Electric Vehicle Whine Noise—Gear Blank Tuning as an Optimization Option

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Introduction

Noise issues from gear and motor excitation whine are commonly faced by many within the EV and HEV industry. In this paper we present an advanced CAE methodology for troubleshooting and optimizing such NVH phenomenon. Experience has shown that to achieve good NVH behavior in such a challenging environment requires a combination of optimization options. A traditional focus on gear micro geometry to minimize TE alone is often seen as not sufficient for achieving the desired targets. Rather all aspects of the noise phenomenon including, amongst others, gear macro geometry, micro geometry, system dynamics, transfer paths, damping and acoustic isolation should be considered. Given the multiple optimization options, multiple objectives and often stringent timescales, a high-fidelity analysis model can be critical in assessing the effects of design changes in the allowable timeframe while minimizing hardware and testing loops. Here we focus on an example using gear blank dynamic tuning as an optimisation option for gear whine dynamic performance.

As a transmission engineering services provider, we are seeing increased and sustained interest in the analysis and optimization of transmissions for whine noise issues. This is particularly the case with the EV and HEV market where there is no, or less, IC engine noise to mask any other noise sources. Further, ever-increasing standards and expectations are pushing noise targets to even more stringent levels. In this dynamic and still-developing market, jumps to new configurations with less incremental development and an abundance of new market entrants with less long term automotive experience are further contributing to these issues. This highly competitive market atmosphere further drives development to tighter and tighter timescale targets. As a result, there is an ever-increasing need for advanced CAE tools that can help design and optimize for noise performance throughout the development cycle, and significantly reduce hardware and testing loops.

In this paper we present an advanced CAE process—available in SMT's *MASTA* software—for the design and analysis of transmission systems, with a specific focus on the EV automotive market. We introduce a specialist full transmission system analysis model that considers all the multiple performance targets of interest within the design process. Static deflections of the system, durability, efficiency and frequency and time domain dynamics can all be considered. The focus of this paper is gear whine simulation in the frequency domain.

Following an introduction to the analysis model a design optimization option for NVH is discussed. Gear blank dynamic tuning involves changing the dynamics of a gear blank via geometry changes to rim and web to reduce the dynamic mesh force at the gear mesh and subsequently reduce the vibration/noise response of the system. The fundamentals of gear blank tuning are introduced; a novel, automated process implemented within the CAE tool is then discussed. An example is demonstrated of the dynamic tuning of the gear blank of one stage within a typical two-stage helical automotive EV transmission.

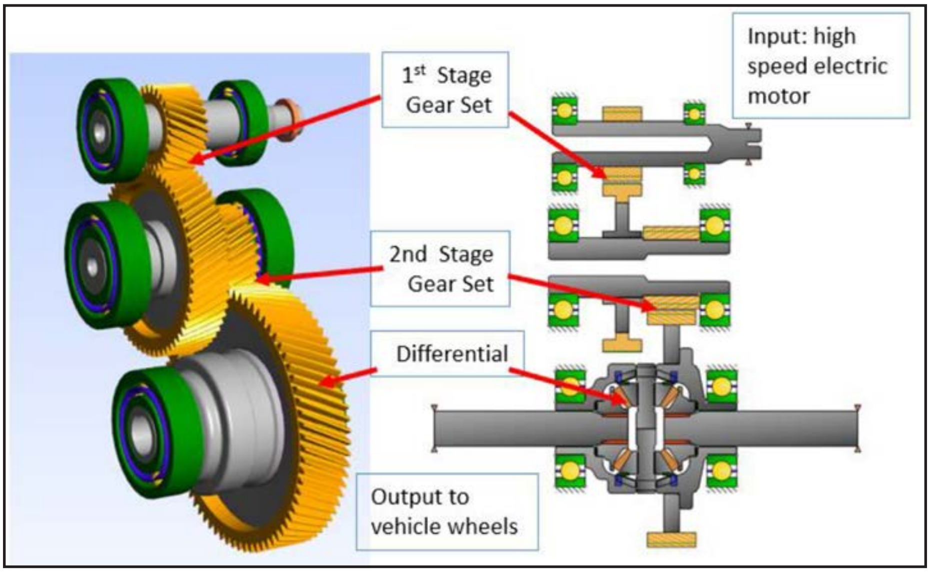

Figure 1 Typical two-stage helical E-Drive transmission.

Whine Noise in Electric Vehicles

EVs have a range of different noise sources. Excitation from transmission error at the gear meshes, torque ripple, and radial out of balance at the rotor, dynamic tangential and radial forces at the stator teeth of the motor, and switching frequencies of the controller can all lead to irritating tonal noises within the vehicle as the frequency of excitation crosses certain resonances of the mechanical system.

One classic E-Drive architecture with some mature and competing products within the market consists of a permanent magnet motor connected to a two-stage helical speed-reducing transmission to provide the required torque to the wheels (Fig. 1). Some mature products with this architecture and their maximum quoted operating speeds include the Nissan LEAF (10 K rpm), Borg Warner eGear (14 K rpm), and the GKN eTransmission (15 K rpm). Although many other architectures are available in the marketplace—and are currently being developed—the simplicity and low cost of this single-speed solution is still very attractive.

Motor speeds within this group range up to 15,000 rpm. Speeds are being pushed higher and higher in current and planned future developments driven by the benefits for the motor. High-speed motors can be designed smaller for the same power output with higher power-to-weight ratios. For permanent magnet (PM) machines, smaller, higher-speed motors require less magnets—thus leading to significant cost reductions.

The evolution towards higher-speed motors has led to higher and more challenging demands on the transmissions required to reduce the speed and increase the torque to the wheels. The design challenges, including those for NVH performance, require significant engineering tools and ingenuity to be overcome.

Figure 2 shows typical noise and vibration responses for a mature EV.

The vibration results show clear gear orders and their harmonics, motor orders, and higher-frequency "fans" due to controller PWM switching frequencies. A potential resonance is seen at around 4,500 Hz. Within the driver's ear noise measurements only the gear orders are visible, but are also mostly masked by other noise sources, such as road noise.

Figure 3 shows order cuts through the gear and motor orders of the noise results in Figure 2.

A small region of high 1st stage gear order can be seen. An interesting question that arises is how to quantify gear noise objective targets; here the difference between total noise and gear order noise is greater than 20 dBA. The maximum absolute gear order noise is approximately 37 dBA. By studying the tone-to- noise ratio of the peaks on the order lines, one can see if they show up as prominent tones that will be heard above external noise. In this case all the peaks—including the one circled in red in Figure 3—pass the criteria set out in ECMA-74 and so won't be heard as distinct tones during a speed sweep of the motor. This result may be expected for a mature product such as this.

