Noise Reduction in an EV Hub Drive Using a Full Test and Simulation Methodology

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With the ongoing push towards electric vehicles (EVs), there is likely to be increasing focus on the noise impact of the gearing required for the transmission of power from the (high-speed) electric motor to the road. Understanding automotive noise, vibration and harshness (NVH) and methodologies for total in-vehicle noise presupposes relatively large, internal combustion (IC) contributions, compared to gear noise. Further, it may be advantageous to run the electric motors at significantly higher rotational speed than conventional automotive IC engines, sending geartrains into yet higher speed ranges. Thus the move to EV or hybrid electric vehicles (HEVs) places greater or different demands on geartrain noise. This work combines both a traditional NVH approach (in-vehicle and rig noise, waterfall plots, Campbell diagrams and Fourier analysis)—with highly detailed transmission error measurement and simulation of the complete drivetrain—to fully understand noise sources within an EV hub drive. A detailed methodology is presented, combining both a full series of tests and advanced simulation to troubleshoot and optimize an EV hub drive for noise reduction.

Introduction

A combination of regulations and consumer expectations drives the demand for reduced noise in all drivetrain components. Further demand is driven by the growing trend towards electric vehicles (EVs) and hybrid electric vehicles (HEVs), where noise from internal combustion engines is intermittent or no longer present, and the contribution of transmission noise to overall vehicle noise becomes dominant—making it more difficult to achieve customer satisfaction.

In this paper we share our experience of how NVH issues can be addressed at both design and development stages with a combination of detailed measurement and computer-aided engineering (CAE) simulations. The methodologies given for CAE simulation should be used during the design stages to try to minimize the risk of any NVH issues prior to development. However, NVH issues in fact often first arise during prototyping and in-vehicle installation. Time scales for solutions are then short, and commercial pressures high. We believe that the rapid NVH troubleshooting required is best supported by a combination of comprehensive tools, methodologies and expertise.

An Electric Vehicle Hub Drive

The gear drive to be considered is the hub drive for a recently developed, fully electric bus. The vehicle has been widely distributed worldwide as a demonstrator vehicle and represents a good, potential solution to contribute to "green" urban transport. But the vehicle is perceived by some potential customers to exhibit high noise, and so a program was undertaken to assess, understand and reduce it.

Figure 1 shows the details of the drive, of which there are two per vehicle—driving the left and right rear wheels. Key points to note include:

- The hub unit is an integrated unit, which is comprised of the electric motor, geartrain and wheel hub bearings
- The drivetrain is a reduction ratio of approximately 17
- The ratio is achieved using three gear stages
- Progressing from the electric motor, the first and second gear stages are parallel helical gearsets; the third is a helical planetary set
- The typical operating range is 0–7,000 rpm for the electric motor or drivetrain input

Noise and Transmission Error Measurements

A first step in investigating any perceived noise issue is to take detailed, objective measurements. To do this the authors developed their own in-house, detailed measurement system—*MEASA* (Ref. 1). The system combines all of the necessary hardware, data capture, and data analysis software in one integrated solution. This approach allows for more detailed work than potentially possible with commercial, off-the-shelf solutions (e.g., providing detailed transmission error measurements). It has also enabled the authors to integrate the test and CAE simulation results described in this paper in greater detail.

Figure 2 shows the details of the measurements system; it

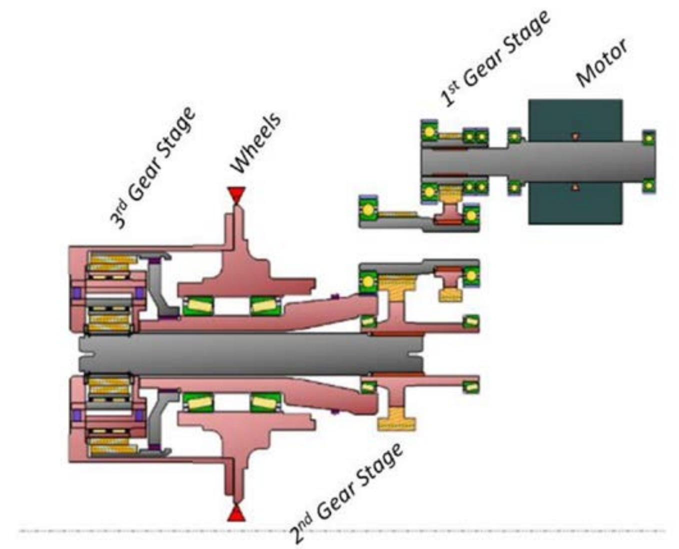

Figure 1 Cross-section showing geartrain in electric bus hub drive.

combines both traditional noise and accelerometer measurements with loaded transmission error (TE) measurements. Transmission error will be discussed in more detail below; we believe both measurement and simulation of TE can be critical in solving gear-related noise.

Noise and acceleration testing; results. The EV hub drive has been tested — in both the vehicle and on an isolated test rig; the former clearly provides the best direct indication of the unit's performance with respect to customer experience. The test rig is useful during troubleshooting, as the test conditions and environment can be much more carefully controlled than with in-vehicle testing on a test track.

It is important to note that the full test conditions do need to be more comprehensive for an electric vehicle drive with power regeneration. They are:

- Forward drive or acceleration under the full range of throttle positions, which will engage the "drive" flanks of the gears
- Forward coast or de-acceleration under the full range of breaking and power regeneration, which will engage the "coast" flanks of the gears
- Similar for reverse — although this is often considered to be much less critical — as its percentage of the drive cycle is much less than forward

Figure 3 shows the classic "waterfall-type" plots during in-vehicle testing. Note that the noise was recorded at a range of microphone positions throughout the vehicle, with the most critical found to be in a passenger seat position directly above the wheel hubs. The waterfall plots show:

- Very clear order lines are present; these are indications of quite distinct frequencies that are linearly increasing with speed. Such noise content is often described as "whine" and is a tonal noise that can stand out in subjective assessment.
- There is less evidence of more broadband noise, which is often associated with a rattling-type motion. For example, in drivetrains with an internal combustion (IC) engine, torque oscillations from the engine can generate a rattling in the gears. This EV application shows much more of a whine-type noise signature.

The next step in the troubleshooting process is to analyze the orders to identify which components can be associated with each order line.

Figure 2 Authors' in-house hardware solution for noise/vibration testing and transmission error testing, showing portable hardware used within project.

