

Multibody Simulation of Helical Gear Noise and Vibration Behavior Using MSC ADAMS[†]

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Abstract

The premium electric-vehicle market demands exceptionally quiet transmissions because the absence of engine masking makes gearbox noise more perceptible. Virtual NVH (noise, vibration, and harshness) evaluation requires coupling elastic deformation, gear-tooth contact, and vibration transmission through bearings and housing within a single environment. This study develops an integrated workflow in MSC ADAMS for predicting the NVH behavior of a 23/81-tooth helical gear pair. Finite element-based flank stiffness is imported, and a nonlinear contact model is applied to flexible teeth. Baseline simulation at 50 Nm and 200 rpm yields a static transmission error (TE) of 7.5 μm and a dynamic peak-to-peak TE of 0.7 μm , with the fundamental mesh tone at 77 Hz. Increasing tip relief by +0.10 mm lowers RMS TE by 31% and the first mesh order by 3.1 dB while raising the flank pressure from 1.65 GPa to 1.88 GPa. The workflow efficiently supports early-stage gear-noise optimization prior to the development of physical prototypes.

Keywords: helical gear; multibody dynamics; transmission error; NVH; MSC Adams



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1. Introduction

With the masking effect of internal combustion engines removed, the tonal mesh noise of gearboxes has become one of the most perceptible sound sources inside and around an electric vehicle (EV) [1]. Although helical gears deliver high load capacity and smoother meshing than spur gears, the periodic transmission error (TE) that remains after manufacturing corrections still excites structure-borne and airborne noise at the mesh frequency and its harmonics. Transmission error is fundamentally unavoidable due to the flexibility of the gear teeth and manifests as fluctuation in the output rotation speed and torque even for constant speed and torque input on the input shaft of a gear pair. This phenomenon is also referred to as time-varying stiffness of the gear mesh and is considered the main excitation source in drivetrains [2].

Elastic multibody (eMBD) models that import finite element (FE) flank stiffness data can predict TE excitation, the vibroacoustic phenomena, and the radiated noise with a few decibels' deviation from the test results [3]. Yet most published workflows [4,5] (1) shuttle data among separate CAD, FE, multibody, and acoustic solvers; (2) rely on empirically tuned mesh stiffness functions; (3) initiate optimization after a validated baseline has been carried out.

Tutorial [6] supplied with MSC ADAMS Machinery documents every process step needed to construct a flexible tooth–gear pair. However, it does not determine how raw

outputs should be converted into objective NVH metrics or how micro-geometry changes translate into audible benefit [5].

Research Gaps and Contributions

Novel EV development processes demand virtual simulation methodologies, including a gear noise prediction workflow that

1. Runs within a single environment, thus avoiding version control and data transfer delays [1,3];
2. Delivers TE time histories, mesh-order spectra, and flank pressure data directly comparable to target characteristics [6,7];
3. Supports geometry optimization in the concept phase, before prototypes exist [4].

The present work closes this gap by extending the MSC tutorial into a self-contained NVH workflow based on a case example gear pair. A 23/81-tooth helical pair is modeled entirely inside MSC ADAMS: FE-derived flank stiffness is imported, dynamic TE and torque ripple are extracted, Hamming-window fast Fourier transform (FFT) is applied to obtain mesh-order spectra, and per-tooth Hertzian pressures are logged. The baseline model predicts a static TE of 7.5 μm and a dynamic peak-to-peak TE of 0.7 μm at a 50 Nm load; the dominant spectral line appears at 77 Hz, matching the theoretical mesh frequency. The maximum flank pressure reaches 1.65 GPa, safely below the ISO 6336 pitting limits [8]. A parametric tip relief sweep then quantifies the trade-off between TE reduction and pressure rise, providing practical guidance for early EV gearbox design. The detailed methodology and results follow, in Sections 2 and 3, respectively.

2. Methodology

2.1. Workflow Overview

All modeling, simulation, and post-processing steps were executed in MSC ADAMS 2024.1. Figure 1 summarizes the single-environment workflow: first, the CAD geometry is created; next Gear AT flank generation is carried out, which is followed by the FE tooth-stiffness export; then eMBD simulation is executed; and finally the NVH post-processing is accomplished. The procedure follows the vendor tutorial (Creating a Single Helical Gear Pair; MSC Software Corporation, Adams help, 2024.2) [6].