Advanced CAE Methodology for Whine Simulation

To design for good NVH performance, or to solve NVH problems, a combination of experience and the right tools are required to find solutions with minimal cost and timescale. CAE tools have been typically used within this process for system model static analysis and prediction of gear misalignments, gear macro and micro geometry design for low transmission error (TE), and transmission mount tuning to limit structure-borne noise.

With the increasing complexity and higher demands of the modern EV market, more attention also needs to be paid to system dynamics. A high-fidelity model can be used to investigate in detail

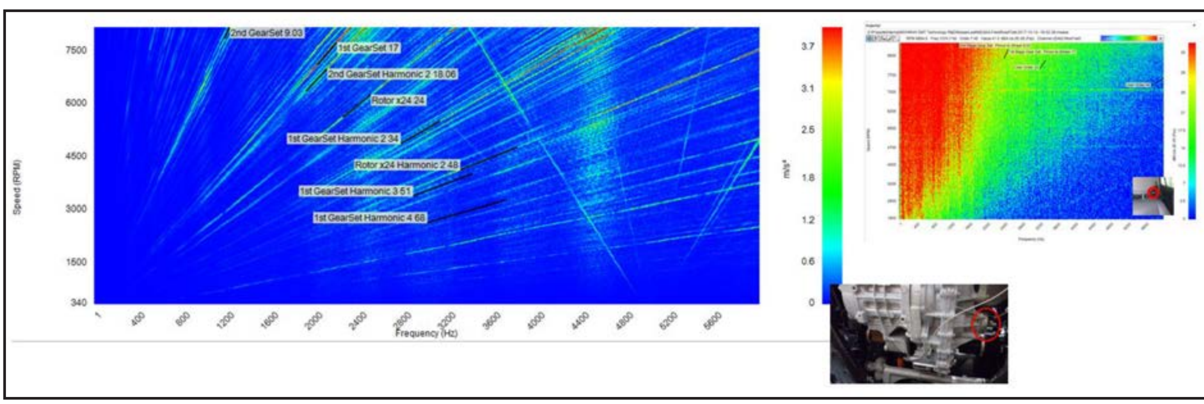

Figure 2 Typical vibration responses and noise at driver's ear for a mature EV.

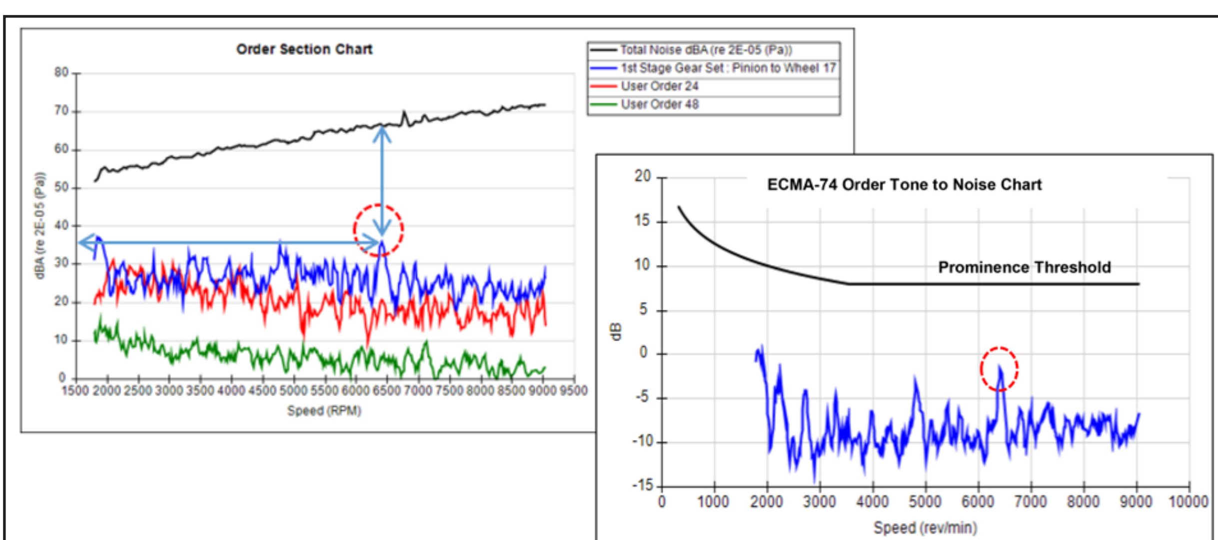

Figure 3 Order cuts of noise response at driver's ear for a mature EV, and associated tone to noise chart for the 1st stage gear order.

the full dynamic system and to tune for low dynamic transmission of vibration and noise sources.

The model used in this study consists of a fully integrated, mechanical FE-based model of the motor and transmission system (Ref. 1) (Fig. 4). Within the analysis model, shafts are included as Timoshenko beams or full solid FE representations. Gear blanks, gearbox casing and motor casing — including the motor stator — are included as full-solid FE models. Bearings and gear mesh contacts are included as bespoke nonlinear stiffness models. The model can be used to run static, frequency domain dynamic, and time domain dynamic analyses.

Here we focus on the frequency domain harmonic response of the EV gearbox model to excitation by gear TE, motor torque ripple, and stator tooth harmonic forces.

For excitation of the model by TE in the frequency domain a standard assumption is made that the excitation is the static transmission error. The static transmission error is calculated via a hybrid Hertzian- and FE-based tooth contact analysis model (Ref. 2), and introduced to the dynamic system level model as an enforced relative displacement in the line of action at the gear mesh. Taking care to recall that transmission error is defined in the transverse line of action, whereas the relative displacement is to be applied normal to the flank, i.e. — normal to the helix. The static TE is enforced into the model using a method well documented by Steyer et al (Refs. 3–4). First the dynamic compliances are calculated on either side of the gear mesh. The compliance at one side is calculated by applying a unit harmonic force in the line of action at that side and calculating the resulting harmonic displacement at the mesh in the line of action. The total compliance is then calculated as the sum of pinion and wheel side compliances. The dynamic mesh stiffness is calculated as the inverse of the mesh compliance. The dynamic mesh force for a given harmonic of TE is then calculated as the product of the TE and the dynamic stiffness.

$$C_{mesh}(\omega) = C_p(\omega) + C(\omega)$$
$$D(\omega) = (C_{mesh}(\omega))^{-1}$$
$$Fi(\omega) = D(\omega)\,\delta_i$$

Where:

$C_{p,w}(\omega)$ — Dynamic compliance in the line of action at the mesh, at the pinion (p) and wheel (w) sides, at frequency ω

$C_{mesh}(\omega)$ — Total compliance at the mesh in the line of action

$D(\omega)$ — Dynamic mesh stiffness in the line of action

$F_i(\omega)$ — Dynamic mesh force for the ith harmonic of the TE

δ_i — The ith harmonic of the TE — transformed normal to the flank, normal to the helix

At a specified frequency the dynamic mesh force can be considered to be the harmonic force required to be applied both equally and opposite at the gear mesh in the line of action, such that the resulting relative displacement in the line of action at the mesh is given by the static transmission error. The dynamic mesh force is calculated and applied to the FE model for a sweep of input speeds to give the dynamic response of the whole system.