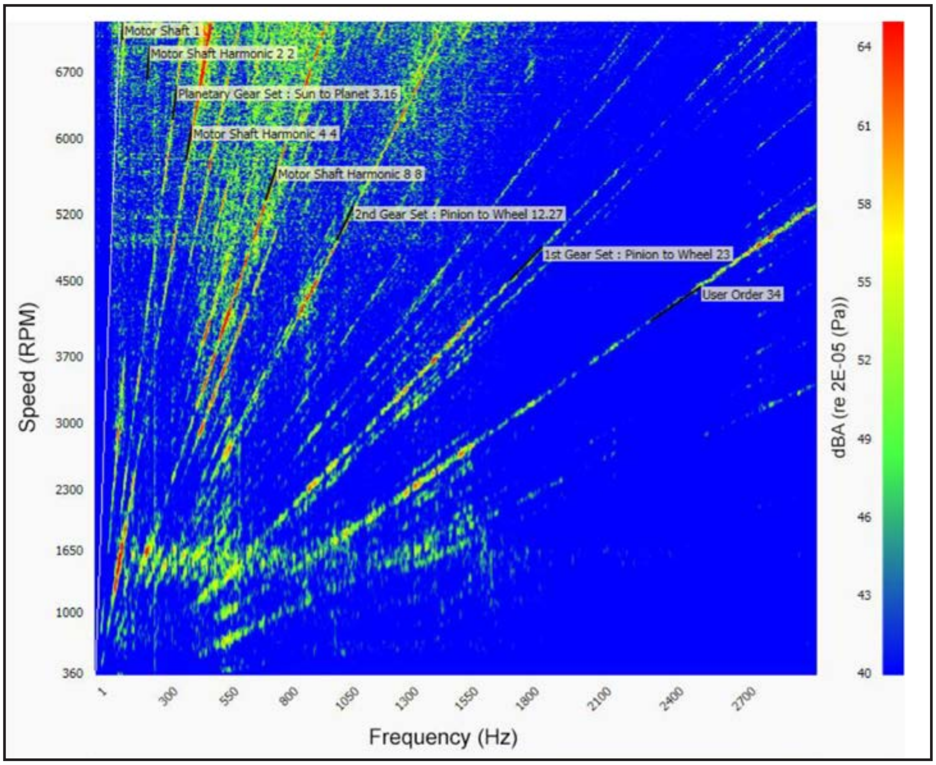

Figure 3 Waterfall plot showing typical noise measurements from in-vehicle testing of electric bus; this is drive condition with component orders superimposed.

Table 1 lists the orders corresponding to the gear stages, shafts, and bearings. These are relatively simple to calculate by hand (for example, the first-stage pinion has 23 teeth, so its order with respect to the input shaft is 23), but will become increasingly complex as transmission complexity and power flows increase as, for example, with a hybrid electric vehicle. As described in the CAE section, the authors use the commercial *MASTA* software solution to automatically generate all these orders and transfer them to the *MEASA* data analysis software (Ref. 1).

Table 1 Component orders and frequencies in EV hub drive

	Order (with harmonics)	Frequency, Hz (at input speed of 7000 rpm)
First gear stage, tooth-passing	23 (46, 69)	2683 Hz
Second gear stage, tooth-passing	12.27 (24.54, 36.81)	1432 Hz
Third gear stage, tooth-passing	3.16 (6.33, 9.49)	369 Hz
Input and motor shaft, once-per-rev	1	117 Hz
Intermediate shaft 1, once-per-rev	0.51	60 Hz
Intermediate shaft 2, once-per-rev	0.22	26 Hz
Output shaft, once-per-rev	0.056	7 Hz
Motor shaft rolling bearings	6.07, 5.57	708 Hz, 650 Hz
Input shaft rolling bearings	4.08, 6.06, 6.06	476 Hz, 707 Hz, 707 Hz
Intermediate shaft 1 rolling bearings	2.33, 2.08	272 Hz, 243 Hz
Intermediate shaft 2 rolling bearings	2.95, 2.95	344 Hz, 344 Hz
Output shaft (hub) rolling bearings	0.77	90 Hz

The results in Table 1 show how the component orders decrease as the speed reduces through the drivetrain. Also shown are the corresponding frequencies at the maximum motor/input operating speed of 7,000 rpm. It can be seen that the third gear stage frequencies are relatively low-frequency and may not be as objectionable to a passenger, compared to the higher-frequency first and second stages.

The results of Table 1 can now be superimposed on the waterfall plot to qualitatively identify which components are contributing noise; this is given in Figure 3 and shows that:

- All three gear stages can be clearly seen in the noise waterfall plots; further harmonics of the fundamental orders can be seen.
- In addition, some very clear harmonics of the once-per-revolution order of the input shaft are present — in particular, orders 4–9; these have been identified as orders associated with the electric motor and are beyond the scope of this paper and the geared drivetrain optimization.
- Little evidence of explicit bearing passing orders is seen.
- An unexplained order 34 is present; a potential source is a "ghost" gear frequency — i.e., an artefact of the gear manufacturing process. This was confirmed during the project and this order was eliminated with optimized gears.

In order to quantify the contributions of these identified orders, order slices — or order plots tracking the contribution versus speed — are plotted in Figure 4; total noise is also given for comparison. These order plots reveal a clear indication of how much each gear mesh contributes to the total noise and identify any speed regions where system resonances excited by a component occur.

A critical metric often considered is the minimum difference between the gear order noise and the total noise across the full speed range. For an IC engine application, the authors believe that targeting a minimum difference of 12 dBA in the in-vehicle noise will ensure that no gear whine is detected by the passenger. For EV and HEV applications the gear noise contribution can be greater, as a percentage of total noise due to an intermittent or non-existing IC contribution. In this project an absolute reduction in gear order noise was targeted.

Introduction to Theoretical Transmission Error and Gear Whine.

As seen in the previous section, noise and vibration testing can be used to identify clear tonal noise corresponding to once-per-tooth passing orders of gears; this is often referred to as gear whine (Ref. 2).

Gear whine is an NVH phenomenon, most commonly sourced from transmission error (TE) at engaged gear meshes. Theoretically, an infinitely stiff gearset with perfect involute form and no misalignment would transfer angular veloc-

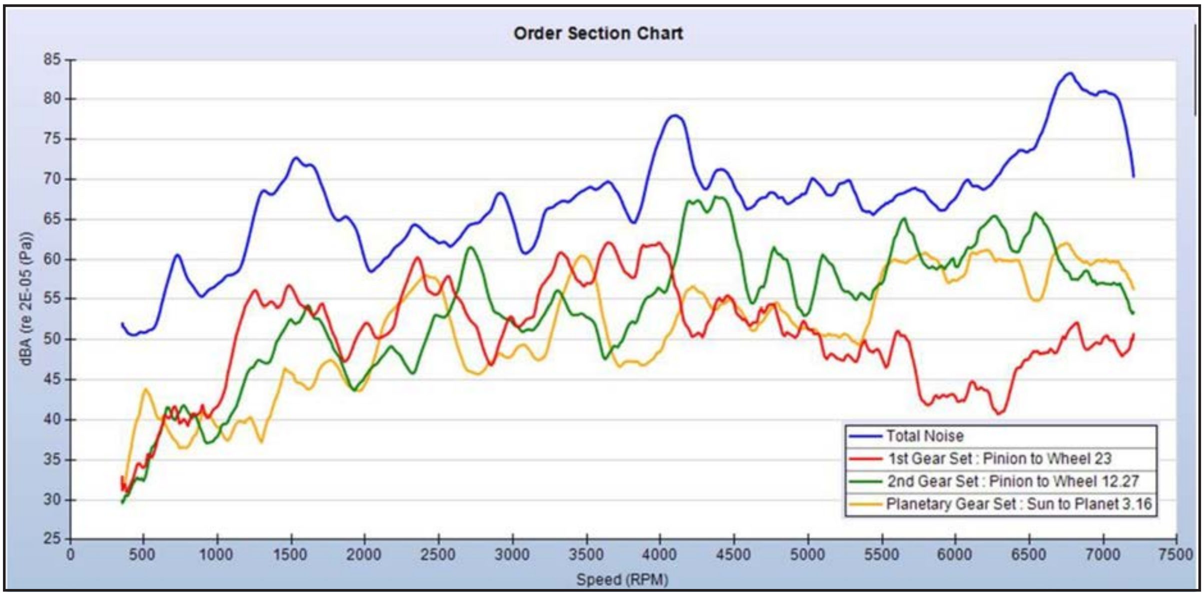

Figure 4 Order cuts through waterfall plot, showing typical noise measurements from in-vehicle testing of electric bus; this is the drive condition.