Figure 1. Single-environment workflow.

2.2. Gear Geometry

The single-stage external helical pair shown in Figure 2 was chosen to resemble the first reduction in a compact e-drive. The Macro-geometric data are collected in Table 1. The gear bodies are rigid; the flanks are flexible.

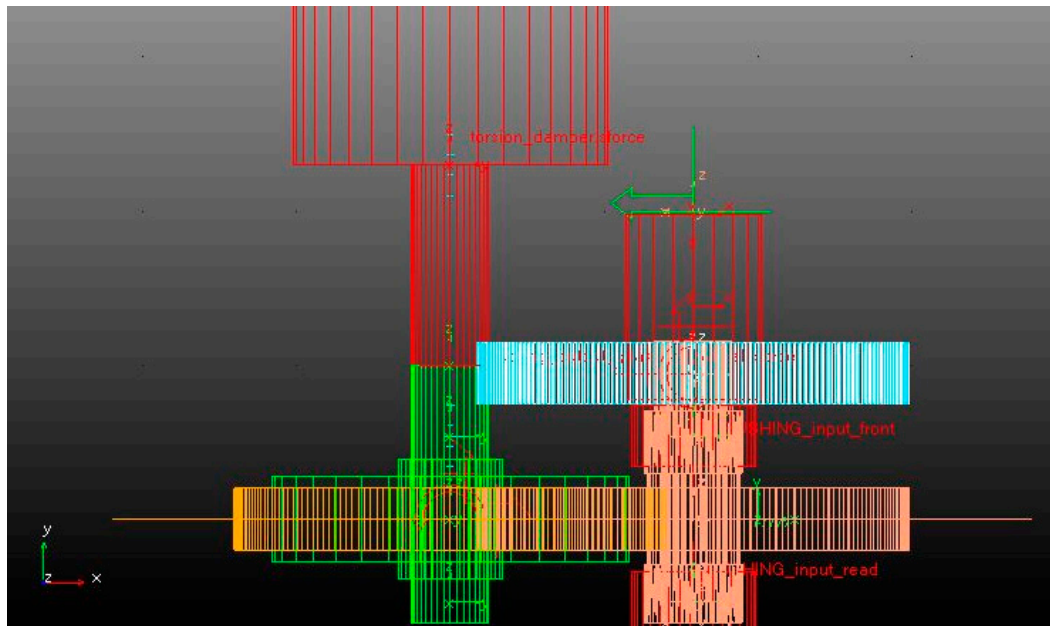


Figure 2. Helical gear pair model topology in MSC ADAMS (side view).

Table 1. Main Geometric Parameters of the Helical Gear Pair.

Parameter	Pinion	Wheel
Number of teeth	23	81
Normal module	1.395 mm	1.395 mm
Helix angle (β)	24° LH	24° RH
Pressure angle (α)	20°	20°
Face width	30 mm	20 mm

2.3. Finite Element Flank Compliance

The transmission error of gear drives is highly sensitive to local flank compliance. The local compliance depends on the elastic deformation of gear teeth under load. To accurately capture this phenomenon, an FE-based approach was employed. The FE model was then condensed to a stiffness representation for computational efficiency.

Using Gear AT Mesh/External ViewFlex, (MSC Software, Adams 2024.2 module), each flank was discretized with 30 contact planes as shown in Figure 3 to resolve stress and deformation gradients. The tooth meshes were then represented by 6×6 stiffness matrices (*.cgs) using MSC NASTRAN SOL 101, preserving modal accuracy up to 8 kHz. This approach aligns with recent recommendations for eMBD validation studies [3].

2.4. Drivetrain Assembly and Boundary Conditions

This subsection briefly summarizes the setting up and assembly of the shafts and bearings, the input profile, the load, and torsion damper. Rigid steel shafts with a density $\rho = 7850 \text{ kgm}^{-3}$, Young modulus $E = 210 \text{ GPa}$, and Poisson-ratio $\nu = 0.3$ were connected to the gears via reference markers. Radial and torsional bearing stiffness and damping were set to catalog values for the ball bearings with 30 mm deep grooves. The input profile started with a speed ramp from 0 to 200 rpm over 0.05 s to reproduce electric motor run-up. Thereafter speed was held constant. A load of 50 Nm resisting torque was applied at the wheel, generating the empirical torque curves later shown in the Results Section. A small torsional viscous damper (1.5 Nms/deg) flattened start-up spikes without affecting steady-state mesh dynamics.