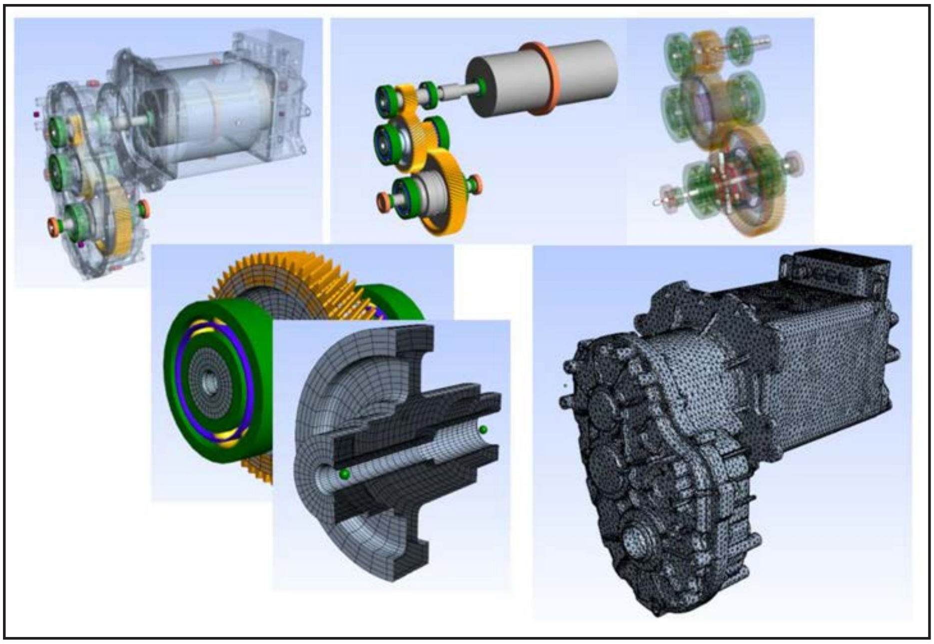

Figure 4 FE-based bespoke EV system analysis model.

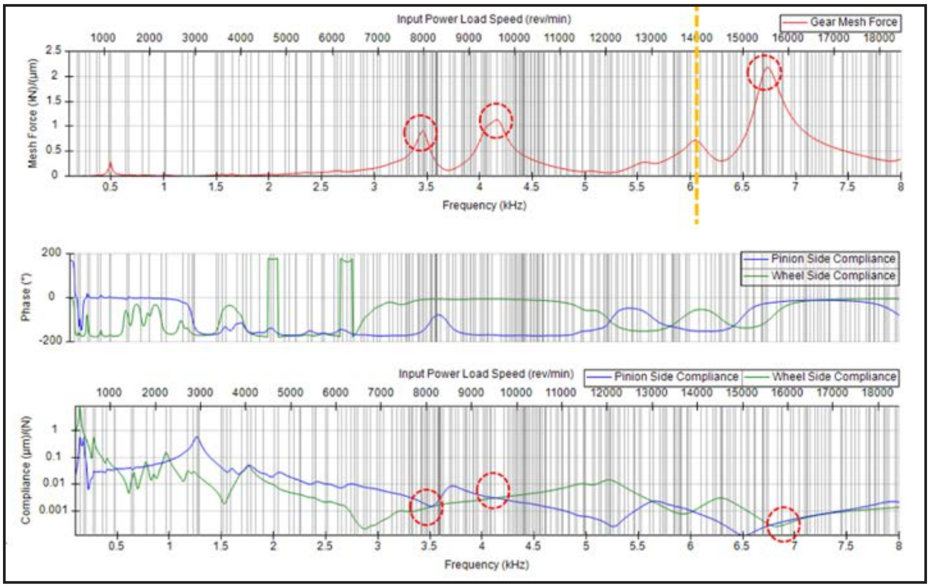

Figure 5 Dynamic mesh force peaks where compliances are equal in magnitude and opposite in phase. The dotted line is the maximum operating speed of this transmission of 14,000 rpm.

As this is a frequency domain analysis it is fast and lends itself well to design optimization. The understanding of the dynamic compliances at the gear mesh and their relation to the dynamic mesh force enables a mechanism for tuning the dynamics to decrease the dynamic mesh force. Gear mesh force peaks occur when the compliances on either side of the mesh are equal in magnitude but opposite in phase (Fig. 5).

Understanding how the compliances are made up helps in optimizing the dynamics. The high-fidelity model had many degrees of freedom and therefore many modes. Only a small number of key modes with respect to the pinion and wheel sides control the compliances. The gear blank modes in particular in this transmission architecture are seen to be very involved in the wheel side compliance (Fig. 6).

The gear blank modes can be tuned to try to avoid high gear mesh forces within the operating range. It is important, however, to also check transfer paths as well, using predicted bearing and housing responses. A change in gear blank dynamics influences both the dynamic mesh force and the transmission of that force through the system.

The excitation from the electric motor is applied to the model in a simpler way than the transmission error. A third-party electric motor analysis tool is used to calculate the dynamic forces at the rotor and stator teeth at several speed operating points. These forces are imported into the mechanical motor and transmission model. For a given speed the imported forces are interpolated and applied as harmonic forces directly to the model. The response to the forces from the motor and the TE are calculated within the same analysis (Fig. 7).