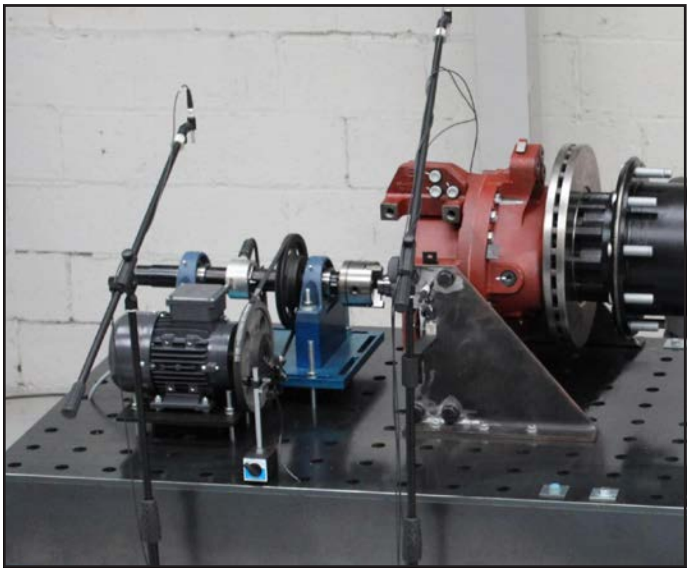

Figure 5 **Test rig for making transmission error, noise and accelerometer measurements while running with light load.**

ity exactly in accordance with the designed ratio. However, in reality no gear is perfect and, for example, tooth bending and misalignment caused by deflections of the system contribute to real gears not performing to this ideal. TE is the difference between the angular position that the output shaft of a drive would occupy if the drive were perfect — and the actual position of the output. Note that other authors may use the actual position — minus the expected position. In this paper we use expected minus actual. Under this convention TE values are usually positive, as the actual position is typically less than the expected, due to take-up of backlash and compression of the system under load. Other potential, but less common, sources of gear whine whose fundamental frequency is also at once-per-tooth, include axial shuttling forces where the axial location of the resultant force varies through the mesh cycle, resulting in a varying moment on the gears, and friction forces from the relative sliding at the gear mesh (Ref. 3).

Transmission error can be considered a periodic, relative displacement at the gear mesh in the line of action, caused by less-than-ideal meshing conditions. TE can dynamically excite the transmission via a path from the gear mesh, through the shafts and bearings, and on into the transmission housing. Gear whine is the resulting tonal noise radiated from the housing or transmitted from the housing and radiated elsewhere. Gear whine should therefore be considered not just a gear problem — it is a systemic problem — with the gears as the exciters of the system.

Transmission error: testing and results. For a multi-mesh drivetrain such as this we will refer to the drivetrain TE (with respect to the drivetrain input and output) as the "system TE." The concept of TE is often associated with single gearsets. The system TE will be the summation of the three gearsets' TEs, noting that TEs are periodic and their relative phases are important.

Figure 5 shows the experimental set-up used to measure the system TE of the EV hub. As discussed above, this measurement should fully capture the TE as the source of noise. In this set-up we have measured the full drivetrain, as assembled; this ensures that we capture all the potential sources of the TE. For example, build quality misaligning the gears and gear manufacture quality, as well as the fundamental gear design. In addition, by moving the position of the input and output encoders it was possible to measure sub-parts of the drivetrain (e.g., just the 2nd stage).

In order to derive the individual gear mesh TEs, it is necessary to use Fourier analysis to decompose the total system TE signal into those components corresponding to each mesh. This will use the different gear orders (with respect to the input) given in Table 1. Figure 6 shows this process. Key points to note include:

- The total TE has quite large periodic variations — at once-per-revolution of input, intermediate and output shafts. These can be attributed to run-out of these shafts or the encoders, which is likely to be present to some degree in all drivetrains. These oscillations, from a noise perspective, will be low-frequency and not relevant to gear whine.
- Where the data is examined on shorter (higher frequency) timescales, the once-per-tooth oscillations corresponding to the gear meshes are seen.
- The Fourier analysis clearly shows how total TE is composed

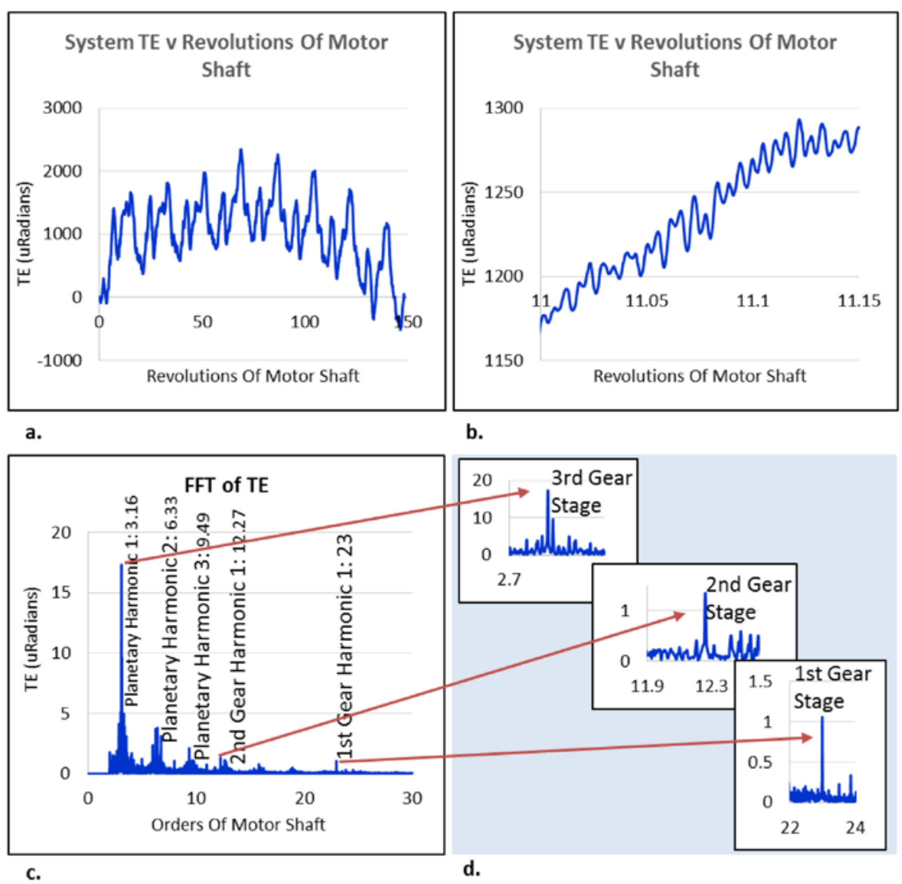

Figure 6 **Transmission error testing results showing data analyses required to extract individual gear mesh TEs. From top: graph (a) full raw system TE; (b) zoom to smaller timescale showing oscillations due to tooth passing; (c) and (d) show Fourier transformation.**

of the TE due to the gear meshes. The meshes of the three gearsets and their corresponding harmonics can be clearly identified.