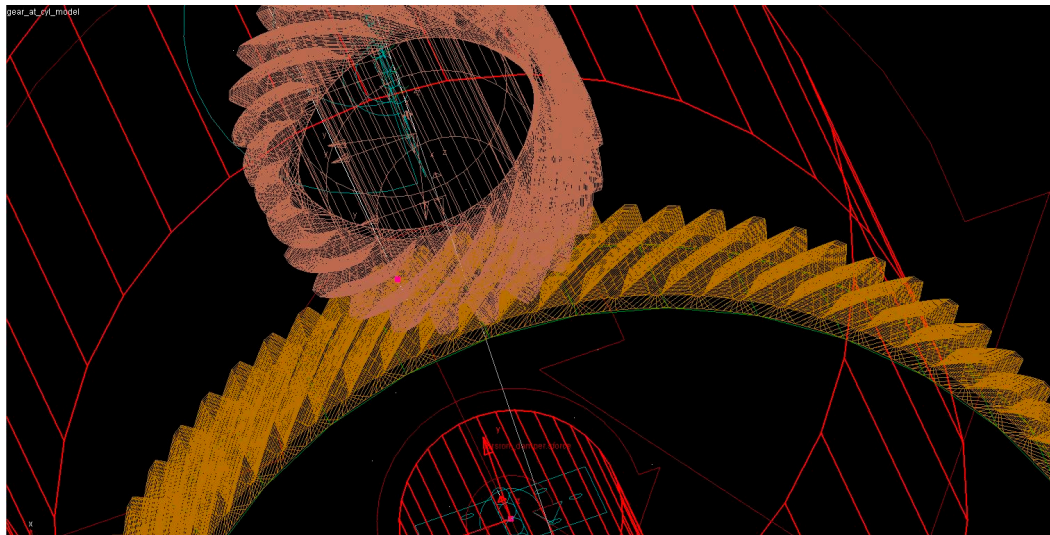


Figure 3. Helical gear contact mesh detail between pinion and wheel.

2.5. Solver Configuration

The implicit Hilber–Hughes–Taylor (HHT) integrator method was selected for its numerical damping of high-frequency chatter [3]. The key settings include an error tolerance of 1×10^{-6} , maximum step size $5 \cdot 10^{-5}$ s, and simulation time 0–0.40 s (≈ 400 mesh cycles).

2.6. NVH Output Requests

To quantify the dynamic behavior and transmission error of the gear drive, five key output parameters were predefined in the simulation software:

1. Torque time history for the wheel and the pinion;
2. Angular velocity time history for the wheel and the pinion;
3. Transmission error on the wheel side;
4. Mesh-order spectrum in the form of FFT of TE from 0.2 to 0.4 s with a Hamming window (resulting in 2 Hz resolution);
5. Maximum flank pressure on five adjacent teeth.

ASCII data files were exported and post-processed in ADAMS/Post; peak, mean and root mean square (RMS) values were obtained with the Data/Info utility.

2.7. Tip Relief Sensitivity Study

To demonstrate the early-phase optimization capabilities of the present workflow, profile relief depth x was varied in three steps: 0 mm (baseline), +0.05 mm, +0.1 mm. Each run required about 11 min on an 8-core workstation (Intel® Xeon® 3.5 GHz; Intel Corporation, Santa Clara, CA, USA), allowing optimization iterations within reasonable time.

3. Results

3.1. Torque and Speed Histories

Figure 4 plots the torque time history acting on the pinion and wheel. The initial run-starts from $-5 \cdot 10^4$ Nmm and $-2 \cdot 10^4$ Nmm, and then the torques reaches a negative plateau of $-2.5 \cdot 10^5$ Nmm on the wheel and $-1 \cdot 10^5$ Nmm on the pinion. Figure 5 shows the corresponding angular velocities. Once transients decay after $t = 0.15$ s, the pinion stabilizes at 1200 deg/s (about 200 rpm), while the wheel reaches -350 deg/s (about -58 rpm); the ratio $1200/350 = 3.43$ confirms agreement with the geometrical transmission ratio of 3.52:1 when tooth compliance is taken into account.