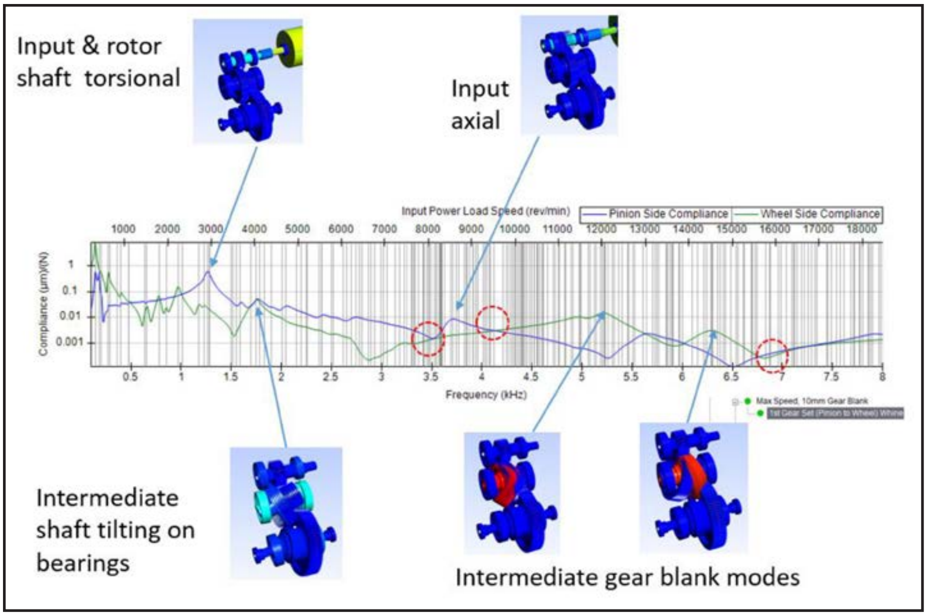

Figure 6 Gear blank mode influence on wheel-side compliance.

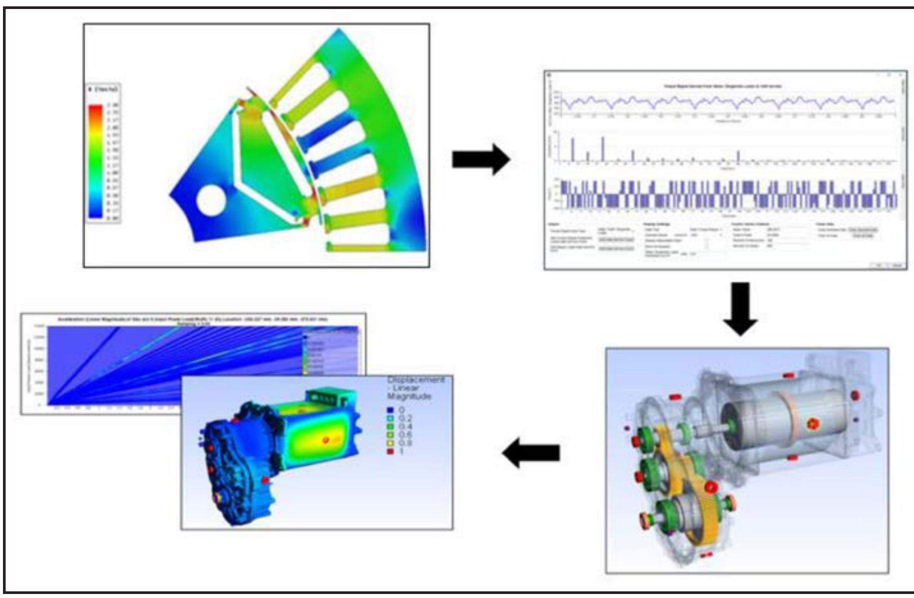

Figure 7 Electric motor forces calculated from a motor analysis tool and imported to the mechanical motor and transmission model.

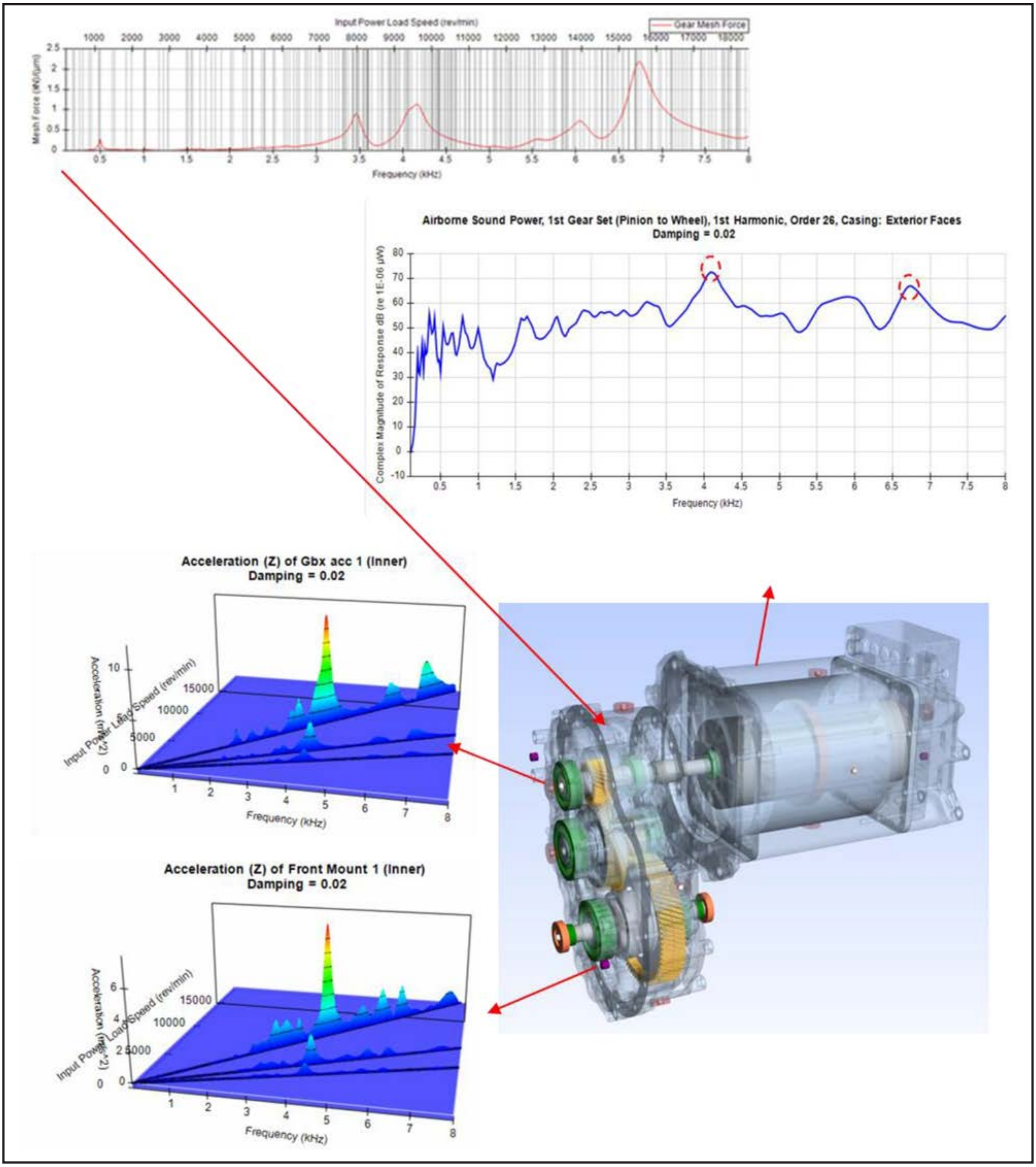

Figure 8 System response to TE and electric motor excitation.