- Some sideband structures can be seen. For the second stage the sidebands are predominately at one-order intervals with respect to its pinion shaft, suggesting pitch error or other manufacturing errors of the second stage pinion gear. The sidebands structure of the third stage may not necessarily be due to manufacturing error, as sideband structures are a known artefact of planetary gear sets (Ref. 4).

For cylindrical gears, TE values are typically given as a linear value derived from the angular error across a mesh multiplied by the base radius of the reference gear. Table 2 lists these for the first and second stages. It is necessary, when extracting the TE of individual meshes, to include the speed ratios through the drivetrain. The system angular TE is a measurement of the combined, relative angular position error due to all meshes in the drivetrain; it is the product of not only each individual gear mesh TE, but also the speed ratio between each mesh and the reference point of the measurement (output). Therefore, in Figure 7/part c, the contribution of each mesh will appear greater or smaller than its equivalent linear mesh TE, depending upon whether the base radius of the gear used to calculate linear TE is rotating more slowly or more quickly than the measurement point. The TEs of the slower speed meshes appear higher within the FFT of the system angular TE, but are not necessarily higher when recalculated as an individual gear mesh linear TE.

Relationship between noise/vibration and transmission error testing results. A further inspection of both the standard noise and acceleration and detailed system TE test results is instructive in the troubleshooting process. Figure 7 shows results from rig testing of the same unit on the same rig. It can be clearly seen that the characteristic sideband structure seen in the system TE is also present in the acceleration and noise data, thus confirming the underlying assumption that the gear TE generates both the casing acceleration and noise.

Testing and data analysis conclusions. In conclusion, the testing methodology is set out, highlighting the importance of both noise and vibration and transmission error testing. In the next section, we describe the role of CAE in supporting the interpretation of the results and developing solutions to reduce noise.

CAE for Gearbox Noise Reduction

In this section we discuss — using the EV hub dive — how CAE can be used to support noise reduction. Where appropriate, comparisons are drawn between the previous test results and the CAE simulation.

A *MASTA* (Ref. 1) model has been built to simulate the EV hub drivetrain, as described in the following sections.

Power flow and excitation orders. This has already been covered in the (Noise and Acceleration Testing Results) section. A "power flow-type" model is required to calculate all the component speeds and derive the excitation orders. In particular, we have identified the tooth passing orders for the gear meshes and superimposed them on the noise and TE measurement results. An integrated solution between the software used to analyze the test data and the CAE software proves advantageous for speed and accuracy in performing this operation. In this study the order data from the corresponding *MASTA* model was exported from *MASTA*, via an XML file, and imported into SMT's *MEASA* software for the analysis of the measurement results. This process enables the automatic identification of excitation orders on the measured data plots (Fig. 3).

Table 2 Individual gear mesh TE results expressed as linear peak-to-peak

Gear-set	Peak-to-peak linear TE (urn)	
	Drive	Coast
First gear stage	1.55	4.16
Second gear stage	3.88	4.24

System deflection. Simulation and testing show that gear mesh misalignment will both potentially degrade gear contact patches (reducing life) and increase transmission error — and thus noise. A full CAE model is required to model and simulate the deflection under load of the full system. The EV hub model consists of:

- Full gear geometry from which the gear forces can be accurately calculated
- Shafts modeled using finite element beam elements or a 3-D stiffness and mass imported from a full finite element model, where appropriate (e.g., for gear blanks)

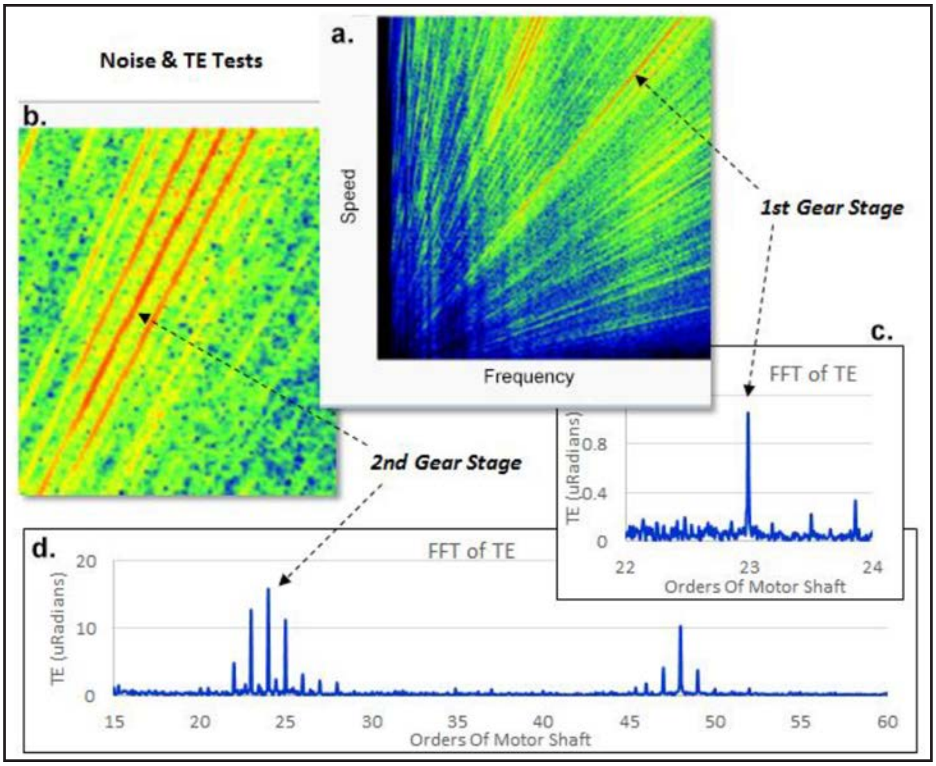

Figure 7 Results showing casing acceleration and transmission error testing results of EV hub drive; same unit with same light load is used for all tests. Detail: (a) waterfall plot of casing acceleration with clear 1st and 2nd stage gear order lines; (b) zoom of second-stage order showing clear sideband structure; (c) and (d) show TE results with respect to these two meshes.

- Bearings modeled using non-linear, load-dependent six-degree-of-freedom stiffnesses, derived using bespoke methods from the bearing internal geometry details and a Hertzian contact model (Ref. 5)
- Casing modeled using 3-D dynamic stiffness and mass reduction imported from a full finite element model

Typical results are given in Figure 8, showing the deflection of the full system; the critical gear mesh misalignments are given in Figure 9. An important role of CAE is that it allows for the simulation and identification of those components which contribute the most flexibility and are thus candidates for optimization (Ref. 6). Figure 9 also includes misalignment results showing the effect of the casing flexibility. For this unit, we can rule out poor bulk casing flexibility as a contributing factor to the noise issue. The model shows that the gear misalignments are due to shaft and bearing deflections under load. A methodology the authors often use in projects is to build further models with stiff/flexible shafts and bearings in order to identify exactly how much misalignment can be attributed to each component. This level of detail is not given here for brevity.

Loaded tooth contact analysis (LTCA). Loaded tooth contact analysis (LTCA) is used to calculate the loaded contact conditions for gears as they progress through the meshing cycle. One critical input for this calculation is the gear mesh misalignment, as calculated in the previous section. The key result with respect to noise is the TE. Torque, misalignment and gear macro- and microgeometry are also used as inputs.

