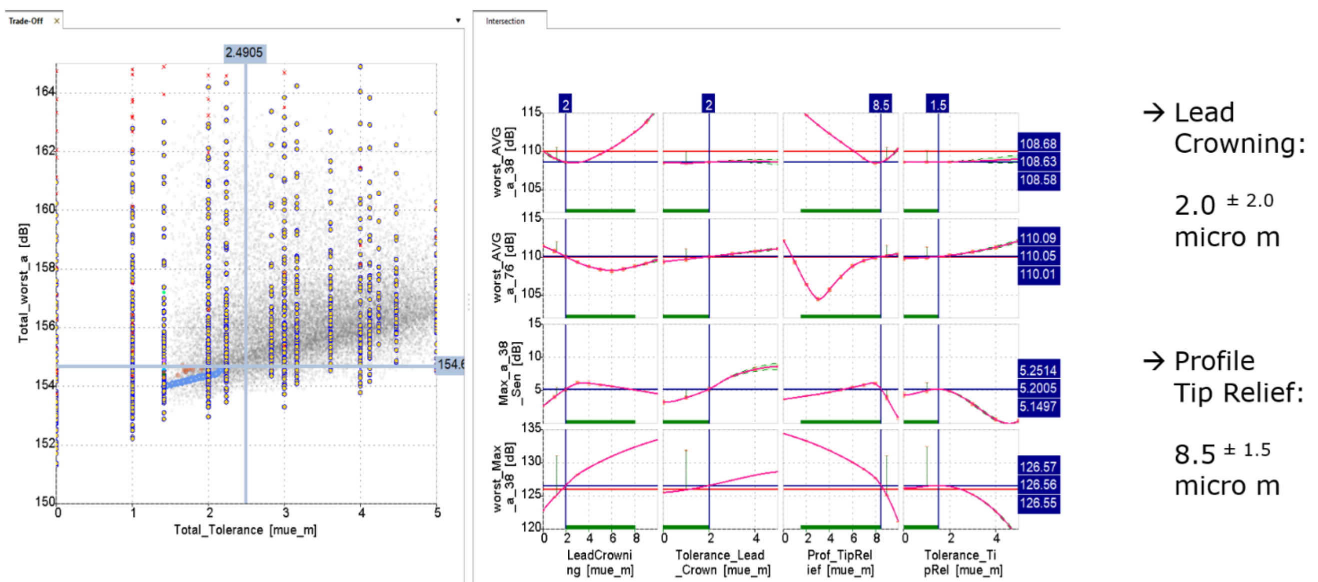
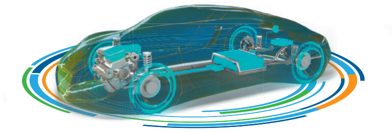


Figure 11: Trade-off and intersection models as a start point for discussion



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