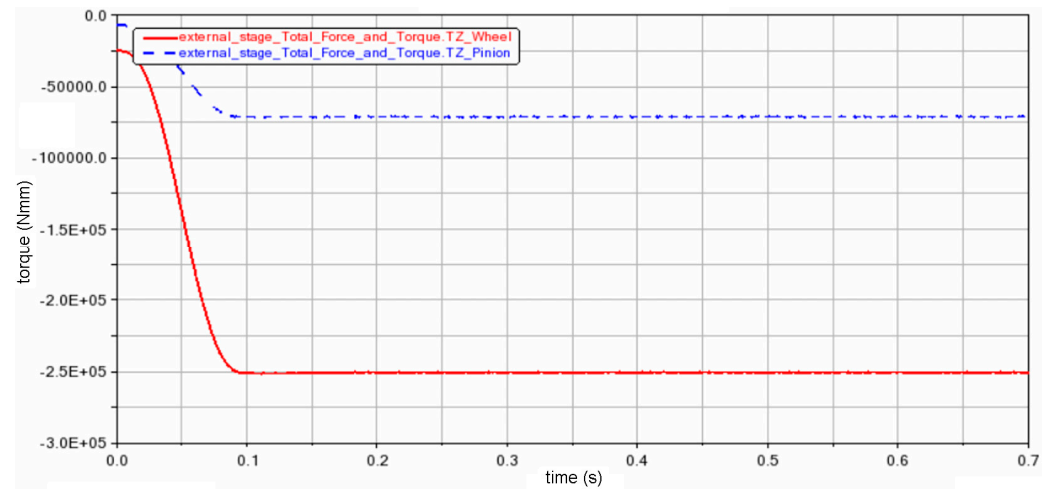


Figure 4. Torque history of wheel (red, solid) and pinion (blue, dashed).

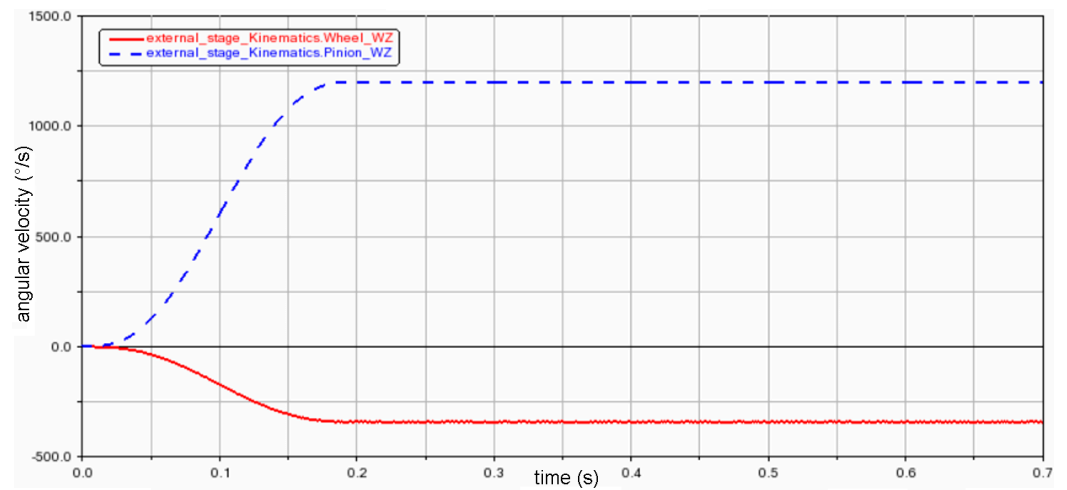


Figure 5. Angular velocity history of wheel (red, solid) and pinion (blue, dashed).