It is important to use an accurate LTCA in order to get an accurate calculation of TE. A hybrid FE and Hertzian contact-based formalism (see Ref. 7 for a formalism similar to that used) is used to accurately capture the stiffness at each contact location, while providing a fast calculation suitable for assessing microgeometry parameter changes and robustness to tolerances (Fig. 10). Such a calculation is comparable in accuracy to a full FE contact analysis, while also being many orders of magnitude faster. An FE model of the gear macrogeometry is built automatically in the

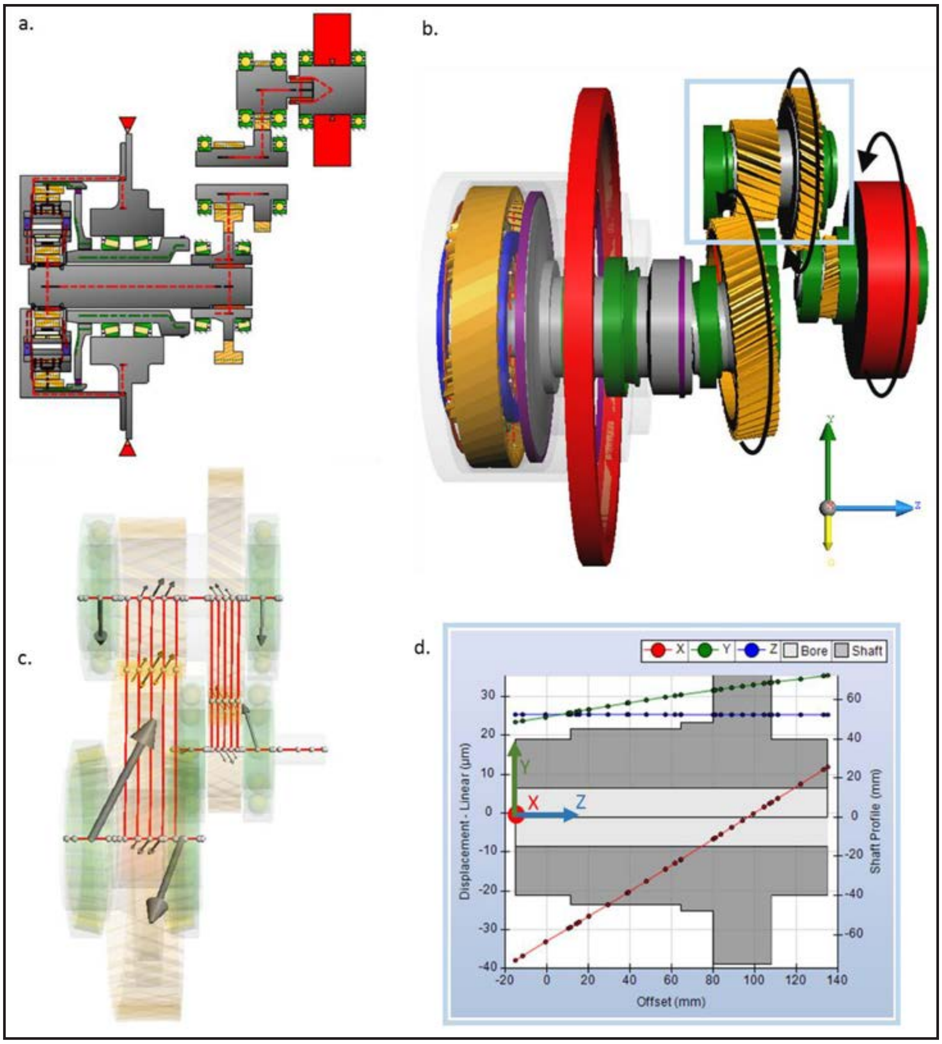

Figure 8 Predicted deflection of EV hub drive. Image a) power flow; b) deflection of full drivetrain; c) gear and bearings loads on first two stages; d) predicted deflection of intermediate shaft 1.

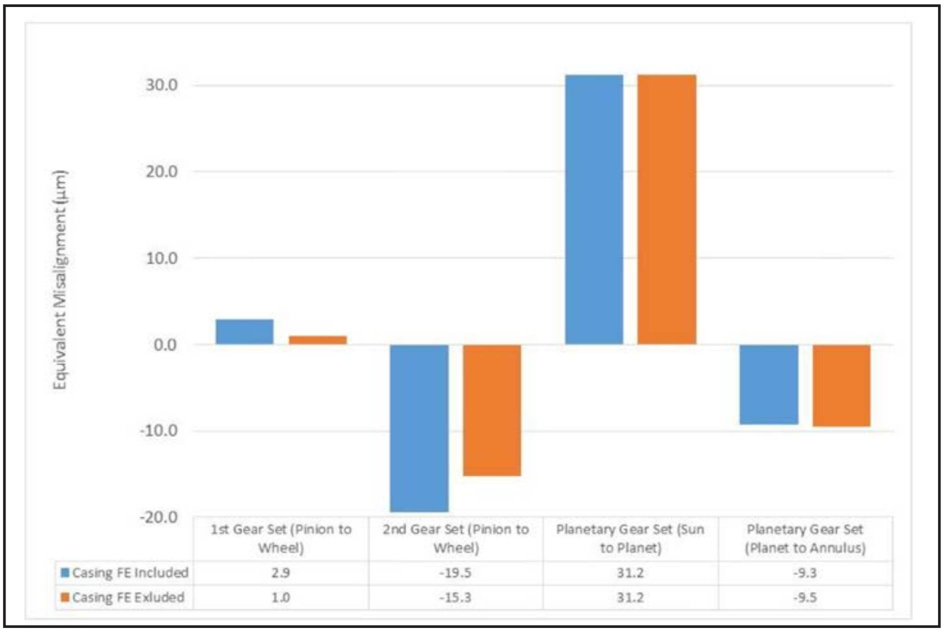

Figure 9 Predicted gear mesh misalignments—with and without including effects of casing stiffness.

software and is used to obtain the overall bending and base rotation stiffness of the gear teeth, with consideration made for coupling between teeth. This bending stiffness is combined with a Hertzian line contact formalism to calculate the overall stiffness of any potential contact points. Potential contact lines are split into strips, and force balance and compatibility conditions are formulated and solved to calculate the load distribution across the mesh and the transmission error for the input torque. This loaded tooth contact analysis can be used to optimize both gear microgeometry and macrogeometry for minimal transmission error. Consideration must be given to the entire operating range of loads and the robustness of the proposed design to variation in load and misalignments, as well as variation in gear microgeometry within the manufacturable tolerance range.

The correlations between test and simulation are often insightful with respect to solving noise issues. If correlation is good, this boosts confidence in the analysis model, implying that calculated misalignments, microgeometry inputs, calculated load distribution and TE are accurate and the model can be taken forward to explore design changes. Conversely, differences between test and simulation can often highlight manufacturing problems. In this case the model showed good correlation for the contact patches (Fig. 11). The TE analysis and test results proved important in highlighting manufacturing issues to be corrected, including the second-stage sideband structure in test and considerably higher measured peak-to-peak TE than predicted from the nominal gear design geometries. There will always be some manufacturing variation with respect to the nominal design, but the increases in measured TE compared to simulation in Figure 12 were judged to be too great, and a thorough audit of manufactured tolerances was undertaken. Poor control of a number of critical tolerances was identified and rectified, yielding improved noise performance.