The system response to the transmission error at the gears and harmonic excitations from the motor can be checked at different locations in the model (Fig. 8). Typical metrics for assessing system response include the bearing dynamic response, casing accelerations (which are often compared against accelerometer tests), mount dynamic responses (which give an indication of structure borne vibration), and housing response as sound power via ISO 7849.

Velocity response of the housing can be further exported to acoustic simulation packages for full radiated noise predictions (Fig. 9).

Radiated noise prediction is a slow process and does not fit into typical optimization loops; however, it can provide a high-fidelity check of final design changes.

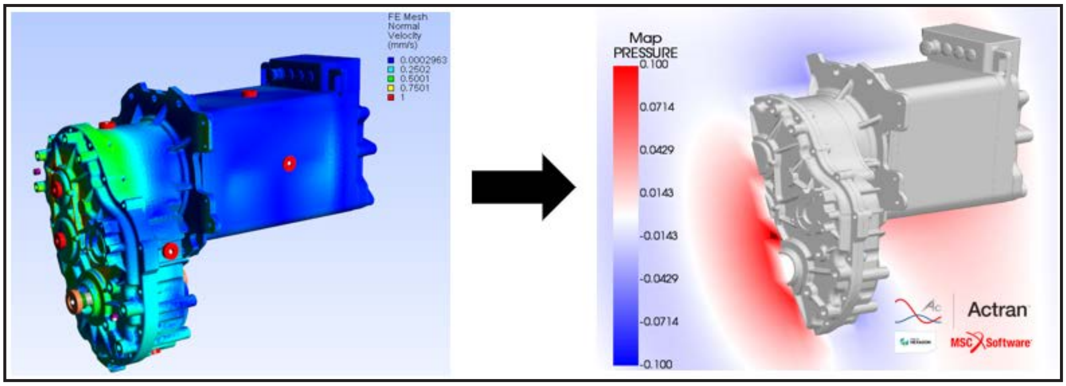

Figure 9 Export of housing velocities and subsequent radiated noise predictions in Actran.

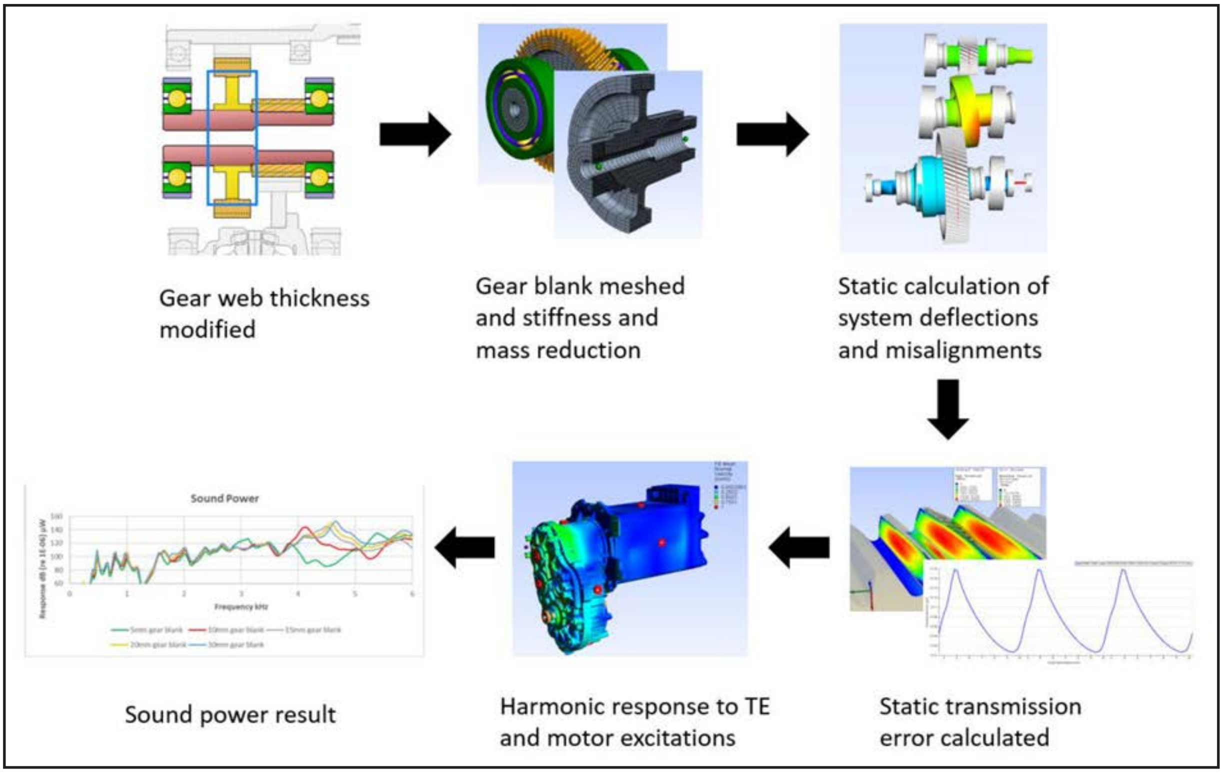

Figure 10 Automated optimization methodology used to investigate the effect of gear blank web stiffness on system dynamics.

Optimization via Automated Gear Blank Dynamic Tuning

The implemented simulation tool and analysis model includes general integrated design of experiments and external batch running capabilities. A huge number of model and analysis parameters are available to these functions, providing a broad range of possibilities for automated design optimization.

In this study such capabilities were utilized for automated optimization of gear blank geometry to minimize dynamic mesh force and system response. Thinner gear blanks are lighter, but may give higher gear mesh misalignments — especially at high loads. Thinner blanks may be beneficial dynamically as they add compliance to the system and can therefore reduce the dynamic mesh force. However, changing the blank dynamics can also change the transmissibility of that dynamic mesh force from gear mesh to housing.