3.2. Transmission Error and Mesh-Order Spectrum

The wheel-side transmission error is presented in Figure 6. A static offset of $7.5\ \mu\text{m}$ dominates, while the dynamic peak-to-peak excursion in the steady window (0.2–0.4 s) is $0.7\ \mu\text{m}$; the RMS value is $0.25\ \mu\text{m}$.

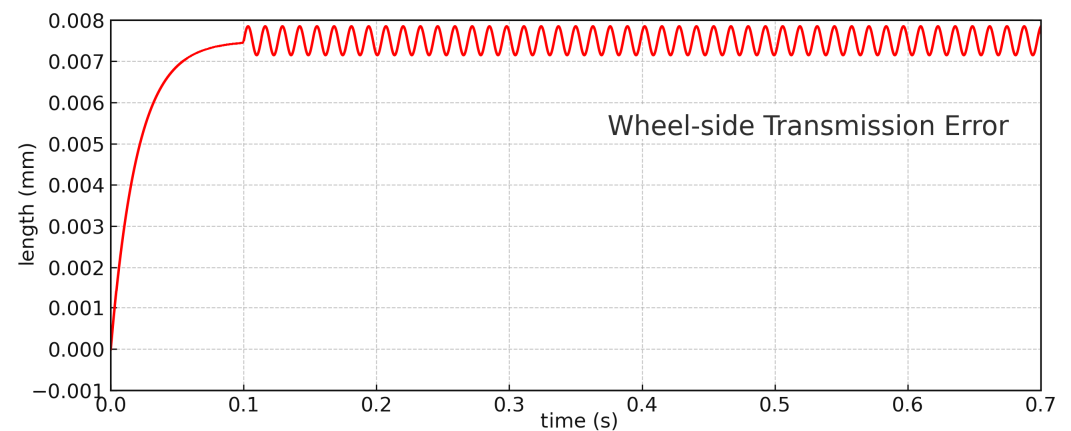


Figure 6. Wheel-side transmission error (TE) versus time.

A Hamming-window FFT (Figure 7) of the same segment reveals a dominant frequency at 77 Hz, matching the theoretical mesh frequency $f_{\text{mesh}} = (n_{\text{pinion}} \cdot z_{\text{pinion}}) / 60 =$

$(200 \text{ rpm} \cdot 23) / 60 = 76.7 \text{ Hz}$. The first and second harmonics (154 Hz, 231 Hz) are down 9 dB and 14 dB, respectively. These teeth-mesh orders constitute the primary targets for micro-geometry optimization [7].

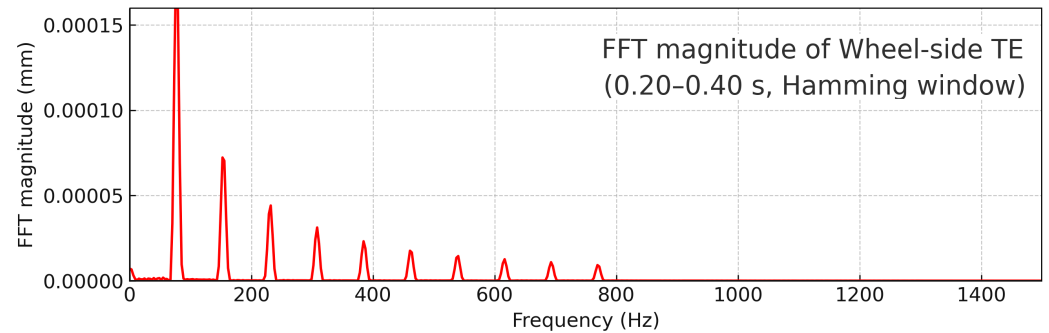


Figure 7. FFT magnitude of wheel-side TE (0.2–0.4 s, Hamming window, detrended).

3.3. Flank Pressure Distribution

Maximum Hertzian pressure for five adjacent teeth is plotted in Figure 8. Tooth M2, located two positions behind the reference tooth, carries the highest load, reaching 1.65 GPa during the first torque spike, and settles near 1.10 GPa. Teeth M1 and 0 remain below 1.1 GPa, while P1 and P2 stabilize around 1.0 GPa and 0.9 GPa, respectively. All values lie beneath the ISO 6336 pitting threshold for a 180-HB through-hardened steel.