Advanced system deflection and system TE. A further, more advanced simulation is to combine the LTCA and system deflection calculations — known as "advanced system deflection" (ASD) — in *MASTA*. In the ASD the LTCA assumption that the misalignment is constant through the meshing cycle is relaxed, the misalignment is recalculated based on the gear load distribution from the LTCA and, conversely, the gear load distribution is recalculated with the corrected misalignment. An iterative solution is followed to reach equilibrium for each meshing position.

In a number of important cases, such as the tooth contact conditions of a planet gear, the interaction between two or more meshes of the planet means that the system deflection and tooth contact conditions are best solved with this coupled ASD calculation. Further, for planetary systems where contact conditions may vary as the planet carrier rotates, such a coupled calculation can be used to calculate the load distribution and transmission error as the planets precess.

For the EV hub, this calculation was used to simulate the full system TE — including gear tooth pitch errors (Fig. 13). The simulation was run with nominal geometry; comparison to test confirmed that the second stage had significant contributions from manufacturing errors, giving a sideband structure.

Modal analysis and gear whine simulation. The full CAE model described above can also be used to analyze the dynamics of the system via modal and harmonic response analyses.

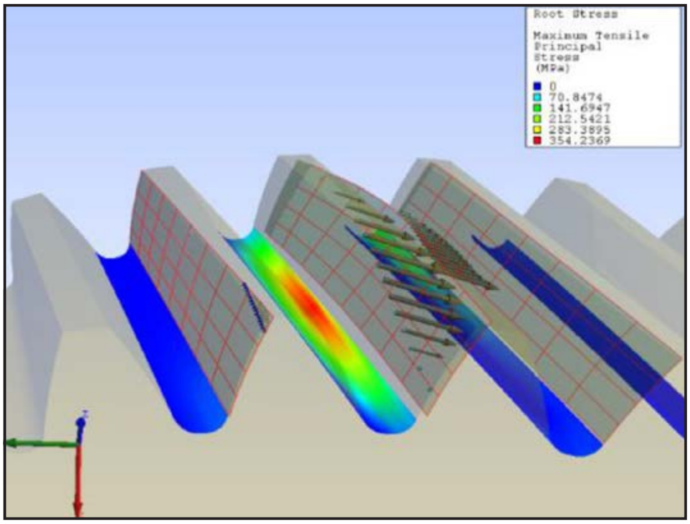

Figure 10 Hybrid FE and Hertzian-based LTCA used to calculate gear mesh load distribution, contact and root stress (shown in this figure for one roll angle) and transmission error, in order to optimize gear geometry for low noise.

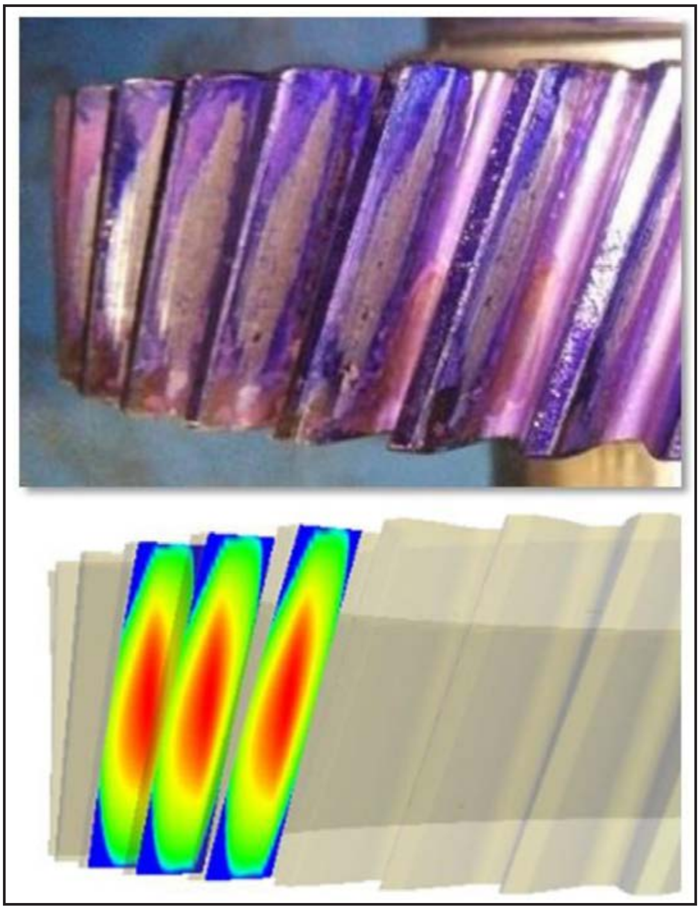

Figure 11 Test and predicted gear contact patch results from EV hub drive.

For a modal analysis of the system at a given input load, a linearized model of the non-linear static analysis model is used.

The calculated natural frequencies, mode shapes, modal strain and kinetic energies, and Campbell diagrams can be used to identify potential excitations of the system where, for example, gear mesh frequencies or their harmonics cross the natural frequencies of the system; further inspection of the energy content (strain or kinetic) of the mode shapes can be used to identify the main contributing components to those potential resonances. For example, modes with significant strain energy in the gear mesh modeling element are most likely to be excited by TE. One target would be to minimize the number of natural frequencies within the operating range while also separating any that happen to lie within the range from of each other.

The method of calculation of the system response to the TE introduced by (Ref. 8) can be used to calculate the casing acceleration at virtual accelerometer locations. As the excitation is periodic and the stiffness around the loaded condition can be considered linear, calculation can be performed very quickly in the frequency domain. The static TE described above is the assumed excitation input of the system and the first step is to calculate the dynamic force at the gear meshes that leads in turn to a relative displacement at the mesh given by this transmission error. This force is known as the "dynamic mesh force" and is calculated from the dynamic compliances at each side of the gear meshes. The dynamic mesh force is then applied as an excitation to the system model to calculate the response (at any point on the system) to this excitation. Waterfall plots of dynamic response for any point on the model can be shown and compared with accelerometer and/or microphone data obtained via noise and vibration tests.

Figure 14 shows a comparison of test and simulated accelerations on the casing of the EV hub drive. Plots are given here for SPL (sound pressure level) versus speed, which is of most direct interest to a vehicle manufacturer; however, SPL versus frequency plots is also