In order to find an optimum gear blank design, a parametric study was set up to modify the web thickness of the blank between 5mm and 30mm, and the rim thickness between 3mm and 15mm. In this process the gear web is modified automatically by the software. The gear blank is then automatically meshed and a stiffness and mass matrix dynamic reduction automatically run in order to capture the dynamics of the gear blank in a reduced model. Static deflection analysis of the system is run to calculate the system stiffness under the specified operating load. The transmission error is then calculated. Frequency response analysis to excitation by transmission error and the electric motor excitations is automatically run and the dynamics results are recorded; Figure 10 shows a workflow of this automated process.

The outcome of this methodology was an optimized gear blank with a web thickness of 11mm and a rim thickness of 3.5mm, chosen for giving the lowest combination of peak results in gear dynamic mesh forces and system dynamic response over the frequency range of interest, while complying with the minimum rim thickness required for the gears' tooth height according to ISO 6336-3.

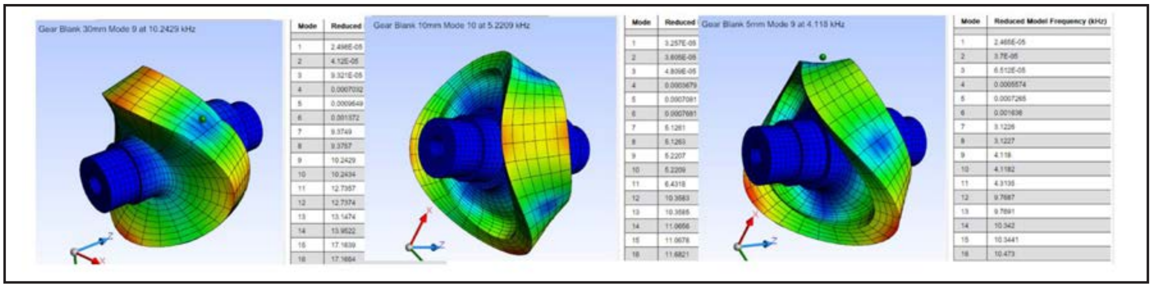

Figure 11 Free-free natural frequencies and first gear blank "potato chip" mode shape. For models with a rim thickness of 12.25 mm and web thickness of 30, 10 and 5 mm.

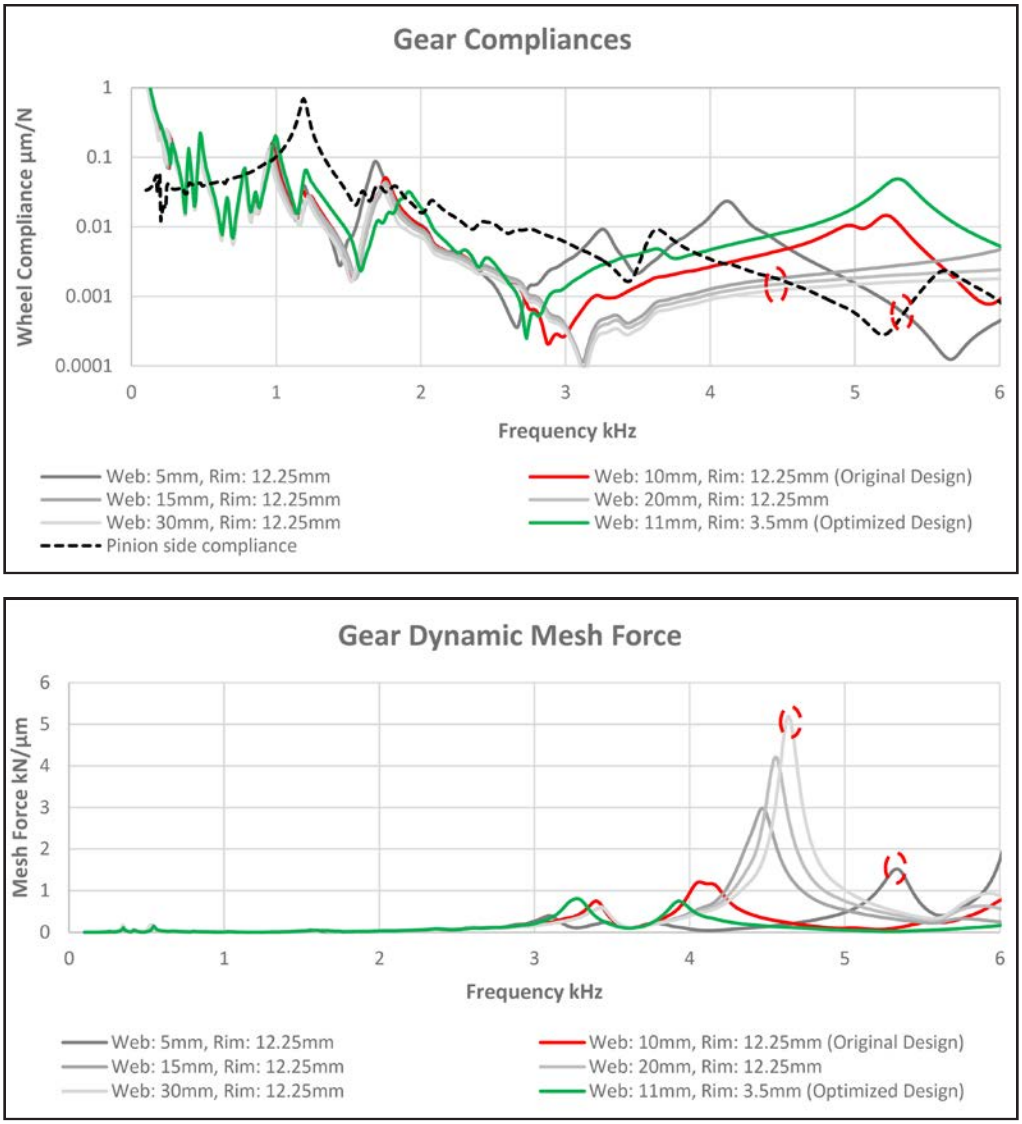

Figure 12 Dynamic compliances and mesh force for different gear blank dimensions.

Figure 13 Gear mesh misalignments for different gear blank dimensions at maximum torque.