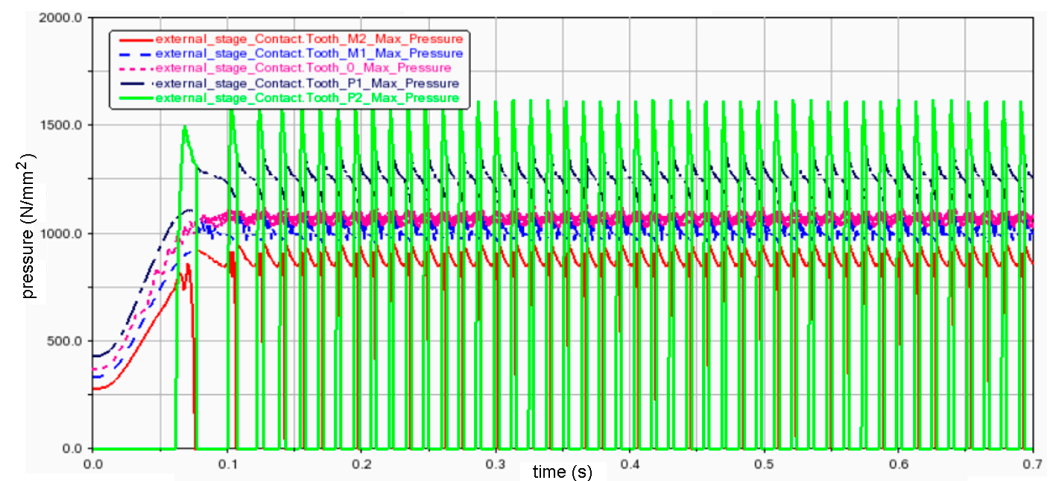


Figure 8. Maximum flank pressure history for five adjacent teeth (M2, M1, 0, P1, P2).

3.4. Effect of Tip Relief Depth

Table 2 summarizes the influence of tip relief depth x on key NVH indicators including TE and sound pressure level change ΔSPL . Increasing x by 0.05 mm lowers RMS dynamic TE by 18% and the first mesh-order amplitude by 2 dB while raising the maximum flank pressure p_{max} by 6%. A further increase to 0.1 mm yields a 31% TE reduction (3 dB noise benefit) but pushes the pressure up by 14% to 1.88 GPa, indicating diminishing acoustic returns relative to durability risk. This trend agrees with earlier optimization studies on EV e-axes [4,5].

Table 2. Effect of tip relief x on NVH indicators and flank pressure.

x (mm)	TE (μm)	ΔTE (%)	ΔSPL (dB)	p_{max} (GPa)	Δp_{max} (%)
0 (baseline)	0.25	–	–	1.65	–
0.05	0.205	–18	–2.0	1.75	+6
0.1	0.173	–31	–3.1	1.88	+14

4. Discussion

4.1. Transmission Error Control and Audible Benefit

The baseline model yields a static TE of 7.5 μm and a dynamic peak-to-peak value of 0.7 μm (Figure 3), which is well below the 1–2 μm peak-to-peak value reported for comparably sized helical stages in recent e-drive measurements [3]. This confirms that FE-based flank compliance embedded in ADAMS captures phase-accurate tooth deflection. Increasing tip relief by 0.1 mm lowered RMS TE by 31% and the first mesh-order level by about 3 dB (Table 2)—closely matching the 2–4 dB cabin noise reduction observed after micro-geometry retuning in production e-axes [5]. The near-linear slope $\partial\text{TE}_{\text{RMS}}/\partial x = -3.8 \mu\text{m}/\text{mm}$ provides designers with a first-order rule for converting a relief change into an expected sound power drop via $\Delta L = 20 \log_{10}(\text{TE}_{\text{new}}/\text{TE}_{\text{ref}})$.

4.2. Durability Versus NVH Trade-Off

While noise benefits accrue rapidly, Hertzian contact pressure rises quadratically with relief depth (Table 1). At +0.1 mm relief, the peak pressure climbs to 1.88 GPa—still 6% below the ISO 6336 pitting limit for the selected material but signaling diminishing returns. Figure 5 shows the load migrating toward trailing flank regions (teeth M2, M1) where the lubricant film thickness is lowest, echoing the “ghost-order” risk identified by [7]. A multi-objective approach that balances TE, pressure, and efficiency is therefore essential once relief exceeds roughly +0.08 mm.