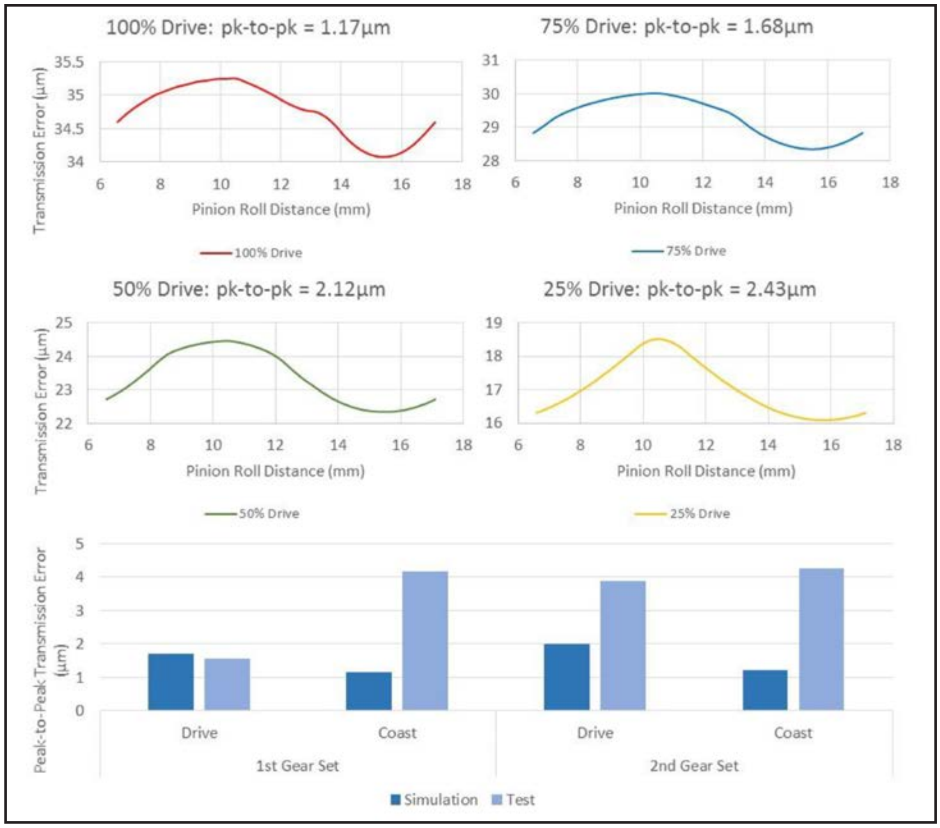

Figure 12 Transmission errors; simulated transmission errors in top four graphs showing sensitivity to load; bottom two graphs show simulation and measurements of rejected hub drive unit — both at light load. 100% drive denotes max motor/input torque over typical drive cycle.

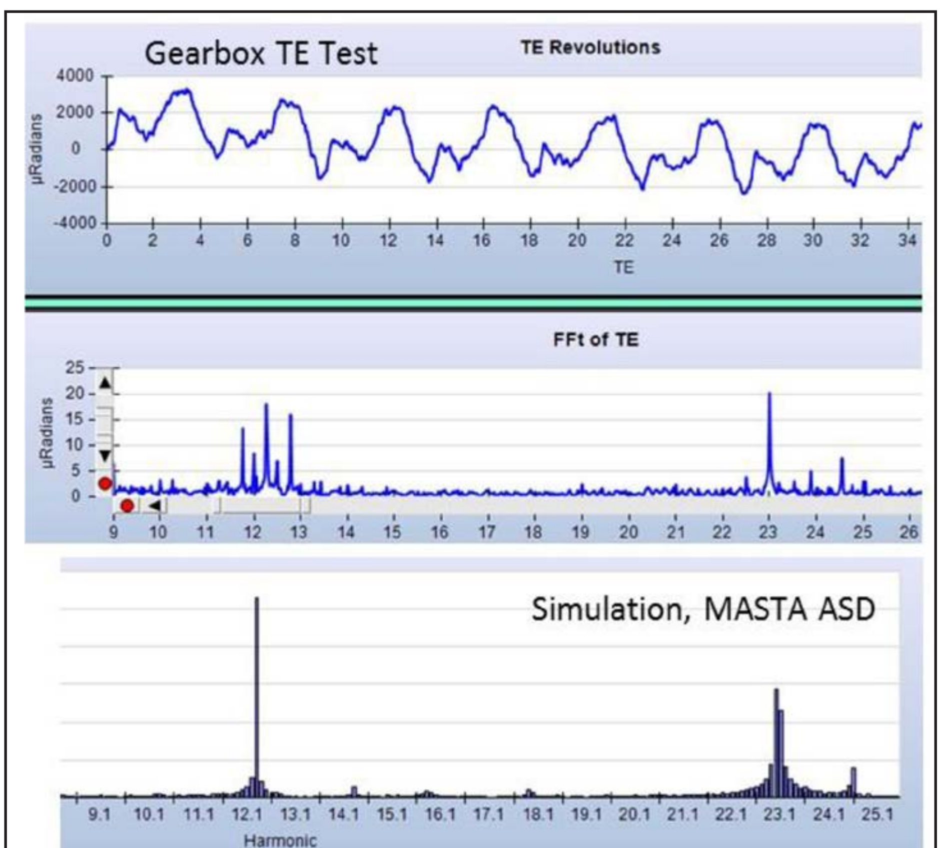

Figure 13 Predicted and test results for full drivetrain system TE.

instructive; e.g., checking for presence of resonances). The results provide a qualitative comparison and were used to guide the troubleshooting team in their noise reduction strategy. As an example, no clear, isolated resonances were identified, suggesting that casing dynamic optimization was not necessarily a good route, but gear geometry optimization, inspection and improvement of manufacturing quality were.

Design optimization for noise reduction. The first stage of the optimization process is presented here; further optimizations are in progress. This first stage focused on redesign of both the gears' macro- and microgeometries to reduce TE. Advanced CAE tools described in the previous sections were used to make the modifications and guide and asses the design improvements (Fig. 15). The main optimization tasks undertaken within this stage can be summarized as follows:

- Assess gear macrogeometry for potential tooth number changes to improve contact ratios while taking care not to move tooth passing frequencies to coincide with any system resonances within the operating range.
- Further macrogeometry optimization of gear module, helix angle, pressure angle, and tooth height to improve contact ratios and minimize predicted TE across full operating range of loads.
- Microgeometry redefinition and optimization again, for minimized predicted TE across full operating range of loads; note that the full *MASTA* CAE model of the gearbox is used to provide the gear misalignment input to the TE calculations.
- Ensure that predicted durability results are not compromised by the design changes for reduced noise.

The results of prototype testing of this first phase of design changes are given in Figure 16. These in-vehicle noise measurements show significant individual gear noise contribution reductions. Total noise reductions of about 12 dBA, with respect to the peak value across the operating range, have been achieved.

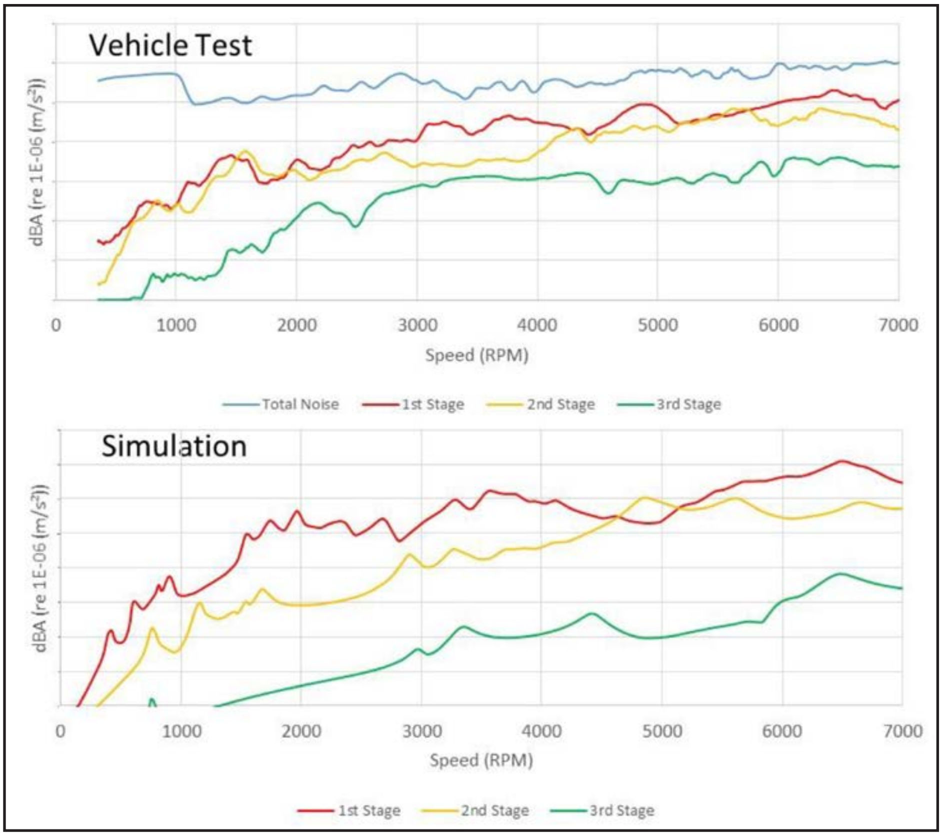

Figure 14 Comparison of test and simulated accelerations on casing of EV hub drive. On the vertical axis, there are 10 dB per division.