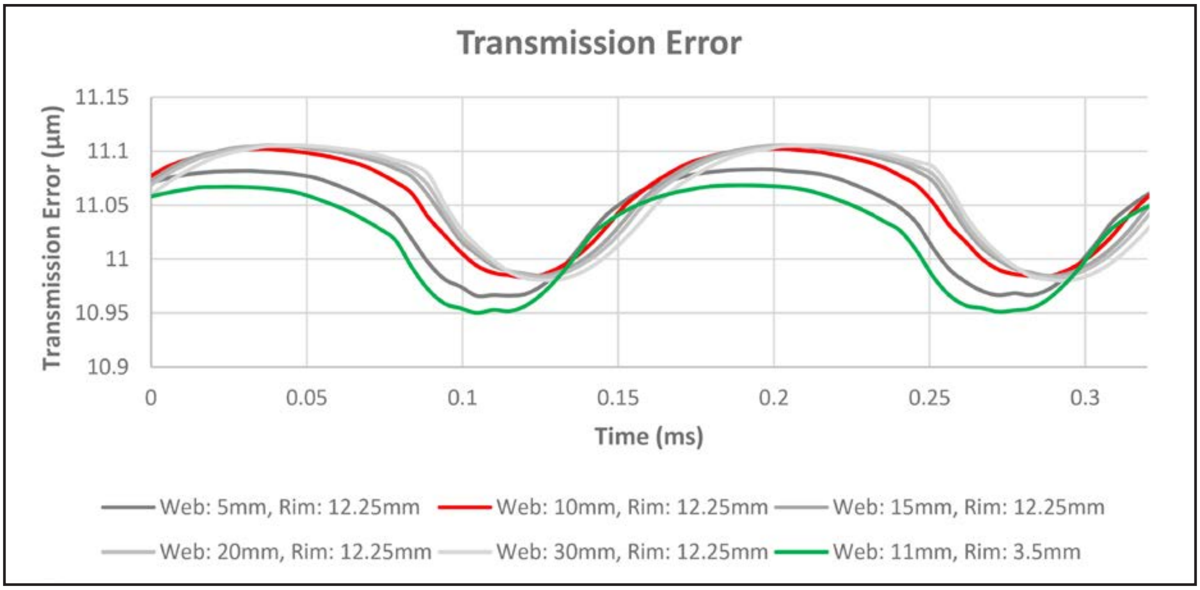

Figure 14 Transmission error for different gear blank dimensions.

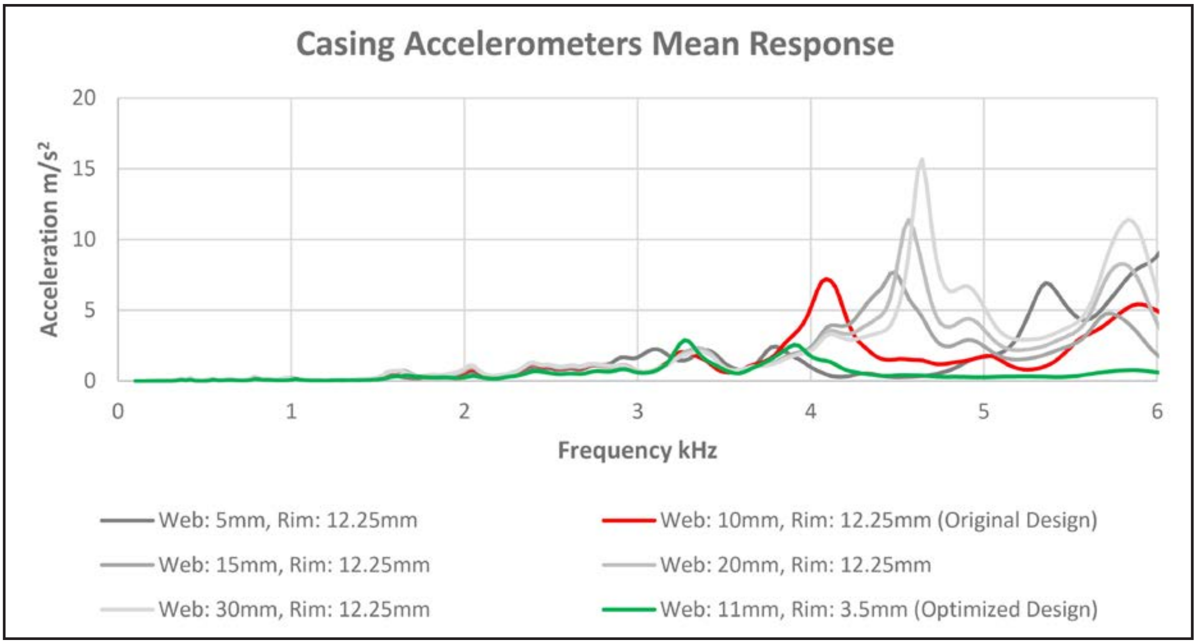

Figure 15 Casing accelerometer mean response for different gear blank dimensions.

In order to give some indication of the blank dynamics it is instructive to look at the free-free modes of the gear blanks first in isolation. Figure 11 shows the natural frequencies and the mode shapes of the first gear blank "potato chip" mode.

This first mode is important for the dynamics and transmissibility of the TE excitation. For a 30 mm blank the first mode is at 10.24 kHz; for 15 mm it is 7.5 kHz, and for 10 mm it is 5.6 kHz. For the 5mm gear blank there are 5 modes from 3 to 5 kHz.

Figure 12 shows that the change in gear blank dynamics has a very significant effect on the dynamic compliances of the system and the derived dynamic mesh forces.

The thicker and stiffer blanks have potential resonances at higher frequencies, outside the operating range for web thicknesses greater than 15 mm. However, thicker blanks have less compliance, and this gives higher gear dynamic mesh forces within the operating range. The optimum solution provides a good balance of these effects, with an associated decrease in mesh force due to a high compliance from a low rim thickness.

In Figure 13, very thin- or thick-webbed blanks can be seen to have higher misalignments, while in Figure 14 thinner blanks can be seen to have slightly lower transmission errors — but the same peak-to-peak TE values. The optimum solution had the lowest transmission error, but also had a higher misalignment than the wider-rimmed 5 to 30 mm web thickness blanks — one minor drawback to the optimization process. It should be noted here that a single micro geometry design was used for all gear geometries so that the effect of blank changes on the TE could be seen, although in practice the micro geometry would be optimized for every gear geometry.