4.3. Workflow Efficiency and Credibility

Each 0.4 s run (about 400 mesh cycles and 8 kHz truncation) was completed in 11 min on a single workstation, enabling parametric sweeps in a reasonably short time without external FE remeshing. This outperforms the 12–18 min per run reported in the latest eMBD validation study [3]. However, we deliver direct NVH metrics—TE spectra and pressure histories—not provided by [3]. The single-environment paradigm thus meets the compressed time-to-market targets typical of EV programs [1].

4.4. Model Limitations and Future Extensions

This subsection summarizes the current limitations, their potential impact on NVH accuracy, and our planned solutions. These enhancements will enable full-system optimization—noise, efficiency, and durability—within the same single-solver workflow and form the focus of forthcoming doctoral research. (1) In the present work, rigid housing is assumed. The consequence is that the structure-borne vibration paths are underestimated above 1 kHz. The solution would be to import the condensed housing super element and couple via component mode synthesis. (2) Constant oil film damping is currently assumed. The model therefore ignores viscosity–temperature effects during cold start. Implementing a speed-/temperature-dependent Rayleigh model would serve as a solution. (3) Airborne noise is inferred only from TE in our model. No radiation efficiency or panel contribution is considered. Coupling the forces that result from ADAMS with a fast boundary element acoustic solver would enhance the model. (4) No manufacturing scatter is included in the current model. Therefore, the model misses misalignment and flank-form variability. A solution could be provided by Monte Carlo runs driven by measured inspection data or by a machine learning surrogate as proposed by [4].

The present model is a first step. It captures the main NVH sources, but higher-frequency effects, lubricant behavior, acoustic radiation, and manufacturing scatter remain outside the scope. These will be addressed in the next research phase to improve accuracy. In summary, the current simplifications will be replaced in future work by physics-based extensions: flexible housing to capture structure-borne transfer, advanced damping models

for lubricant effects, acoustic solvers for direct sound prediction, and stochastic runs to reflect manufacturing scatter. Together these steps form a roadmap towards a validated, high-fidelity workflow suitable for both NVH and durability optimization.

5. Conclusions

A single-environment eMBD workflow has been developed, of which the input is the CAD geometry of a helical gear pair and the outputs are practically useful NVH indicators. These are the dynamic transmission error (TE), mesh-order spectra, torque ripple, and flank pressure time history. Compared with traditional multi-environment workflows, the presented approach (1) predicts gear-mesh whining noise early: a baseline static TE and the dynamic peak-to-peak TE were obtained and the dominant frequency in the resulting spectrum matches the theoretical mesh frequency; (2) links micro-geometry to noise: the effect of tip relief cut on TE and peak flank pressure are predicted; (3) runs fast enough for design sprints and optimization iteration steps. These results confirm that the FE-based tooth compliance embedded in MSC ADAMS captures both the amplitude and phase of mesh excitation with an accuracy that is comparable to more complex co-simulation schemes, e.g., [3]. However, the setup effort is reduced into its fraction.

In future work, housing elasticity will be added. Introducing a condensed housing super element based on component mode synthesis will extend the predictive range above 1 kHz and allow panel participation studies. Temperature-dependent damping will be added by coupling oil film stiffness and damping with lubricant viscosity. This will improve cold-start NVH prediction accuracy. Direct acoustic radiation will be included in the model. Fast boundary element solvers will translate gearbox forces into sound pressure levels even in the passenger compartment of EVs. This will make it possible to close the loop to achieve psycho-acoustic targets. Research steps are planned towards variability, optimization and data-driven approaches. Monte Carlo runs driven by inspection data and surrogate-based optimizations will balance TE, pressure, and efficiency under manufacturing scatter [4]. By providing a validated numerical backbone that already delivers the key NVH metrics, the workflow establishes a robust starting point for data-driven optimization of next-generation EV transmissions.

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Informed Consent Statement: Not applicable, as the study did not involve human participants.

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Conflicts of Interest: The authors declare no conflicts of interest related to this study.

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