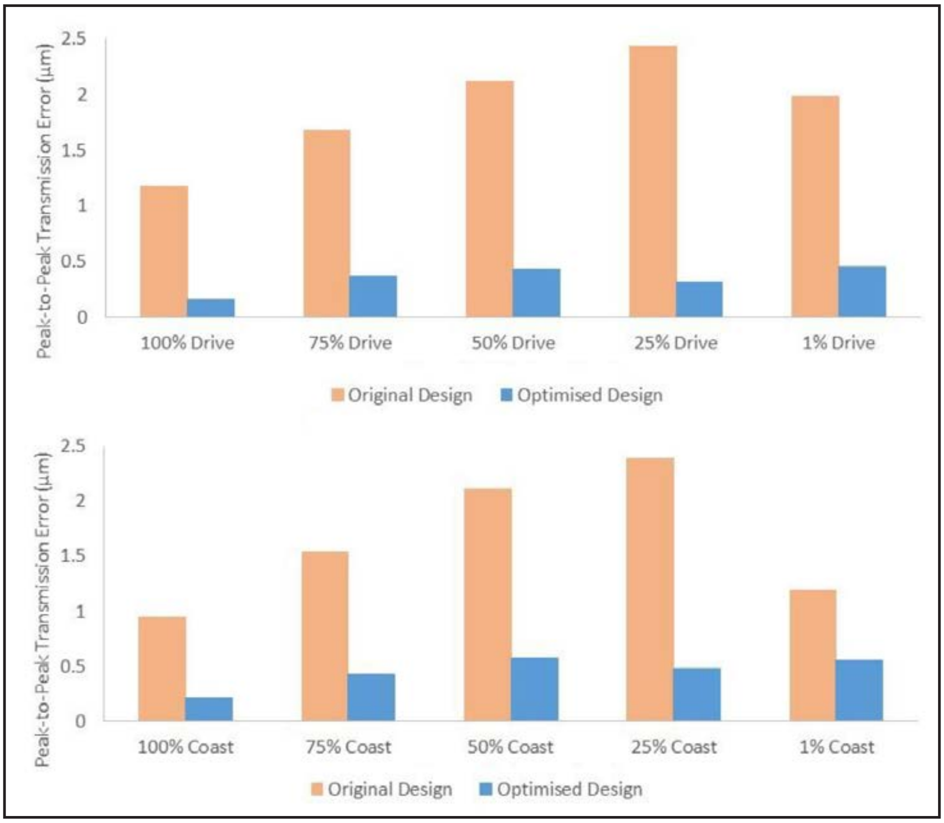

Figure 15 Results for simulated TE for second-stage gearset across full load range — both drive and coast conditions for original and optimized gear designs.

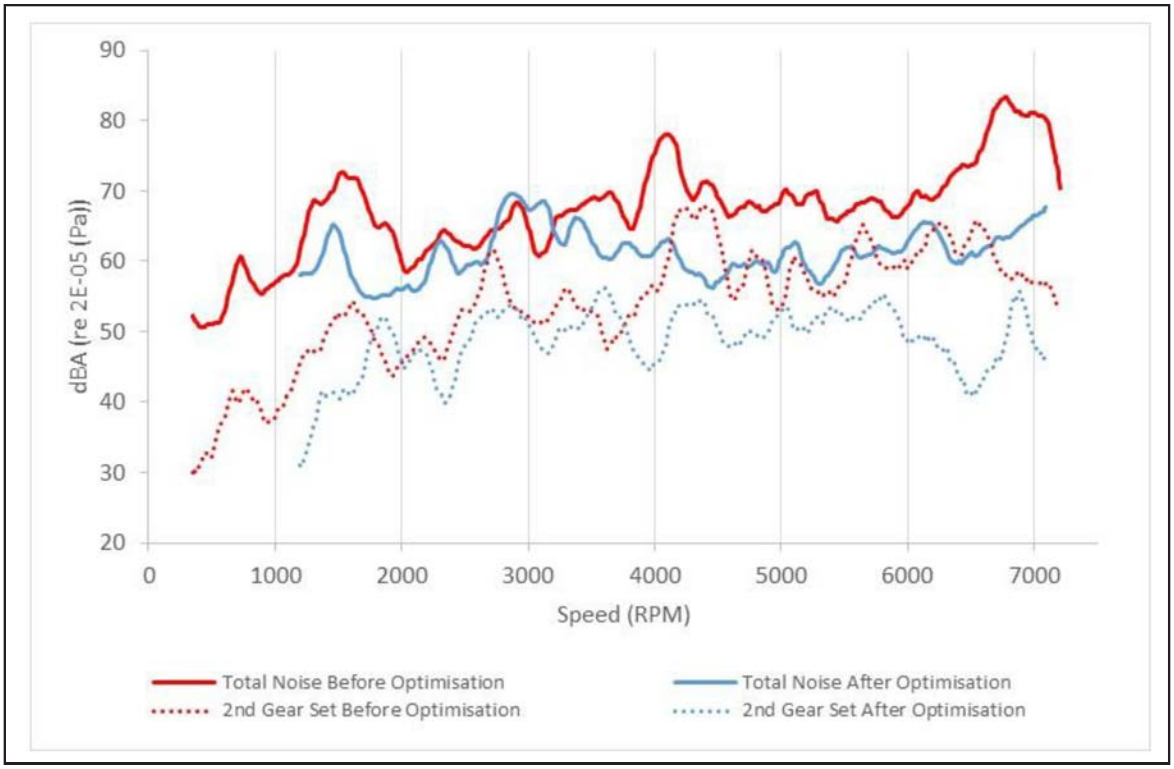

Figure 16 In-vehicle total noise and second-stage noise contribution—before and after a first optimization of EV hub drive; peak value across speed range was 82 dBA in original design and 70 dBA after optimization.

Conclusions and Future Work

This paper describes how gear-related NVH issues can be important during the development of EV and HEVs. It describes how a strong combination of testing and CAE tools, together with a solid methodology, can provide efficient solutions for such issues. A system-level approach to both processes is recommended to fully capture the interactions of all components. A case study of a hub drive for an electric bus was presented, showing how such an approach led to significant noise reductions within a first round of design optimizations.

Some of the authors' future work will be focused on further integration of test data analysis and CAE simulation tools, so more rapid comparison of test and simulation results can be made.

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