Figure 15 shows the dynamic response in terms of casing acceleration based on

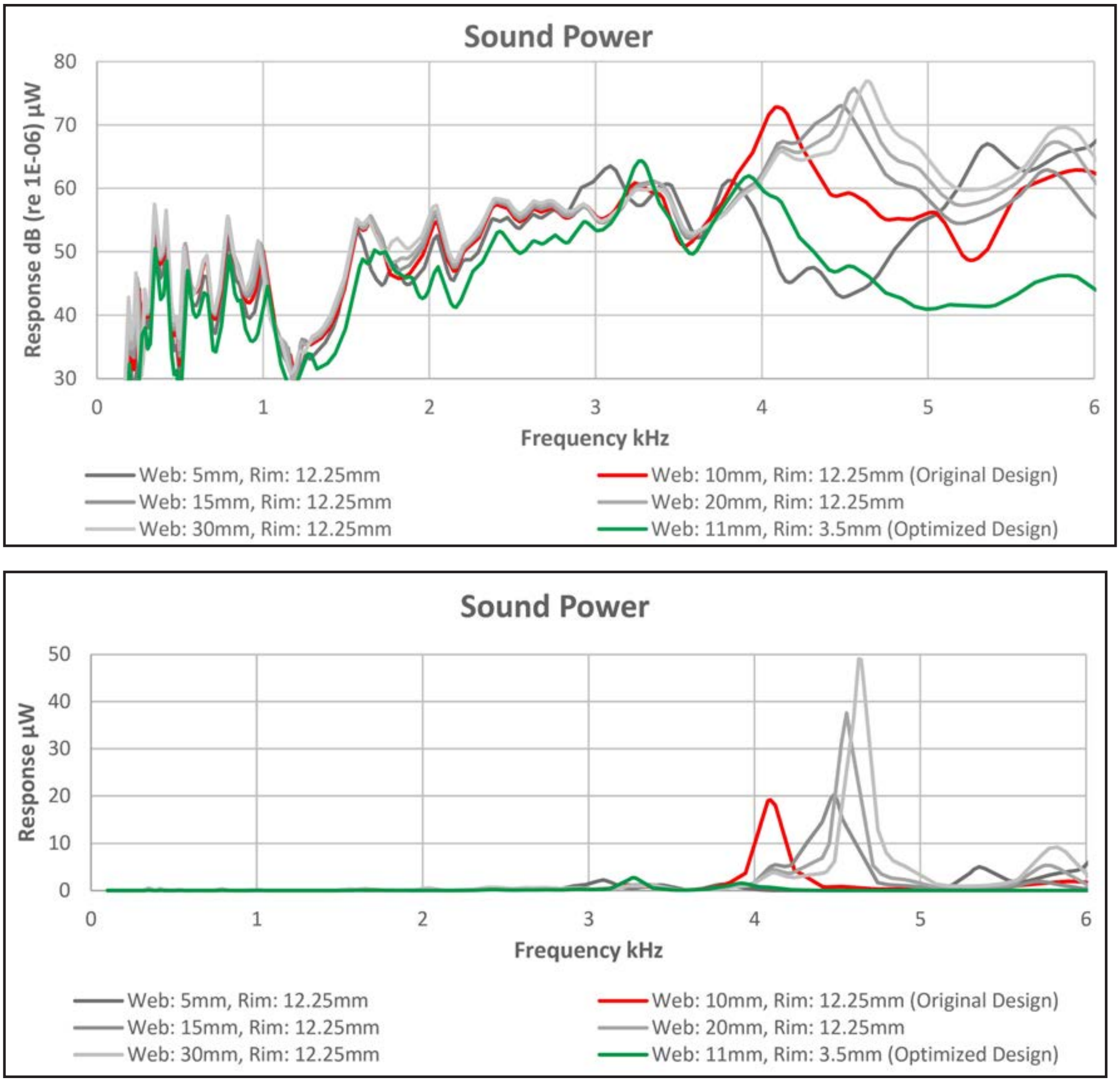

Figure 16 Casing sound power in logarithmic and absolute scale. Original design (red), optimized design (green). The other lines demonstrate the sensitivity to the web thickness alone

the mean value of 12 accelerometers at a range of points on the casing for the different gear blank dimensions, while Figure 16 shows the calculated sound power of the casing for the different gear blank dimensions.

Significant reductions are seen in casing acceleration response and sound power for the optimized solution. Comparing Figures 12 and 16, it is seen that in this case the reduction in dynamic mesh force is reflected in the reduction in sound power results at the same frequencies for each gear blank design.

It should be noted however that, in general, modifying the blank thickness modifies the TE excitation, the dynamic mesh force, and the transfer path of the dynamic mesh force from mesh to housing. The optimization is a careful balance of these effects.

Conclusion

This paper discussed the issues facing EV and HEV gearbox design to achieve demanding NVH requirements. An advanced CAE methodology was presented to analyze and optimize such designs for motor and gear whine. The method presented is available in SMT's *MASTA* software. A good understanding of the system dynamics is required. The paper used an example of gear blank tuning as an optimization option. The study shows how significant predicted noise reductions can be achieved using careful dynamic tuning. The example used in this paper was a generic non confidential example. In a real application an additional step is required to design accurate gear blank geometry which reflects the improvements seen in the study in terms of gear blank dynamics. The presented methodology has been used successfully in several engineering services projects by the authors and their clients, optimizing gear blank web and rim thicknesses for the multiple targets of:

- Weight: cost
- Gear mesh misalignment: durability and TE/noise
- Gear mesh dynamic force: NVH and gear durability application factors
- System dynamic response as bearing loads, casing accelerations, mount loads and casing radiated sound power.

The high-fidelity CAE and system dynamic optimization methodology presented provides a route to achieve quieter gearbox designs or to troubleshoot existing problem designs. Such designs can lead to significant cost savings by, for example, reducing the need for very high gear manufacturing quality and the use of

Dr. Owen Harris, a graduate of Trinity College Cambridge., has worked in the analysis of transmissions and geared systems for over fifteen years. He was instrumental in writing some of the first commercial software codes for housing influence, system modal analysis and gear whine and planetary load sharing. Harris has filled many roles in over ten years working at Smart Manufacturing Technology Ltd. (SMT). He has worked on SMT's state-of-the-art MASTA software, while at the same time being heavily involved in many engineering projects. Harris's current focus is to lead SMT's research department.

Dr. Paul Langlois is the Software Engineering Director at SMT. Having worked for SMT for 13 years, he has extensive knowledge of transmission analysis methods and their software implementation. He manages the development of SMT's software products and is a main contributor to many aspects of the technical software development. As a member of the BSI MCE/005 committee, Langlois contributes to ISO standards development for cylindrical and bevel gears.

Andy Gale is a research engineer at SMT. He has a background in mechanical engineering, and research experience in a variety of fields including additive manufacture and control systems. He has recently been a main contributor to acoustics research and development at SMT.

additional sound proofing materials within an EV or HEV vehicle.

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